IMPROVEMENT OF BARGE TOWING: TRANSLATIONS OF SELECTED JAPANESE AND RUSSIAN TECHNICAL ARTICLES

Translated by Robert Latorre

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PREFACE

In response to commerical demands of river, coastwise, trans-ocean, and offshore operations, numerous barge designs are being developed. One of the problem areas confronted by the barge designer is the towing performance of the barge. This includes the barge towing resistance and the barge course stability during hawser towing.

The towing performance affects the barge's operation, scheduling, and ultimately its commerical success in service. The economy of barge and other waterborn transport is becoming more attractive with rising fuel costs and this will cause more congestion on the waterways. This congestion makes the towed barge course stability a critical safety factor and the rising fuel costs require that the barge towing resistance be reduced as much as possible.

Various Japanese and Russian researchers have studied these problems. They have extended the classic work of Strandhagen, Schoenherr, and Kobayashi published in the 1950 SNAME Transactions to model the effects of the barge skegs and the towing bridle. Systematic barge model tests have been made to study the effects of various skeg designs on the course stability as well as to improve the barge hull resistance. The basic part of this work has not been translated and remains relatively inaccessible to the rest of the maritime community.

The papers included in this report present the results of experimental and theoretical investigations of barge towing performance. They deal with the resistance of simple hull forms, as well as the improvement of the towed barge course stability from various skeg designs, skeg locations, towrope length, and bridle length.

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<div align="right">Robert   Latorre</div>

CONTENTS

ON THE IMPROVEMENT FOR VARIOUS PERFORMANCES OF WORK VESSELS[1]

BY

S. ANDO[2]

ABSTRACT

As ocean development becomes active, construction work using various kinds of work vessels has been increasing. A series of tests were performed in order to improve the form of work vessels with rectangular cross sections. As a result of the experiments it was shown that the propulsive performance can be improved by using a beveled bottom at the stem and stern. The beveled bottom's effect on the vessel's seaworthiness and maneuverability was also investigated. The investigations described in this paper concern the propulsive performance, seaworthiness, for the design, construction, and operation of work vessels.

TRANSLATION[3]

1. INTRODUCTION

The development of Japan's national economy and industrial base calls for improving the efficient use of the ocean regions. There is a rapid increase in harbor civil engineering, large scale construction of deep water harbors, construction of break waters, and the building of sea beds. This is also accompanied by a boom in the construction of undersea cables, oil exploration, and the manufacture of large scale wave observation buoys and towers.

Since these various operations and construction work are done at sea, various types of work vessels play a major role. Consequently, these vessels are classified by their functions and fittings. Typical classifications include: dredging, landfill, support operations, construction, research, as well as inspection.

These vessels usually operate in the calm ocean and harbors so ocean navigation has not been considered as their main activity. Consequently, with the exception of

a few examples, there is a relatively small amount of data and investigations for improving their seakeeping, maneuvering and safety characteristics.

Nevertheless as ocean engineering gradually expands into deep water ocean regions, these various work vessels will increasingly be confronted with severe weather and ocean waves. In addition to this, the importance of work vessels in the ocean transport of equipment and supplies is widely recognized. This is because unlike land based transportation there are no weight or volume limitations. Consequently there is a tendency towards relatively large size work vessels, as well as the construction of larger numbers of work vessels. At this point it has become important to acquire the basic data necessary for investigating the safety characteristics, operational potential, and operational efficiency of each type of work vessel. This data is also useful in work vessel design and construction.

At the Ship Research Institute, Mitaka, Japan, experimental research is conducted in general ship form and performance improvement. At the institute, work vessels with rectangular cross sections were selected from raft and slab type vessels and used in experimental investigations. Systematic tank tests were then conducted to improve the propulsive, seakeeping, maneuvering, as well as the safety characteristics of the vessels. This was done in addition to acquiring data for the design and construction of this type of vessel.

As mentioned earlier, because of the importance of work vessels, this paper summarizes the design trends and various performance improvements for work vessels with rectangular cross sections. The following is a discussion of such vessels with rectangular cross section (hence-

forth referred to as the box form) treat-
ing the propulsive performance, seakeeping
and maneuverability based on the author's
investigations.

2. PROPULSIVE PERFORMANCE

The resistance and propulsive efficien-
cy are essential factors in deciding the
propulsive performance of a ship form.
However since work vessels are usually
towed rather than self-propelled when
moving to their work site, ocean navigation
has not been considered as their major role.
Consequently work vessel propulsive per-
formance is taken as the total resistance.

The resistance generally refers to the
total resistance which is usually classi-
fied into a frictional resistance compo-
nent and a residual resistance component.
With the box form, the form resistance due
to the fore part of the vessel represents
a major portion of the total resistance.
Namely, the residual resistance coeffi-
cient(obtained by subtracting the Schoen-
herr's frictional coefficient from the
ship's total resistance coefficient)is
about 70% to 80% of the total resistance
coefficient. In the case of the box form,
this is because the trim by the bow causes
an increase in the wave and eddy making
phenomena.

Fig. 1 presents the results of the
box form being towed at various Froude
numbers $F_n(V/\sqrt{gL})$ in calm water. This
figure shows the total resistance coeffi-
cient $r_T = (R_T/1/2\rho V^2 \nabla^{2/3})$, trim $1/2(d_F-d_A) \times$
$100/L$, and sinkage $1/2(d_F+d_A) \times 100/L$. As the
vessel is towed water is pushed ahead of
the bow so the trim by the bow and sink-
age becomes larger as the Froude number
increases. From Fig. 1 it can also be
understood that the resistance increases
is aggravated by the vessel's sinkage and
its trim by the bow.

Consequently, in order to reduce the
box form's resistance the vessel's bow
and stern form were modified by cutting
away some of the fullness. This resulted
in improvements in resistance, sinkage
and trim which are described below. The
power estimation was also made for a self-
propelled work vessel.

2.1 REDUCTION IN RESISTANCE

Simple modifications were made to the
vessel's bow and stern by cutting away
some material as illustrated in Fig 2.
This figure shows the beveled bottom and
beveled side modifications.

i) Beveled Bottom Bow and Stern [1], [2]

The three types of beveled bottoms
used in the bow are shown in the upper por-
tion of Fig. 2. In case a) the bevel lines
go through the intersection of the forward
perpendicular and waterline and the bevel
angle α_B is measured from the perpendicular.
As shown in Fig. 3, there is a tendency for
the resistance coefficient, trim and sink-
age to decrease as the value of α_B increases.

This reduction in the resistance coef-
ficient with α_B appears in the results for
various L/B and B/d ratios in Fig. 4.

For the various cases shown in this
figure there is a reduction in the resis-
tance coefficient up to values of α_B near
60°. Conversely, a stern bevel has little
effect on the resistance coefficient until
the value of the stern bevel angle $\alpha_s = 60°$
as shown in the middle graph of Fig. 4.
This may be caused by the effect due to the
wake fraction at the stern and base drag.

In case b) the bevel lines pass through
a point defined by $\alpha_B = 50°$. Here the plot
of the results for the bevel angle θ_B have
been omitted. However, the trends showed
that the minimum values for the resistance
coefficient, trim and sinkage were in the
vicinity of $\theta_B = 22°$. In case c) the bevel
lines form a constant angle of $\theta_B = 20°$.

The reduction of the total resistance coef-
ficient with the amount of material removed
at the F.P. (ℓ_d) is shown in the bottom graph
of Fig. 4. In this figure, up to $\ell_d = 50\%$ there
is a large reduction in the resistance coef-
ficient and for values of ℓ_d larger than

50% there is only a slight reduction in the
resistance coefficient.

From the above results a value of α
around 70° is the best for making the bow
and stern bottom bevel from the viewpoint
of resistance trim and sinkage. However,
since navigating the ocean is not the
main activity of work vessels, from con-
siderations of efficiently using the work-
ing area and loading cargo it is good to
use a bevel angle $\alpha = 70°$ taken at a point
of 1/2 draft.

ii) Beveled Sided Bow and Stern [1],[2],[5]

Modifications of the bow and stern
sides are accomplished by making simple
miter cuts or circular cuts to develop the
three designs shown in the bottom of Fig.2.
In this figure the side modification a)
is made by starting at the point at the
side determined by a taper with $\beta_B = 50°$.
The resistance coefficient relationship
with the bevel angle β_B is shown in the
upper graph of Fig. 5. In Fig. 5 the
total resistance coefficient has its mini-

mum value when β_B is approximately 15~20°. Also while the figures are not shown, there was almost no variation in the trim and sinkage when β_B was changed. Following these tests, the stern was modified in the same manner, and there was almost no variation in the resistance coefficient as indicated by the results in the middle graph of Fig. 5.

The radius (R) cut b) was reported by Ikebuchi of the Shipbuilding Research Center of Japan in the magazine Senpaku [5]. The results are shown in the bottom graph of Fig. 5. When the radius R becomes smaller this graph shows the drastic reduction in the resistance coefficient. When this radius becomes larger its effect on the reduction in the resistance coefficient becomes small. It is natural for the ogive cut c) that there is a reduction in the resistance coefficient when the radius of the ogive becomes larger.

From this discussion, a bow bevel angle β_B = 15° is the optimum angle for bow side bevel angle of work vessels.

iii) Various Form Modification Schemes [1], [2], [3]

It appears that the bow and stern of work vessels will be assembled using more complex configurations than the simple modifications previously discussed.

Therefore, Fig. 6 shows a comparison of the total resistance coefficients of box shaped vessels with four types of complex bow shapes. From the results shown in this figure, the spoon bow form (d) has the lower resistance coefficient in the low speed region while at higher speeds the bow form (c) has a lower resistance coefficient than the spoon bow (d).

It appears from these results that it is possible for such box shaped vessels to obtain good resistance performance using comparatively simple beveled bow forms instead of adopting more complex forms.

2.2　FLOW AROUND THE HULL

The hull form found superior in the systematic resistance test series is shown in Fig. 7. It is a 1/18th model of an oil drilling vessel and this model was used to investigate the flow around the vessel's sides and the wake at the stern.

i)　Flow Around Bow and Sides [3]

In order to observe the flow, 40 mm long tufts were attached 10mm apart from the surface of bow's slanting part and the vessel's sides. The flow was then observed while towing.

The flow pattern along the vessel's side at four different model speeds is

shown in Fig.8 as well as Photo 1 [Photo 1 has been omitted because it does not reproduce well]. The flow pattern along the side near the free surface has the same form as the vessel's wave profile. The flow near the vessel's bottom is aligned with the direction of the main flow. It appears that the flow along the vessel side has extremely simple streamlines.

Although the figure is not shown, it was found that the flow does not pass around the vessel's sides unless the bevel angle of the bow is near 20°. If the angle is different the flow does not pass around the sides and this results in a complicated flow along the bottom of the vessel.

ii) Wake at Stern [3]

The stern flow velocity distribution was obtained by using a spherical five hole pitot tube to measure the flow in the x, y and z directions. The measurement points are shown in Fig. 9. This is a detail of the upper left hand portion of Fig. 7. The results are shown in Figs. 10 and 11 (In these Figures the dashed arrows in the β section represent the measurement without the skeg).

The flow velocity in the x-z direction shown in Fig. 10 follows closely the bevel of the stern. Although the flow velocity in the y-z direction in Fig. 11 is nearly perpendicular along the hull centerplane, in the vicinity of the vessel's sides there is a strong tendency for it to flow outward.

From the above it appears that the skeg exerts very little influence on the notched section in Fig. 7.

The wake fraction at the aft sections are shown in Figs. 12 and 13. Fig.12 shows that because the skeg is fitted at the notched section the wake fraction values are very small irrespective of the model speed and location of each section. However, the extent of the notched section's influence on the wake fraction is evident in Fig. 13, showing the wake fraction with and without the skeg fitted. Therefore, from the viewpoint of the vessel's resistance and maneuverability performance it is worthwhile to fit skegs to the notched section.

2.3　ESTIMATION OF SHAFT HORSEPOWER FOR SELF-PROPULSION OPERATION [3]

In this section a summary is presented of the oil drilling vessel's estimated speed when it is self propelled using the digging equipment's prime mover.

This vessel has a length of 70m, a width of 50m and a draft of 4.58m and is

fitted with a 5,000 P.S. digging equipment's prime mover. The study was made for twin screw, twin rudder vessels disignated as MA and MB. The MA vessel is shown in Fig. 7 while the MB form is fitted with bow section (c) shown in the upper part of Fig.6. The propellers had a MAU blade section, a diameter of 0.1787 m., and were not optimally designed for both hull forms. Their open water characteristics are shown in Fig.14. Fig. 15 shows the self propulsion coefficients measured from the self-propulsion tests with outboard turning propeller. The propulsive efficiency in this figure is extremely low in the range of .40. This is because the propellers tested were not optimum designs for the hull forms. Nevertheless it is possible to attain a speed above 6 knots when using the 5,000 horsepower for propulsion as shown in Fig. 16.

From the above it appears feasible to use the work vessel's machinery for propulsion to enhance its economic and safety effectiveness.

3. SEAKEEPING PERFORMANCE

Since the various types of work vessel forms were evolved from raft type vessels the work vessel hull has a nearly rectangular cross and these vessels have a wide beam in comparison to their length.

Thus, work vessels have a relatively superior hull form in comparison with general cargo or fishing vessels in regards to response in waves and related safety characteristics. Here an investigation is presented concerning the work vessel's Bertin's N coefficient which appears in the regulations related to the ship motion restoring performance. This influences the freeboard due to the relative motion of the vessel and wave surface. It also is used in estimating accelerations required for the design of the crane jib and digging tower. An investigation of the vessel's response in waves is also discussed since there is a wide variation in the center of gravity and radius of gyration of work vessels in comparison with those of the typical ship hull. In addition to this study the following discussion summarizes the effect of modifying the bow and stern on the response of the vessel in waves.

3.1 BERTIN'S N COEFFICIENT FOR BOX FORM VESSELS [8]

Free rolling tests were made in calm water for a wide range of conditions varying the center of gravity and radius of gyration of the models. The models used were a floating crane, grab dredger and pump dredger which have nearly rectangular cross section form. The Bertin's N coefficients obtained in these tests are shown in Fig.17. In this figure the black triangles

indicate the Bertin's N coefficient for a general cargo ship filled with bilge keels. The N coefficient of the rectangular cross section work vessels is an extremely large value which is about 2.5 times the value of the general cargo ship fitted with bilge keels. Although the figure is omitted, the effective wave slope coefficient (γ) in the roll synchronism conditions was determined from the responses obtained from the forced rolling moment in calm water and in waves. For work vessels with OG/d less than 4.0, the value of γ has an extremely small value, $\gamma = 0.07 + 0.175$ OG/d in comparison with that of a general cargo vessel [10], $\gamma = 0.73 + 0.60$ OG/d, whose center of gravity is in the vicinity of its waterline.

From the above facts it can be said that as far as response in waves is concerned, the work vessel has a superior hull form.

3.2 VARIATION OF RELATIVE WATER LEVEL ALONG THE VESSEL SIDES AND VERTICAL ACCELERATION [8]

In order to investigate the amount of variation in the water level along the sides of work vessels, tests were made in head (longitudinal) and beam (transverse) waves and the variation in water level at L/10 from the A.P. and F.P. on both sides of the vessel hull. An example of the results is shown in Fig. 18.

The upper portion of Fig. 18 shows the amount of water level variation in head waves with OG/d = 2.1, $K_{yy}/L = 0.53$ and L/B = 2.2. As expected, this figure shows that the variation in water level is the same along the port and starboard sides, and that the wave height reaches a value of 75% of the draft when the wave height and length ratio is 1/75. In the beam waves, the windward water level variation is about 3 times that of the lee side with short waves whose length is less than double (4 times the beam). This is because the vessel's hull acts as a wave absorber. However, for long waves this wave absorbing effect of the vessel hull diminishes and the differences in the port and sides disappear.

A general investigation of the accelerations was made for the structural strength design of the crane jib or the tower used on the work vessel. The vertical accelerations were measured at a height 1.0L from the vessel bottom and 0.45L forward of the bow for the test conditions: L/B = 2.2, B/d = 6.0, OG/d = 3.5, $k_{yy}/L = 0.53$, $k_{xx}/B = 0.40$. The vertical acceleration response in head and beam waves is shown in Fig. 19. From this figure it can be seen that the values of vertical acceleration (solid line) are much larger than the vertical accelerations estimated from the heaving motion in head waves. This may

be due to the high location where the acceleration was measured. The solid line in this figure shows the acceleration value predicted from the pitching motion, taking the distance between the measurement point and the intersection of the center of bouyancy and waterline as its lever. Therefore for work vessels with a relatively high center of gravity the vertical accelerations can be estimated by considering the pitching motion of the vessel.

The vertical accelerations in beam waves can be approximately estimated from the heaving motion (dashed line). This is because the rolling motion has a second order effect on the heaving motion (the center of the rolling motion is in the neighborhood of 80% to 90% KG).

This same tendency can be seen when a load is hung from the crane jib. (denoted as load in Fig. 19)

3.3 EFFECT OF ROUNDING THE BILGE [7]

As far as rolling motions are concerned vessels with a rectangular cross section possess a large roll damping coefficient. However, the use of construction ropes and chains used in the work vessel service requires adopting a bilge radius. Therefore, forced roll oscillation tests were conducted in calm water using two dimensional rectangular forms and gradually increasing the radius of the bilge at the vessel bottom.

Assuming the equation of rolling motion is given by:

$$(J_{\phi\phi}+A_{\phi\phi})\ddot{\phi} + B_{\phi\phi}\dot{\phi} + W \cdot GM = M_{\phi e}$$

Fig.20 shows the following values

$$(J_{\phi\phi}+A_{\phi\phi}) / \rho\nabla B^2, \quad B_{\phi\phi} \cdot \sqrt{B/2g} / \rho\nabla B^2$$

versus the bilge radius (r).

From this figure for the values of $r/B < 0.08$ there appears almost no variation in the virtual moment of inertia, while up to values of $r/B = 0.04$ there is a drastic reduction in the roll damping moment. This indicates that from the viewpoint of rolling motion it is advantageous to adopt a small bilge radius.

3.4 EFFECT OF CENTER OF GRAVITY AND RADIUS OF GYRATION ON ROLLING MOTION.

To estimate the rolling motion performance of various work vessels, wave tests were conducted with the vessel length, beam, draft, center of gravity as well as radius of gyration being systematically varied. In head waves the pitching motion is small as long as the wave length is below 0.8 times the vessel length which

is the same for typical ship hull forms. While for longer wave lengths at the maximum pitching values, the angle of pitching at its synchronism condition is 1.2 times the maximum wave slope value. The heaving motion has the same value as the wave height.

In beam seas, neglecting special situations, the maximum roll angle in its synchronism condition is 3 to 4 times the maximum wave height which is larger than the heaving motion measured in head waves.

From the above it is concluded that the major problem of the work vessel response in waves is its rolling motion.

The response performance of a model with L/B = 2.2, B/d = 10.0 is shown in Fig.21 illustrating the effects from changing the center of gravity and radius of gyration. Changes in the center of gravity have a larger effect on the rolling motion than changes in the radius of gyration. There is a tendency to reduce the roll angle when the center of gravity is raised because it appears that when the center of gravity is raised the wave length correspondingly becomes longer and the wave slope angle becomes smaller in conditions of constant wave height.

The usual method for estimating approximately the magnitude of the motions in waves is done by using the Ordinary Strip Method (OSM) with the hydrodynamic forces on the two dimensional section determined by the Ursell-Tasai analysis or the source distribution method. With the exception of roll damping the theoretically determined hydrodynamic forces and wave excited forces are in agreement with the experimental values. The experimentally obtained damping coefficient is assumed to be divided into wave damping and viscous damping components. In addition the experimentally obtained damping moment A_R and the theoretical damping moment A_{ro} obtained from the Ursell-Tasai or source distribution method are assumed to be related by

$$A_R = \sqrt{A_{RO} + b\xi\frac{n}{d}}, \quad B_{\phi\phi}=\rho g^2 (A_R \cdot B/2)^2/\omega^3$$

where the constants b and n were shown by Okawa [9] to give the best approximation when b=0.70 and n=2.5.

3.5 EFFECT OF BEVELED BOW AND STERN

The effectiveness of beveled bottom and sides in improving the propulsive performance was discussed in section 2.

This led to towing tank tests which investigated the effects on the responses in waves from beveling the box form vessel's bow and stern. These experiments were made in head and beam waves while maintaining the following parameters constant:

$L=1.80$ m, $L/B=3.0$, $B/d=5.0$, $KG/d=1.67$ $K_{xx}/B=0.35$, $K_{yy}/L=3.0$. The bottom bevel modifications (b) and (e) as well as the mitered beveled side modification (a) shown in Fig.2 were tested.

i) The influence of the bow bevel angle θ_B on the vessel motions is shown in Fig. 22. From this figure it appears that the pitching and heaving motions in head waves and the heaving and swaying motions in beam waves are nearly unaffected by the bevel angle θ_B .

However, the rolling and surging motions appear to be strongly influenced by the value of θ_B . Regarding the rolling motion, it appears that when the bevel angle increases above $\theta_B=20°$ there is a tendency for the roll angles to increase. As for the surge motion, it appears that as the resistance becomes smaller as θ_B increases there is an increase in the surge motion.

ii) The influence of the bow side bevel angle β_B on the motions is shown in Fig.23. In this figure it appears that the miter angle β_B has a small influence on the motions in head waves. Namely near the value $\beta_B=20°$ there is a reduction in the motion. On the other hand, the value of β_B has almost no influence on the the motions in beam waves.

iii) The influence of l_d/d on the motions is shown in Fig.24. In these tests the bow bevel angle was maintained at $\theta_B=20°$

The influence of l_d/d on the motions is the same as for θ_B namely it affects only the rolling and surging motions and there appears to be no effect on the other motions. Considering the influence of l_d/d on the rolling and surging motions, they are reduced when the value of l_d/d is small.

From the above results it can be concluded that a bow form having a bevel angle near $\theta_B=20°$ with a spacing of $l_d/d = 0.5$ and a side miter angle $\beta_B=15°$ which is superior from standpoints of resistance trim and sinkage does not result in poor responses in waves in comparison to those of rectangular form.

4. STEERING PERFORMANCE [1],[3]

Since navigation has not been a major role of work vessels their steering performance has not been considered a major problem. However, from considerations of safety and ecohomy, it is expected in the future the power used for the operations will be utilized for the vessel's propulsion and it is therefore necessary to investigate the vessel's steering performance.

Initially oblique towing and turning tests were made with models having rectangular cross sections. These models had $B/d=10.0$ and length to beam L/B ratios of 1.0,1.5,2.25 and 3.0. The three force components (drag force X, normal force Y, and moment around the center of flotation M) were measured. The results indicated that the trends in the forces for the $L/B=1.0$ model were extremely different than those of the other L/B ratios. (This figure is omitted). The lift force coefficient C_N obtained from the test data of the three forces is shown in Fig.25. The trends in this figure for $L/B=1.0$ lift forces is shown in Fig.25. The trends in this figure for $L/B=1.0$ lift coefficient are extremely different in comparison with the other L/B ratios. The lift force at the beginning of the turn for the $L/B=1.0$ corresponds to the lift force in the opposite direction for L/B ratios larger than 1.5. In these experiments neither propellers nor rudders were fitted. It can be anticipated that when they are fitted the vessel will run in a straight direction in similar situations since the rudder angle used in turning does not result in a large turning force.

Next the self-propelled turning tests were made with these rectangular form work vessels with twin screws and rudders. The results of these tests are presented in the form of non-dimensional turning angle velocity (vessel length/turning radius), maximum advance, maximum transfer, as well as turning radius indicated by the solid lines in Figs. 26 and 27. It can be understood from this figure that below values of $L/B-1.0$ turning is difficult.

Following these tests, self propelled turning tests were conducted using a beveled bow whose angle is 22° fitted to the previously described work vessel models The results of these tests are indicated by the dashed lines in Fig.26 and 27. It can be understood from these figures that the beveled bow causes a large improvement in the turning performance and there is almost no reduction in the course keeping performance. However, for L/B ratios less than 1.0 adopting a beveled bow does not result in an improved turning performance as with the rectangular form

The results of an investigation of the effect of using the beveled bow modification (h) in Fig.2 on the turning performance is shown in Fig. 28. From these results good turning performance is obtained,especially when the bevel angle

$\theta_B=22°$ and the non dimensional angular turning velocity becomes large. Namely, at this value the turning radius is small. At the same time as far as obstacle avoidance is concerned it is important to have a maximum advance and a maximum transfer while at $\theta_B=22°$ these values are minimum.

From the above it is understood that the beveled bottom is also able to change the turning performance.

5. CONCLUSION

The previous discussion has treated the main results obtained from systematic model tests of the resistance, seakeeping, and turning performance of work vessels with rectangular cross sections tested at the Ship Research Institute. The following is an enumeration of the details for the improvement of work vessels by the synthesis of the experimental results including the experimental results not included in this presentation.

i) For the work vessel which has a good resistance performance without making major changes to the hull form it is good to bevel the bow and stern using $\theta_B=20°$ measured from a point half the full load draft and adopting a mitered side with an angle of $\beta=15°$. Also, if possible it is good to round the bevel ends. If it is not possible to use mitered sides, then it is good to use an R/B=0.05 radius. It is also worthwhile in case of self propulsion to use a beveled stern which makes an angle $\theta=20°$ from the bottom to 80% the full load draft.

It appears there remains further possibility to improve the work vessel hull form. After many years of experience, research, and development, the apparent ideal hull form for very large full ship forms ($C_B=0.81$, L/B=5.8, B/d=3.1) was arrived at which has a total resistance coefficient around 0.02=0.04 at a Froude number near 0.2. In comparison a rectangular hull form of the same proportions with the previously described beveled bow and stern has a total resistance coefficient of 0.04-0.06.

ii) When the beveled bow and stern are adopted (as described earlier) to improve the resistance characteristics, they do not have a bad influence on the vessel motions in waves compared to the response of the rectangular hull form.

Because there are many occasions when mooring or working cables are used on the work vessels it is necessary to round the bow, stern and bilge in order to prevent the cables from breaking. For this requirement the previously recommended radius is adequate. In order not to re-

duce the roll damping force by rounding the bottom at the bilge, it is advisable not to make it round but use a radius within 1/100th of the beam (R/B-0.01).

iii) A difficult point in existing work vessels with rectangular cross sections is their turning performance. For the large full ship ($C_B=0.81$, L/B=5.5, B/d=2.6) at a 40° rudder angle the non-dimensional angular turning velocity is 1.0 while in comparison, the previously described optimum hull form has a non-dimensional angular turning velocity of 0.5 to 0.6.

The major factor influencing the work vessel's turning performance is its rectangular cross section. The rectangular cross section is different from the usual ship's midship section featuring a bilge radius.

In addition, it is not anticipated that adopting beveled bottom or sides on vessels with the same length and width (L/B÷1.0) will improve their turning performance.

NOMENCLATURE

L	vessel length
B	vessel beam
d	vessel draft
d_F	forward draft
d_A	aft draft
∇	volumetric displacement
W	displacement
V	vessel speed
F_n	Froude number
R_T	total resistance
r_T	total resistance coefficient
α_B	bow bevel angle [Fig.2]
α_s	stern bevel angle [Fig.2]
	bow bevel angle [Fig.2]
ℓ_d	bow cutting height [Fig2]
β_B	mitered bow angle
β_s	mitered stern angle
v_x	flow velocity in X directions in stern
v_y	flow velocity in Y direction in stern

V_z flow velocity in Z direction in stern

V_m model towing velocity

J advance coefficient

K_T thrust coefficient

K_Q torque coefficient

η_o Propeller open water efficiency

η Propulsive efficienty

η_R relative rotative efficiency

1-t thrust deduction factor

DHP delivered horse power

X_a surging amplitude

Y_a swaying amplitude

Z_a heaving amplitude

θ_a pitching amplitude

ϕ_a rolling amplitude

λ wave length

k wave number ($2\pi/\lambda$)

N Bertin's damping coefficient

ω circular frequency

$A_{\phi\phi}$ added mass moment of inertia

$B_{\phi\phi}$ damping moment

$J_{\phi\phi}$ mass moment of inertia

A_R non-dimensional damping moment

ξ_d non-dimensional circular frequency ($\omega^2 d/g$)

η_a water level amplitude on vessel sides

$\ddot{Z}_a$ vertical accelerations

C_N lift coefficient

β drift angle

δ rudder angle

r' non-dimensional angular velocity

D_A maximum advance

T_R maximum transfer

D_T turning circle

$\overline{OG}$ distance between center of gravity and waterline

$\overline{KG}$ distance between center of gravity and bottom

$\overline{GM}$ distance between center of gravity and bottom

K_{xx} radius of gyration for rolling

K_{yy} radius of gyration for pitching

C_B block coefficient

ζ_a wave amplitude

ρ water density

g acceleration of gravity

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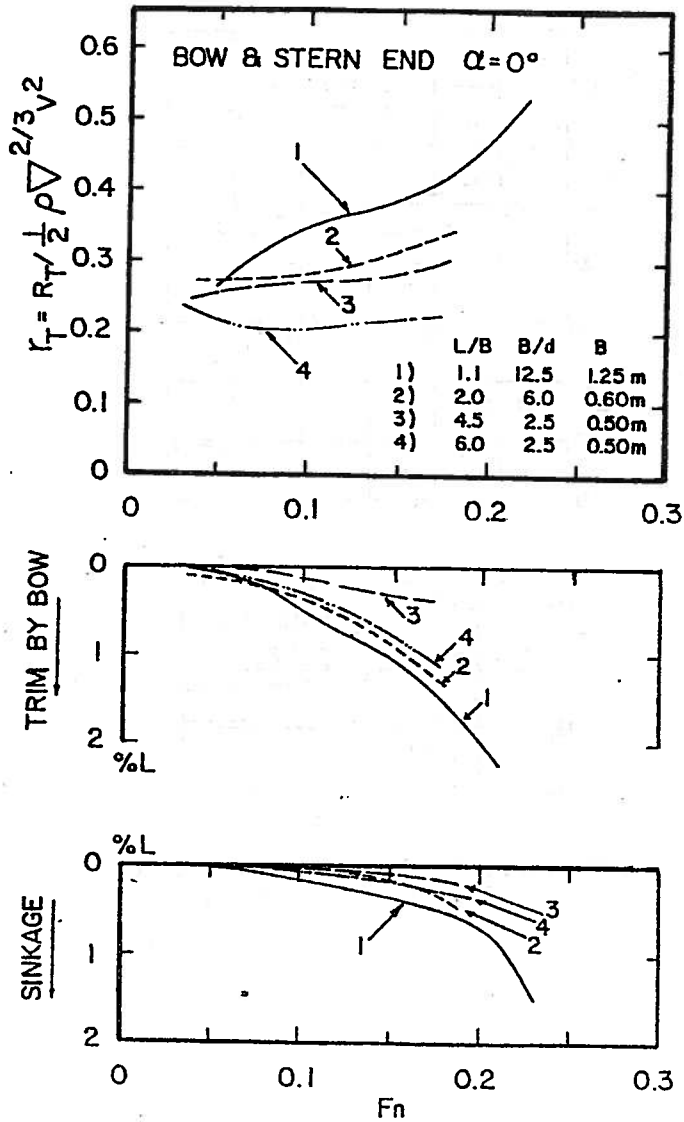
FIG. 1 Total Resistance Coefficient, Trim, and Sinkage of Box Shaped Vessels

FIG. 2 Side and Bottom Bevel
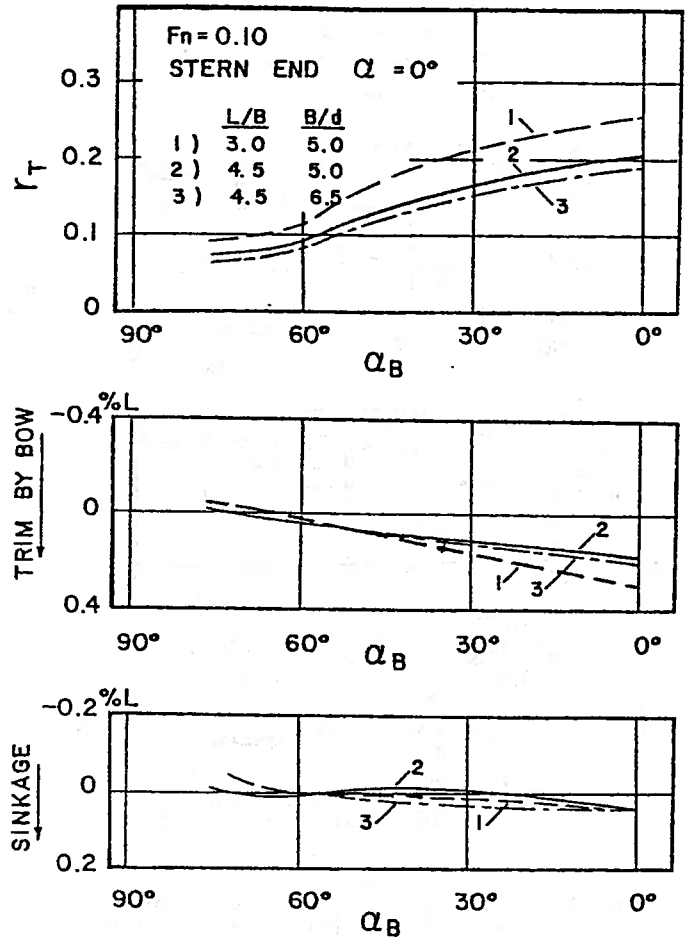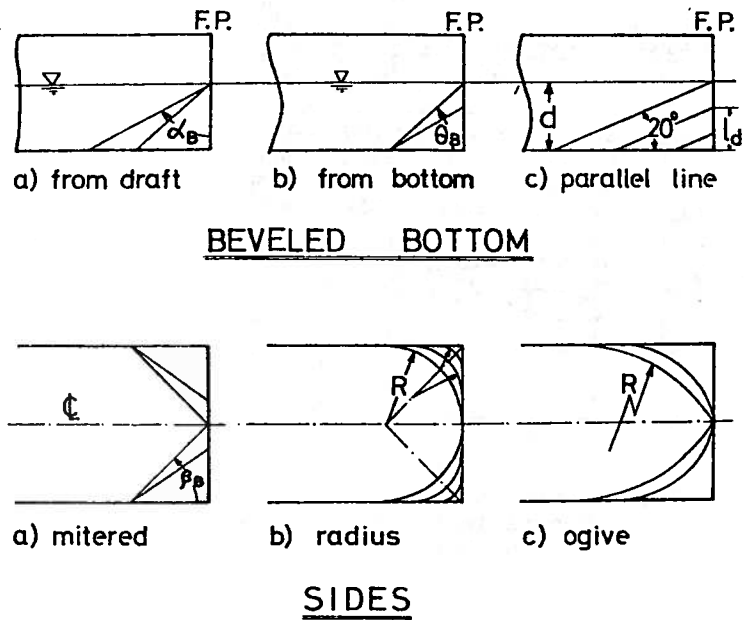
FIG. 3 Effect of Bow Bevel, α_B, on Total Resistance Coeff., Trim and Sinkage

a) from draft   b) from bottom   c) parallel line

BEVELED BOTTOM

a) mitered   b) radius   c) ogive

SIDES
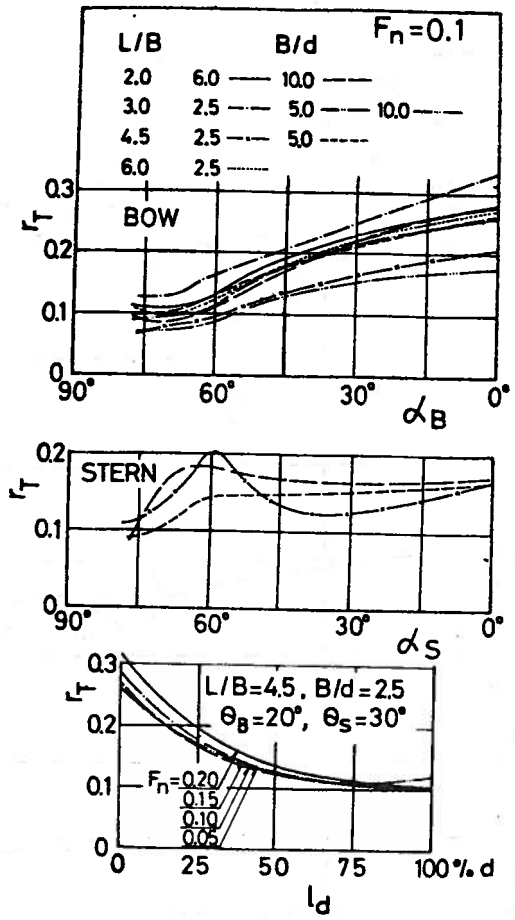
FIG.4 Variation in Total Resistance
Coefficient with Bottom Bevel
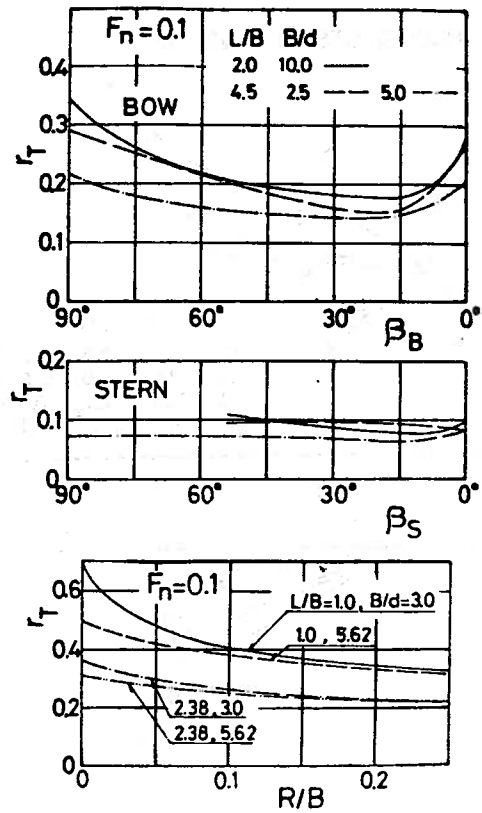
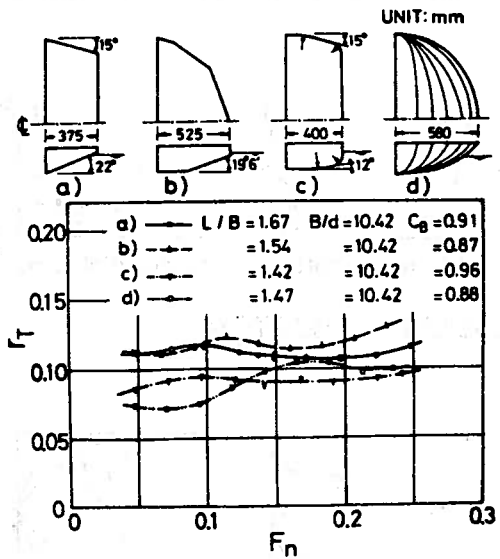FIG. 5 Variation in Total Resistance
Coefficient with Side Bevel

FIG.6 Comparison of Total Resistance
Coefficients of Different
Complex Bow Forms
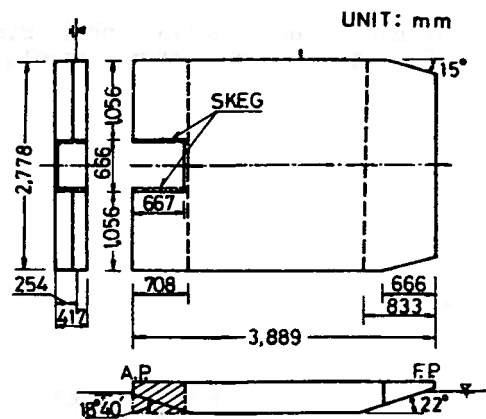
FIG. 7 Plan of Oil Drilling Vessel's
Hull Form

-13-
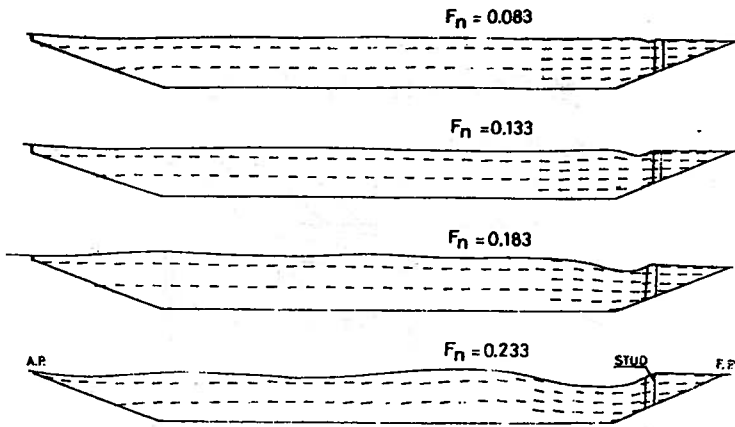
FIG.8 Flow Pattern Along Vessel Side
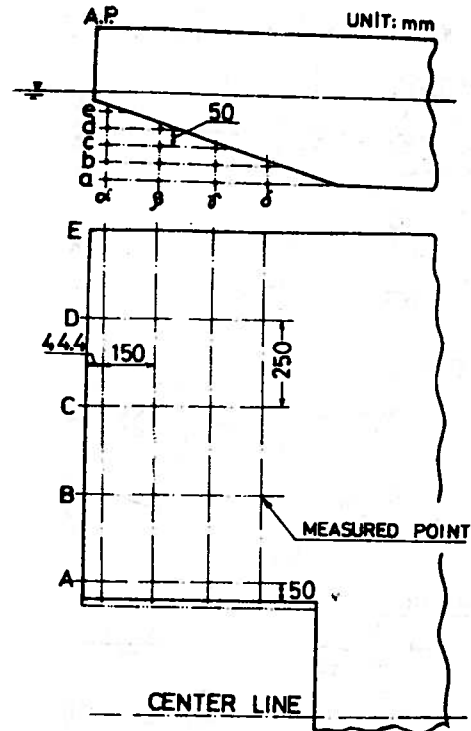
FIG. 9 Location of Flow Measurement on Vessel's Stern
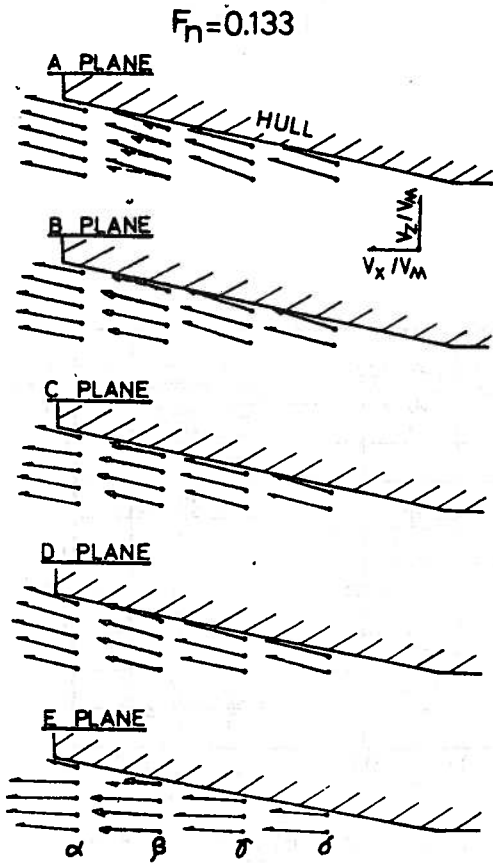
FIG. 10 Velocity in the x-z Direction on Vessel's Stern
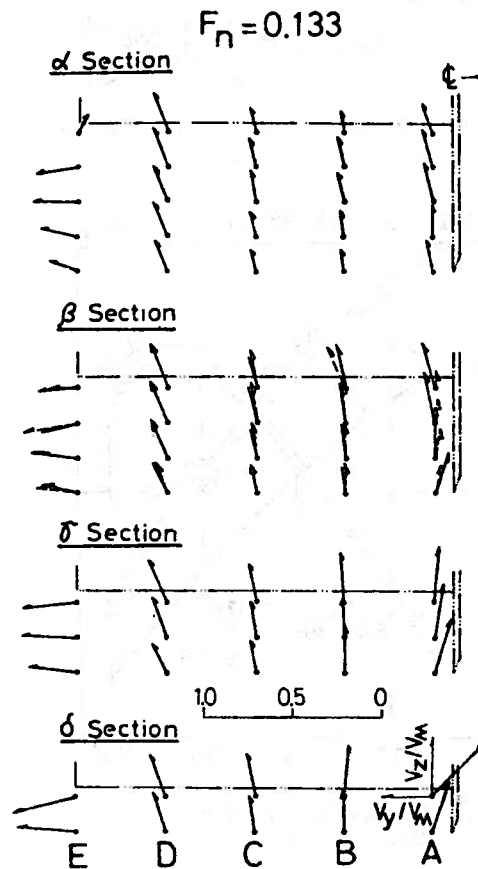
FIG. 11 Velocity in the y-z Direction on Vessel's Stern

With Skeg $F_n = 0.133$

α Section

HULL

0.3
0.2
0) 0.1
0.2

β Section

0.3   0.2
0.1
0) 1.0−V_X/V_M=0.1
0.2

γ Section

0.2
0.1
0
0.1

δ Section

0.1
0.1   0.2
0
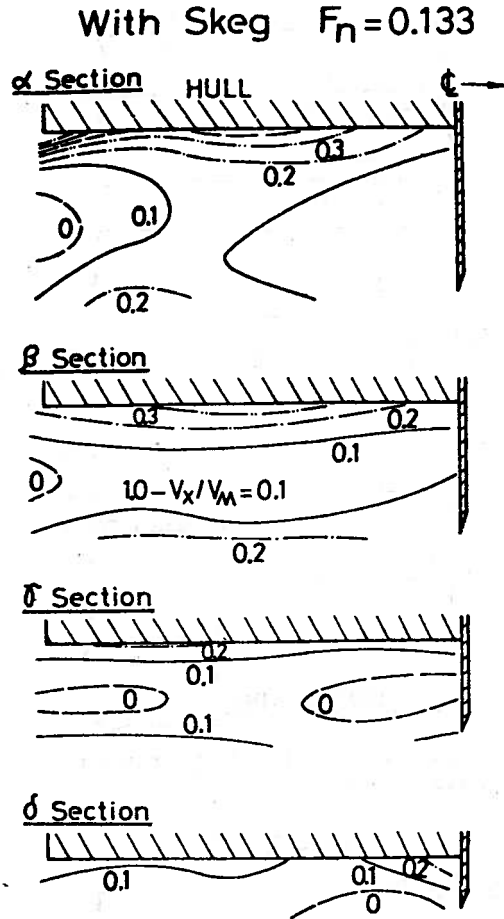
FIG. 12   Wake Fraction along various
Stern Sections

β Section $F_n = 0.133$

WITH SKEG

HULL

0.3
0.2
0.1
0) 1.0−V_X/V_M=0.1
0.2

WITHOUT SKEG

0.3 0.4 0.6
0.5
0) 0.1 0.2
0.1
0.2
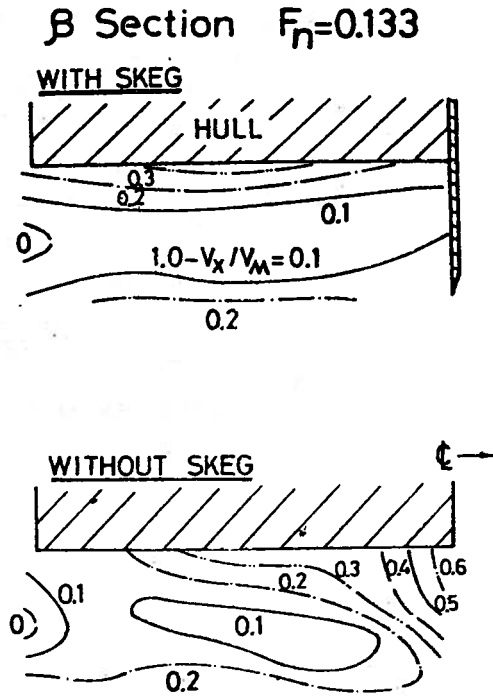
FIG. 13   Compariosn of Stern Wake
Fraction with and without
Skegs Installed

MP NO. 0036 (MAU Type)

K_Q

η_0

K_T

RIGHT
LEFT

Dia   0.1787m
Boss ratio 0.18
Pitch   0.1429m

K_T, $10K_Q$, $2\eta_0$

J
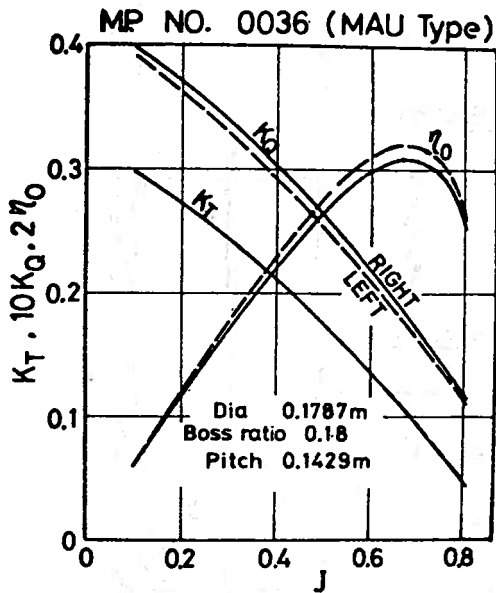
FIG. 14   Propeller Open Water
Characteristics

η   MA1 η
MA2   MB2
MB1
MB1'

η_R η_R η_R

1−t   1−t

1−w_T 1−w_T

F_n F_n
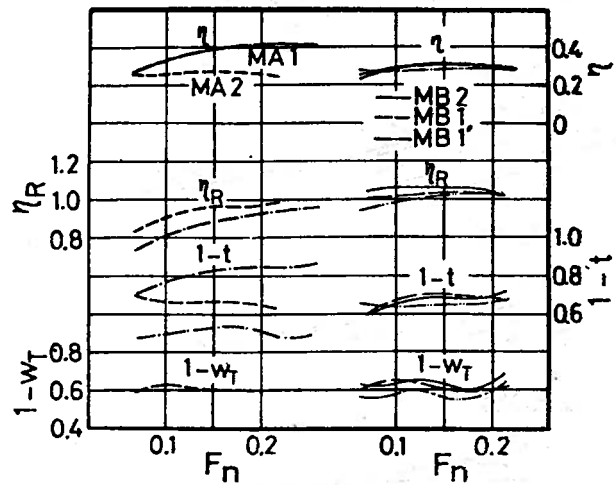
Fig. 15   Self Propulsion Coefficients
for each Hull Form

-15-
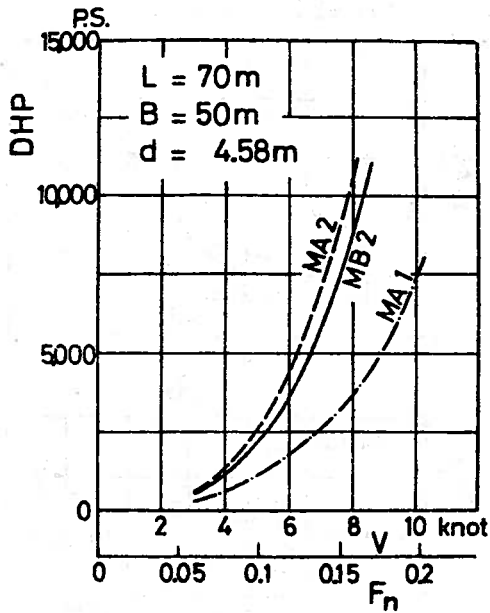
FIG. 16 Estimated Required Power
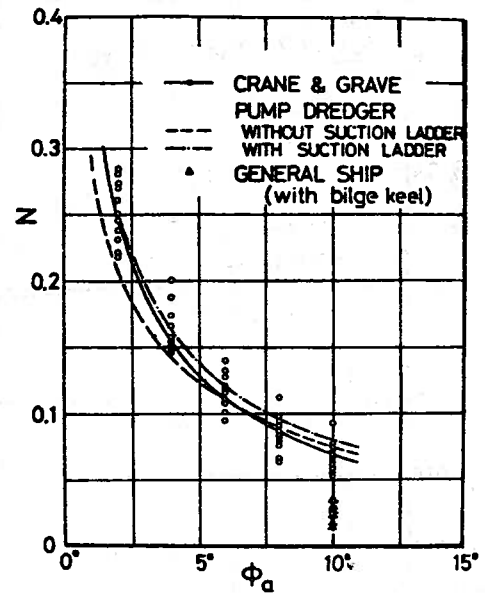
FIG. 17 BERTIN'S N Coefficient for Work Vessels
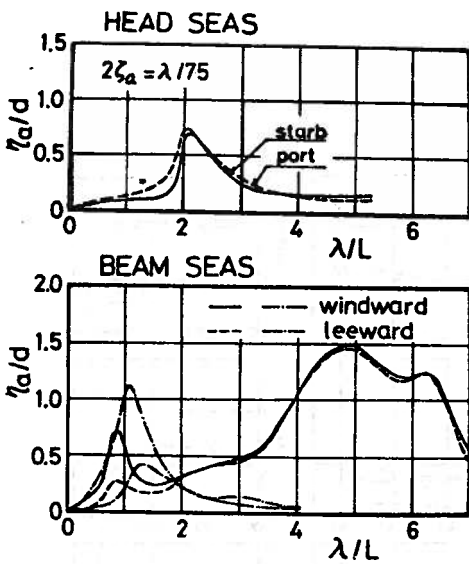
FIG. 18 Variation of Water Level along Sides of Work Vessels
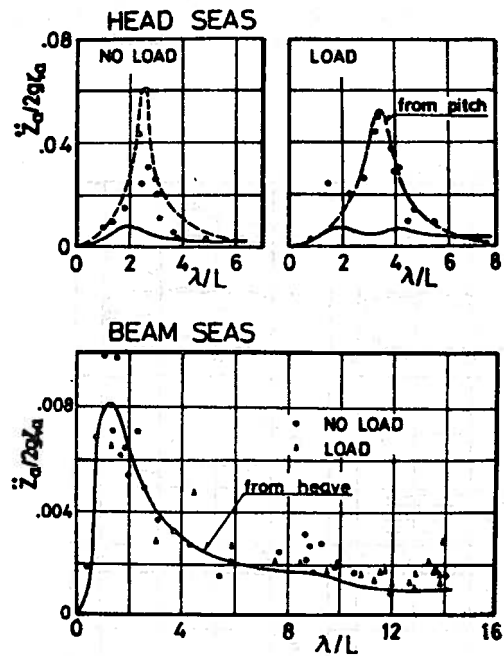
FIG. 19 Vertical Acceleration of Floating Crane with and without a suspended weight
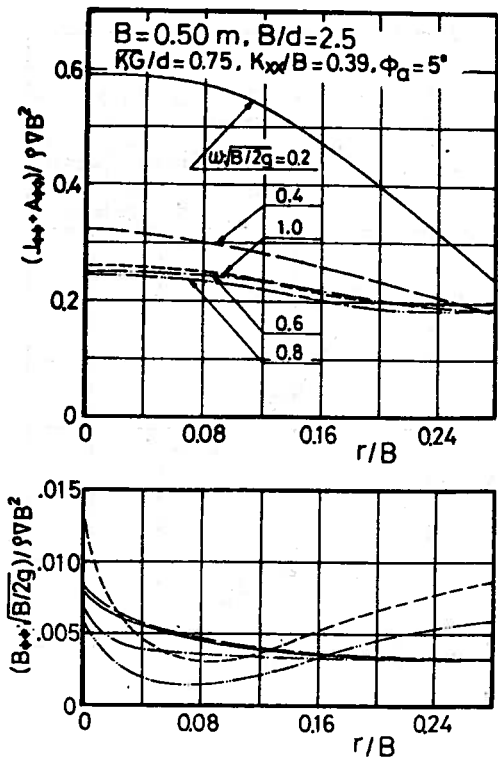
FIG. 20 Influence of Bilge Radius on the Virtual Moment of Inertia and the Roll Damping Moment
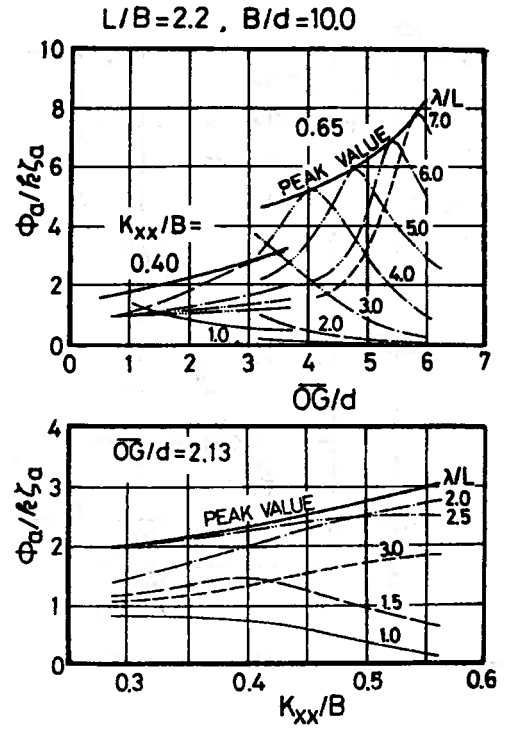
FIG. 21 Effect of the Center of Gravity and Radius of Gyration on the Vessel's Rolling Motion
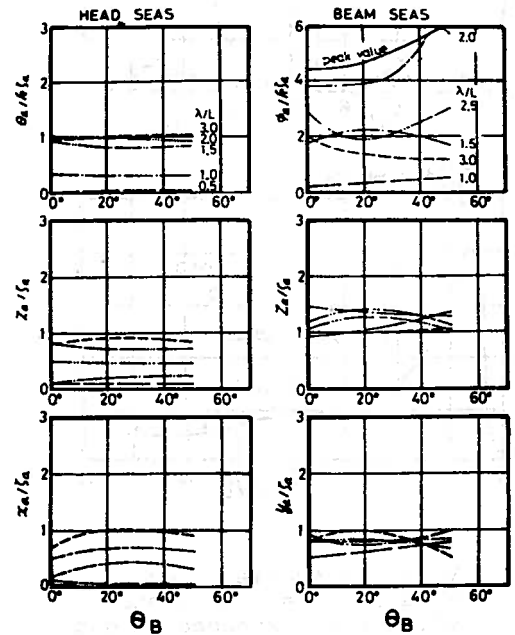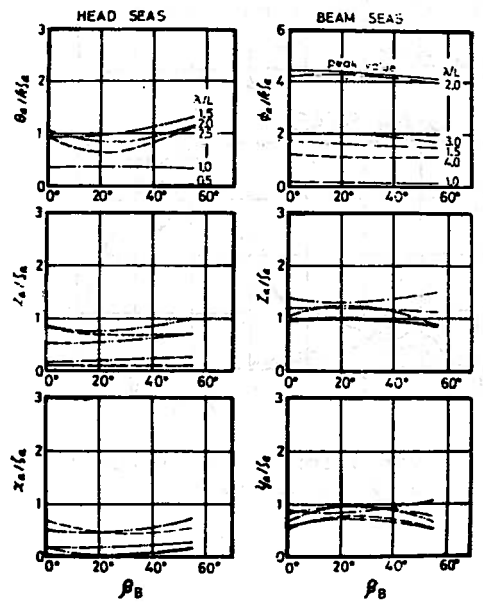

FIG. 22 Effect of Bottom Bevel on the Vessel's Motion

FIG. 23 Effect of Side Bevel on the Vessel's Motion

-17-
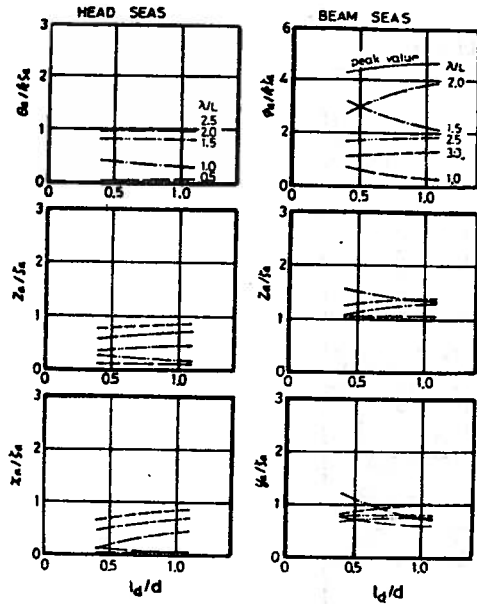
FIG. 24 Effect of Bevel Bottom Cut-
Away on Vessel Motion
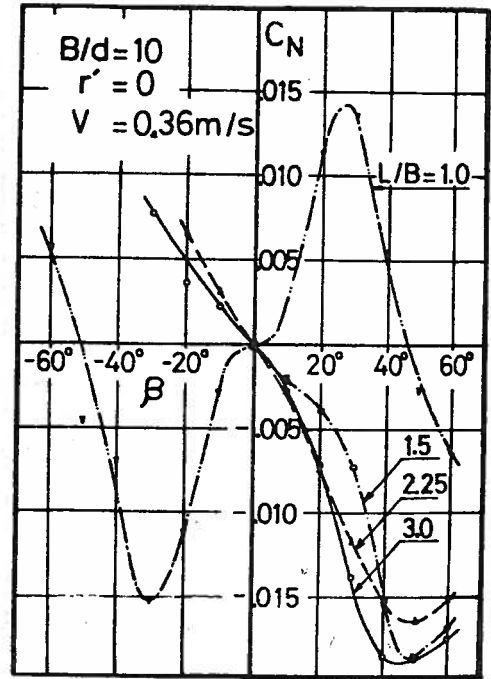
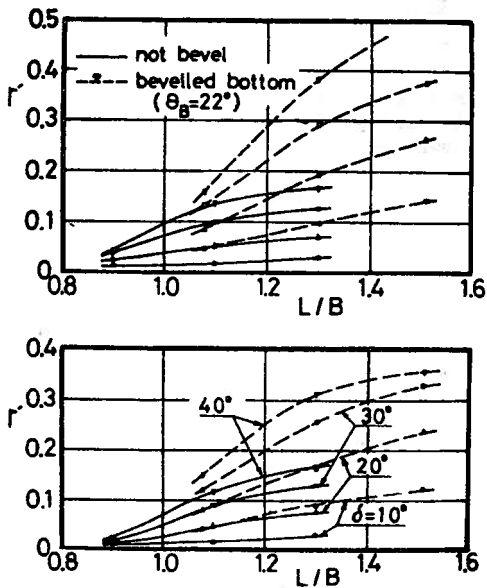FIG. 25 C_N of Rectangular Form Vessels
During Turning

FIG. 26 Influence of L/B Ratio on Advance,
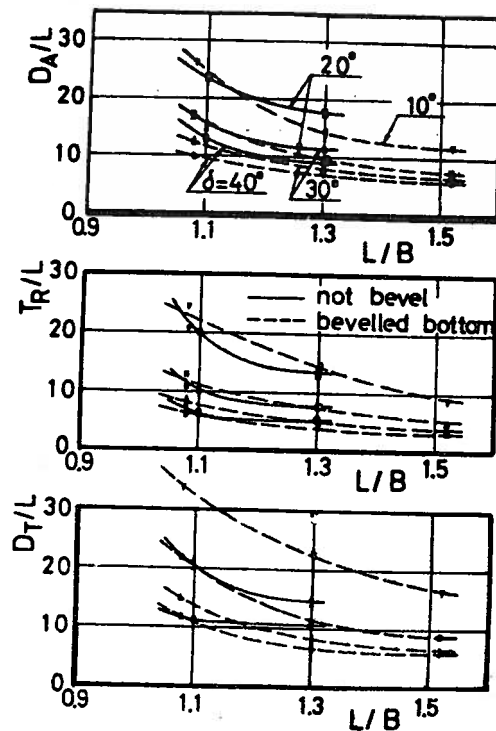Transfer, and Turning Radius

FIG. 27 Influence of Bow Bottom Bevel
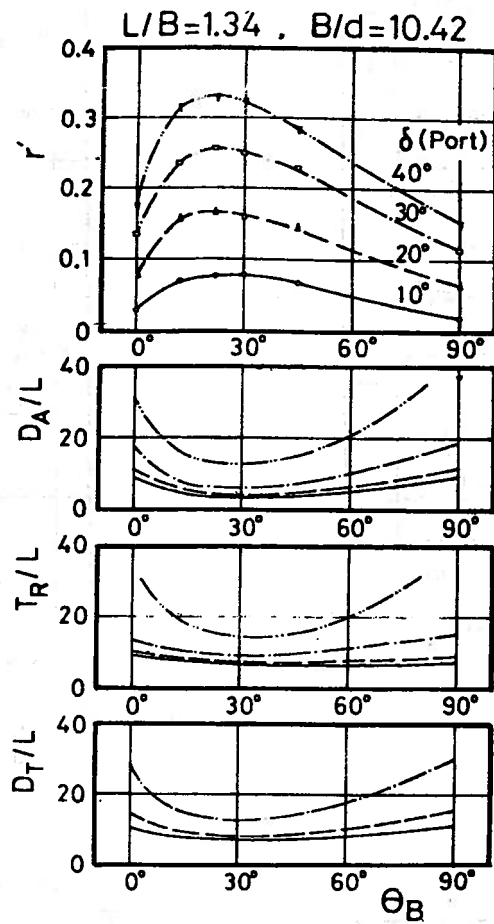Angle on Turning Performance

FIG. 28 Influence of Bow Bottom
Bevel Angle on Vessel's
Turning Performance

SOME STUDIES OF THE COURSE STABILITY
OF TOWED SHIPS SYSTEMS[1]

BY

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ABSTRACT

As we have discussed in this Journal (Nos. 42,43,44 and 46),the course stability for a towed ship system is influenced by the parameters: a) number of towed ships, b) fore- and aft-towing points, c) towing ropes' lengths, e) means of steering, f) course stability criterion of a ship g) radius of gyration, and h) resistance of towed ship. The effects for these parameters are as follows:

a) The larger the number of towed ships, the poorer the course stability, is, and while stable towing of one towed ship is possible by adopting a suitable towing point and the tow rope length, it becomes impossible to tow more than two ships without steering.

b) The effect of aft-towing point is small, and in the case of forward towing point, it is better to place it away from the towed ships' center of gravity.

c) A longer towrope provides more stable towing than shorter towropes.

d) The course stability of a towed ship system increases with the increase of a ship's course stability criterion, and this effect becomes smaller for two and more towed ships.

e) By means of steering, the course stability increases remarkably.

f) The course stability of same sized towed ship is better than the large towed ship in the case of the same parameters of each ship.

g) When the radius of gyration increases, the course stability grows worse, even for one towed ship and when the radius of gyration is greater than 0.50L, it is impossible to have stable towing without steering.

h) As the resistance of the towed ship increases, we obtain stable towing of the towed ship system

When an unstable ship like a barge is towed, its course stability is very poor (described in d)and to obtain stable towing for towed ships such as barges, the course stability of the barge itself should be improved by utilizing skegs. In this paper, towing experiments for a single barge and tugboat and oblique test of the barge for the conditions shown in table 2, have been performed and the results are as follows:

1) When the skeg is filled to the barge, because of the increment of Y' and the decrement of N', course stability of the towed ship system improves very much, so that when the high lift skegs like the slotted flap skeg and the slotted flap with rotor skeg are fitted to the barge, it is not difficult for us to imagine that the course stability of this system improves.

2) For the towing test of towed ships system, due to the effect of towed ship size (see Ref. 1,2,3,4), it is not suitable to test in towing tanks with narrow breadth. In other words, it is necessary that they be towed by a tugboat in a wide surface area performance facility.

TRANSLATION[4]

1. Introduction

The authors have up to now made numerous theoretical studies concerning the towing of towed vessels. The course stabi-

lity of a towed vessel is determined by assuming the towing rope as a weightless string which only transmits the tension force. If the towed vessel can be considered stable with the above assumption, then it not necessary to be concerned with the mass and elasticity of the towrope influencing the course stability. [1],[2],[3],[4].

When unstable vessels such as barges are considered as a towed ship system, their course stability performance is extremely poor. This paper discusses the course stability of a towed barge with skegs fitted to both sides of the stern in order to improve its course stability performance. This is examined by towing experiments as well as oblique towing tests which are presented with a discussion of the towing ship-towed vessel system experiments.

II. FACTORS INFLUENCING THE COURSE STABILITY

In the towing problem, the following factors are influential:

a) number of towed vessels, b) size of towing vessel and towed vessels, c) length of the towing rope, d) location of towing point (forward and aft), e) the quality of each vessel's course stability performance, f) the size of the radius of gyration with respect to the center of gravity of each vessel, g) the increase and decrease of the tension force which can be said to represent the increase and decrease of vessel's resistance, and h) steering.

In (a) when the number of vessels increase the towing becomes unstable [1],[2]. In (b) when the towed vessel becomes larger, the course stability becomes poorer as shown in Fig.2.

In (c) when the towrope becomes longer the course stability improves, but it is not desirable for it to become too long from the viewpoint of turning performance and navigation.

In (d) it is better if the forward towing point extends a distance ahead of the bow. An example of this is the generally used bridle system. Also, it will be best to locate the center of gravity behind the towing point.

In (c) the course stability performance of each vessel can be described by:

$$\Delta = Y_\beta' N_r' - (m_x - Y_r') \cdot N'$$

(where $Y'...Y_r'...$ are called the static and rotary derivatives respectively which can be experimentally as well as theoretically determined) When Δ is large the vessel's course stability performance is good, and when $\Delta < 0$ the vessel becomes course unstable,

Fitting skegs causes a large increase in

the value of Δ. The skeg derivatives are denoted by L_β and L_r. The derivatives of the vessel with skegs are denoted by a bar — . The following relations are obtained when the skegs are fitted in the stern:

$$\overline{Y}_\beta' = Y_\beta' + L_\beta'$$

$$\overline{N}_\beta' \quad N_\beta' - b'L_\beta'$$

$$\overline{Y}_r' = Y_r' + L_r'$$

$$\overline{N}_r' = N_r' + b'L_r'$$

Here b' is the ratio of the distance between the skeg's center of pressure and the vessel's center of gravity to the length of the ship. It is possible to write:

$$L_r' \doteqdot b'L_\beta'$$

Basically the derivatives L_β' and L_r' are associated with the lift L generated by the skegs which causes a corresponding increase in Δ and an improvement in the course stability performance. Consequently skegs which cause increases in L_β' or high lift may be preferable. With a larger value of Δ, it becomes possible to use shorter towrope lengths and to locate the towing point closer to the vessel's center of gravity.

In (f) when the radius of gyration becomes larger, the vessel becomes course unstable. An example is shown in Figs. 3 and 4.

In (g) the increase of resistance causes an improvement of course stability. Consequently, not only from this point of view, but also from a large improvement in Δ (discussed in (e)) it is recommended to fit skegs.

In (h) when the vessel is steered in proportion to its course, its course stability is extremely good, and there is a remarkable improvement in the course stability of the towing boat itself.

III. EQUATION OF MOTION FOR N NUMBER OF TOWED BARGES FITTED WITH BRIDLES

Fig.1 shows the i-1 to i+1 towed vessels of a towed barge system. Here we consider the equilibrium conditions for forces by making use of the notation used in previous four articles [1],[2],[3],[4].

From Fig.1 it can be seen that:

$$\cos\sigma_{1i} = B_i/2\tilde{\ell}_i, \quad \sin\sigma_{1i} = f_{2i}/\tilde{\ell}_i$$

In the case when $\varepsilon_i < \pi/2 - \sigma_{1i}$ from equilibrium of forces:

$$T_{1i}' \cos\left(\frac{\pi}{2} - \sigma_{1i}\right) + T_{2i}' \cos\left(\frac{\pi}{2} - \sigma_{2i}\right)$$
$$= T_i' \cos \epsilon_i$$

(2)

$$T_{1i}' \sin\left(\frac{\pi}{2} - \sigma_{1i}\right) +$$
$$+ T_{2i}' \sin\left(\frac{\pi}{2} - \sigma_{2i}\right) = T_i' \sin \epsilon_i$$

Now assuming that ϵ_i is very small it is possible to make the approximation:

$$\cos \epsilon_i \doteq 1, \quad \sin \epsilon_i \doteq \epsilon_i \text{ and } \sigma_{1i} \doteq \sigma_{2i}$$

Then from (2) the following expression is written:

$$T_{1i}' + T_{2i}' \doteq T_i' \frac{\tilde{\ell}_i}{f_{2i}}$$

(2)'

$$T_{1i}' - T_{2i}' \doteq T_i' \frac{2\tilde{\ell}_i}{B_i} \epsilon_i$$

Consequently the external force and moment, which are exerted by the towrope segment $\tilde{\ell}_i$ of the vessel with Bridle, are expressed as follows:

The external force in the y_i direction:

$$T_{1i}' \cos \sigma_{1i}\ T_{2i}' \cos \sigma_{2i} \doteq (T_{1i}' - T_{2i}') \frac{B_i}{2\tilde{\ell}_i}$$

$$= \left(T_i' \cdot \frac{2\tilde{\ell}_i}{B_i} \epsilon_i\right) \cdot \frac{B_i}{2\tilde{\ell}_i} = T_i' \epsilon_i$$

Moment around the z_i axis:

$$\frac{T_{1i}' B_i}{2L_i \tilde{\ell}_i} \cdot (f_{1i} + f_{2i}) - \frac{T_{2i}'}{2L_i \tilde{\ell}_i} \cdot (f_{1i} + f_{1i})$$

(3)

$$= (T_{1i}' - T_{2i}') \cdot \frac{B_i}{2L_i \tilde{\ell}_i} \cdot (f_{1i} + f_{2i})$$

$$= T_i' \cdot \frac{(f_{1i} + f_{2i})}{L_i} \cdot \epsilon_i \geq \frac{T_i' f_i}{L_i} \epsilon_i$$

where it is assumed that $(f_{1i} + f_{2i}) \geq f_i$

Next the case when $\epsilon_i \geq \frac{\pi}{2} - \sigma_{1i}$

will be considered. From the characteristics of the towrope T_{2i}' must be equal to zero and consideration is again given to the equilibrium of forces:

Forces in y_i direction:

$$T_{1i}' \cos\left(\frac{\pi}{2} - \epsilon_i\right) = T_{1i}' \sin \epsilon_i = T_{1i}' \epsilon_i$$

Moment around the z_i axis: (4)

$$T_{1i}' \cdot \frac{f_i}{L_i} \cos\left(\frac{\pi}{2} - \epsilon_i\right) + \frac{T_{1i}' B_i}{2L_i} \cdot \frac{f_{2i}}{\tilde{\ell}_i}$$

$$= \frac{T_{1i}' f_{1i}}{L_i} \epsilon_i + \frac{T_{1i}' f_{2i}}{L_i} \epsilon_i$$

$$= T_{1i}' \frac{(f_{1i} + f_{2i})}{L_i} \epsilon_i \geq \frac{T_i' f_i}{L_i} \epsilon_i$$

From comparing (3) and (4) it is clear that both relations are exactly the same. On the other hand from geometric conditions:

$$X_i = X_{i-1} - a_{i-1} - \ell_i - (f_{1i} + f_{2i})$$

(5)

$$Y_i = Y_{i-1} - a_{i-1}\theta_{i-1} - \ell_i \epsilon_i - (f_{1i} + f_{2i} + \ell_i)\theta_i$$

are obtained. Differentiating (5) with respect to time t and assuming that

$$f_i \doteq f_{1i} + f_{2i}$$

results in:

$$\dot{X}_i = \dot{X}_0 - \sum_{k=1}^{i} (\dot{a}_{k-1} + \dot{f}_k + \dot{\ell}_{kk}) = \dot{X}_0$$

$$\dot{Y}_i = \dot{Y}_0 - \sum_{k=1}^{i} (a_{k-1}\dot{\theta}_{k-1} + f_k\dot{\theta}_k + \ell_k \cdot$$
$$\cdot (\dot{\epsilon}_k + \dot{\theta}_k))$$

(6)

$$= \dot{Y}_0 - \sum_{k=1}^{i} (a_{k-1}\dot{\theta}_{k-1} + (f_k + \ell_k)\dot{\theta}_k + \ell_k \dot{\epsilon}_k)$$

here X_0, Y_0 are the coordinates of the towing vessel, $T_i', T_{1i}' \ldots$ are the values obtained by dividing T_i, T_{1i} by $1/2\rho A_i U_i^2$ where ρ is fluid density, $A_i = d_i L_i$

(L_i vessel length, d_i vessel draft),

U_i vessel forward velocity.

Taking the forward velocity of the towing vessel as U_0 results in:

$$\dot{X}_0 = U_0 \cos(\theta_0 + B_0) = U_0$$

$$\dot{X}_1 = U_1 \cos(\theta_1 + B_1) = U_1$$

(7)

$$\dot{Y}_o = U_o \sin(\theta_o + B_o) = U_o(\theta_o + B_o)$$

$$\dot{Y}_i = U_i \sin(\theta_i + B_i) = U_i(\theta_i + B_i)$$

From (6) and (7):

$$\dot{X}_o = \dot{X}_i = U_o$$

$$\beta_i = \theta_o + \beta_o - \theta_i -$$

$$- \frac{1}{U_o} \cdot \sum_{k=1}^{i} (a_{k-1}\dot{\theta}_{k-1} + (f_k + \ell_k)\dot{\theta}_k + \ell_k \dot{\varepsilon}_k) \tag{8}$$

Substituting (8) into equations (1) and (2) of [1], after rearranging results in the following equations of motion for the ith towed barge:

$$m_{yi}\left(\frac{L_i}{U_i}\right)^2 \cdot \frac{f_i + \ell_i}{L_i} \ddot{\theta}_i$$

$$+ \frac{1}{U_i}\left[(m_{yi} - m_i + (k_{i2} + \tfrac{1}{2})Y_{\delta i}')L_i + (f_i + \ell_i) \cdot (Y_{\beta i}' + Y_{\delta i}') \right] \dot{\theta}_i$$

$$+ \left[Y_{\beta i}' + (k_{i1} + 1)Y_{\delta i}' + T_{i+1}' \right] \theta_i + m_{yi}\frac{L_i}{U_i^2} \cdot \sum_{k=1}^{i}((a_{k-1} + f_k + \ell_k)\ddot{\theta}_{k-1})$$

$$+ \frac{1}{U_i} \cdot (Y_{\beta i}' + Y_{\delta i}') \cdot \sum_{k=1}^{i}((a_{k-1} + f_k + \ell_k)\dot{\theta}_{k-1})$$

$$- m_{yi}\frac{L_i}{U_i}\dot{\theta}_o - (Y_{\beta i}' + Y_{\delta i}')\theta_o - T_{i+1}'\varepsilon_{i+1}$$

$$- m_{yi}\frac{L_i}{U_o}\dot{B}_o$$

$$- (Y_{\beta i}' + Y_{\delta i}')\beta_o + m_{yi}\frac{L_i \ell_i}{U_i^2}\ddot{\varepsilon}_i + \frac{\ell_i}{U_i} \cdot (Y_{\beta i}' + Y_{\delta i}')\dot{\varepsilon}_i$$

$$+ T_i'\varepsilon_i - T_{i+1}'\varepsilon_{i+1} + m_{yi}\frac{L_i}{U_i^2} \cdot \sum_{k=1}^{i}(\ell_{k-1}\ddot{\varepsilon}_{k-1})$$

$$+ \frac{1}{U_o}(Y_{\beta i}' + Y_{\delta i}') \cdot \sum_{k=1}^{i}(\ell_{k-1}\dot{\varepsilon}_{k-1}) = 0$$

$$n_i\left(\frac{L_i}{U_i}\right)^2 \theta_i$$

$$+ \frac{1}{U_o} \cdot \left[(N_{ri}' + (k_{i2} + \tfrac{1}{2})N_{\delta i}')L_i - (f_i + \ell_i) \cdot N_{\beta i}' - N_{\bar{\delta}i}') \right]\dot{\theta}_i$$

$$+ \left[((k_{i1} + 1)N_{\delta i}' - N_{\beta i}') + \frac{a_i}{L_i} \cdot T_i' \right]\theta_i$$

$$- \frac{1}{U_i}(N_{\beta i}' N_{\delta i}') \cdot \sum_{k=1}^{i}\left[(a_{k-1} + f_{k-1} + \ell_{k-1})\dot{\theta}_{k-1} \right]$$

$$- T_i'\frac{f_i}{L_i}\varepsilon_i - T_{i+1}'\frac{a_i}{L_i}\varepsilon_{i+1} - \frac{1}{U_i}(N_{\beta i}' + N_{\delta i}')\dot{\varepsilon}_i$$

$$- T_{i+1}'\frac{a_i}{L_i}\theta_{i+1} + (N_{\beta i}' - N_{\delta i}')\theta_o + (N_{\beta i}' N_{\delta i}')\beta_o$$

$$- \frac{1}{U_i}(N_{\beta i}' - N_{\delta i}') \cdot \sum_{k=1}^{i}(\ell_{k-1}\dot{\varepsilon}_{k-1}) = 0 \tag{10}$$

These relations are exactly the same as equations (5) and (6) of [1] when f_i is taken as the length to the tip of the bridle and ℓ_i is taken as the length of towrope from the tip of the birdle.

IV INVESTIGATION OF COURSE STABILITY PERFORMANCE OF TOWED VESSELS BY TOWING VESSELS BY TOWING EXPERIMENTS

The quality of the towed vessel course stability performance changes with the size of the towing vessel and towed vessel as shown in Fig.2. However, it does not follow that the course stability of the towing ship-towed vessel system is equal to the towed vessel course stability performance when towed by the towing tank carriage. Therefore, a towing ship - towed vessel system should be used when investigating the course stability performance. (Here the towing ship-towed vessel system designates the towing vessel and all towed vessels.)

To investigate the course stability of the system a study was conducted by changing the towrope length and examining the path of the vessels.

Naturally, it is preferable to calculate the course stability performance using the towed vessel(s) and towing vessel's derivatives. However since the course stability performance is qualitatively shown in Figs.2, 3, and 4, experiments were conducted to determine the minimum tow rope length namely the limit of the course stability of the system. The models used are described

in Table 1. The skegs were fitted in
pairs.   Two types of skegs were used:
the Mariner skeg with a symmetric air-
foil section and the Deformed skeg with
a chambered air-foil section which are
shown in Fig. 5.

Table 5

	Tugboat	Barge
L_{pp}	0.800m	2.500m
B	0.228m	0.600m
d	0.027m	0.143m
Displ.	0.436kg	188.26kg
C_B	0.651	0.875

U = 0.367m/s, F_n=0.07423

In the experiments radio controlled
steering was used. The towrope length
was changed and the model ship's course
stability performance was investigated
by photographing the towing path while
the model moved forward. These results
are shown in Figs. 7, 8, and 9 while a
comparison for q=ℓ/L=2 is presented in
Fig. 10. The results are:

1) Without the skegs the course stability
performance is poor, so that whatever ℓ is
used, straight line towing is not possible.

2) With the Mariner skeg when ℓ>6L,
straight line towing is possible, while
with a shorter towrope length it is im-
possible.

3) With the Deformed skeg there is ade-
quate stability for straight line towing
when ℓ>L_1.

Consequently although the deformed skeg
has a slightly higher value of resistance
than the mariner skeg in forward motion,
it enables course stable performance
to be achieved with a short towrope.

V EXPERIMENTAL STUDY FOR IMPROVING THE
   COURSE STABILITY PERFORMANCE OF A
   SINGLE TOWED VESSEL.

One way to improve the course sta-
bility performance of the towing ship-
towed vessel system is to improve the
course stability performance of the
individual vessels. Consequently the
various improvements from the mariner,
deformed, slotted flap and slotted flap
with rotor skegs were investigated, Figs.
5, 6, and 7, to determine the best skeg
type.

(1) Investigation Using Free Running
     Models

It is known that when a model which
was towed from a towing tank carriage
at constant speed is used in free running

tests there is a reduction in the model's
speed, and it has a slightly poorer course
stability performance. To simply decide
the relative characteristics of skegs
free running model tests were used.

Figs. 11 to 14 present the paths of
the free running models when the skegs are
fitted to the barge. From Fig.10, it is
clear that without skegs, a straight line
path does not occur and the stability per-
formance is bad. It is also evident that the
course stability performance improves when
the mariner skegs or deformed skegs are
fitted. The course stability performance
with the deformed skegs is better than
when the mariner skegs are fitted.

Fig.11 shows the results from using
the slotted flap skeg. From this figure
when η increases the stability performance
is improved. Thus when η>20° a straight
running is possible. However, when η>70°
stalling occurs, and the stability perfor-
mance becomes poor. The model paths of
the case when the slotted flap skegs with
rotors are fitted are shown in Fig.12.
The turning rotor causes a high lift force.
In addition, when η is increased, the
stability becomes better so that even
when η>70° good results are obtained.
A comparison of slotted flaps skegs with
and without rotor action is shown in Fig.
18. When η is small the turning rotor
does not cause a large difference, but
when η>70° its effect is clear. Therefore,
it is understood that the turning rotor
causes a large lift force and the course
stability performance improves. From
these experiments, it is therefore antici-
pated the stability performance is improved
by skegs in the following order:

Mariner skeg   Slotted flap skeg
Deformed skeg   Slotted flap skeg with
                      rotors [best]

(2) Oblique Towing Experiments

To detemine the hydrodynamic forces
in each case, the improvement in the course
stability performance of the barge model
was investigated by comparing the effects
of the various kinds of skegs on the hull.

This was done by oblique towing ex-
periments. These tests were made using the
carriage of the University of Kyushu's
Faculty of Engineering Naval Architecture's
ship motion basin. The skegs used in these
tests are shown in Figs.5 and 6. For the
case of the slotted flap skegs with rotors
experiments were made by changing the
revolutions ω of the rotor as shown in
Table 2.

ω	η						
	0°	5°	10°	20°	30°	50°	70°
0	o	o	o	o	o	o	o
800	o	o	o	o	o	o	o
2000	o	o	o	o	o	o	o

Table 2

The skegs shown in Fig.5 were fitted to both sides of the bottom of the barge stern. They form the three-component force measurements at midships, the hydrodynamic forces in the x and y axis and the moment around the z axis were measured to obtain Y_β' and N_β'

	Without skeg	Mariner skeg	Deformed skeg
X'	0.04316	0.04711	0.05409
Y_β'	0.3124	0.3210	0.3672
N_β'	0.1428	0.0937	0.0607

Table 3

The experimental results are shown in Figs.15 to 25 and the corresponding Y_β' and N_β' values are shown respectively in Figs. 24 and 25. Table 3 summarizes the values of X', Y_β' and N_β' with no skegs, mariner skegs and deformed skegs. When the mariner skeg and deformed skeg are fitted to the barge, there is not a very large difference in the values of Y_β' in comparison with the large differences in the values of N_β'. When the deformed skegs are used, it clear there is a remarkable reduction in the N_β' value and a large contribution to Δ. In addition, when the slotted flap skeg is fitted there is a remarkable increase in the values of Y_β' with increasing η. It is cle... the value of N_β' is negative when η is ne. 0°, as shown in Fig.25, and stall occurs w - η>10°. As Fig.5 indicates as the roto turns energy is supplied preventing stall so a high lift force results. Although there is an increase in Y_β' when both ω and η become larger in Fig.24, there is little noticeable difference in the value of N_β' in Fig.25. Finally it is anticipated that the course stability performance is increased by skegs in the following order:

Mariner Skeg,   Slotted flap skeg
Deformed skeg   Slotted flap skeg
                        with re...

f addit ...

only a small effec .

IV CONCLUSION

From the previous results the following conclusions were obtained.

1) The major effects of the skegs on the course stability performance are to increase L_β and to decrease N_β , as well as causing a slight increase in resistance. Consequently, it is conceivable that the stability performance becomes good when a high lift skeg such as the slotted flap skeg is used.

2) In straight line towing in the towing tank where the towed vessels are towed from the carriage, the effective towing vessel becomes extremely big and the towed vessel extremely small. As Figs. 2 and 3 indicate this represents a situation different from the actual one. Therefore to investigate the course stability performance of the towing ship-towed vessel system, it is necessary to conduct the experiments using a towed vessel system in an installation with a large unrestricted water surface.

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TRANSLATOR'S NOTE: The Interested reader can obtain a summary of [1],[2],[3],[4] in:

Inoue, S., and Lim, S., "The Manoeuverability of Towed Ships System," PROCEEDINGS 14th International Towing Tank Conference, Vol. 2, Ottawa, Canada, 1975. pp. 571-580.

FIG. 1

- DETAIL -

ONE TOWED SHIP WITH TUGBOAT

$\Delta_o = 0.00671$, $r = 0$, $\mathcal{E} = L_1/L_o$, $K = 0.24 L$

$\Delta_1 = 0.00671$
$\Delta_1 = -0.00758$
$\Delta_1 = 0.06494$

FIG. 2

ONE TOWED SHIP WITH TUGBOAT

$\Delta_o = 0.00671$, $r = 0$, $\mathcal{E} = L_1/L_o$, $K = 0.35 L$
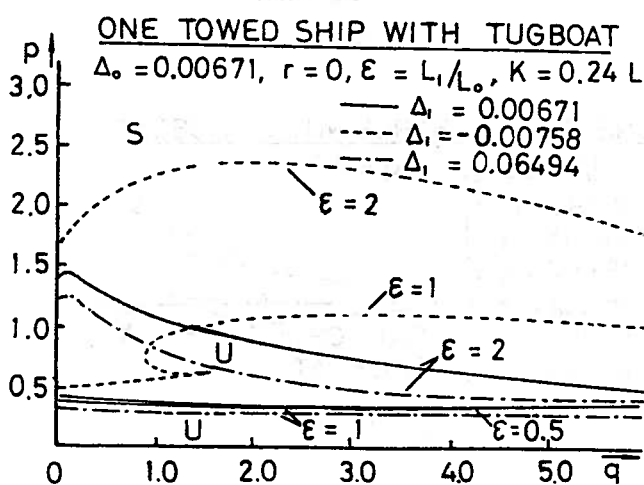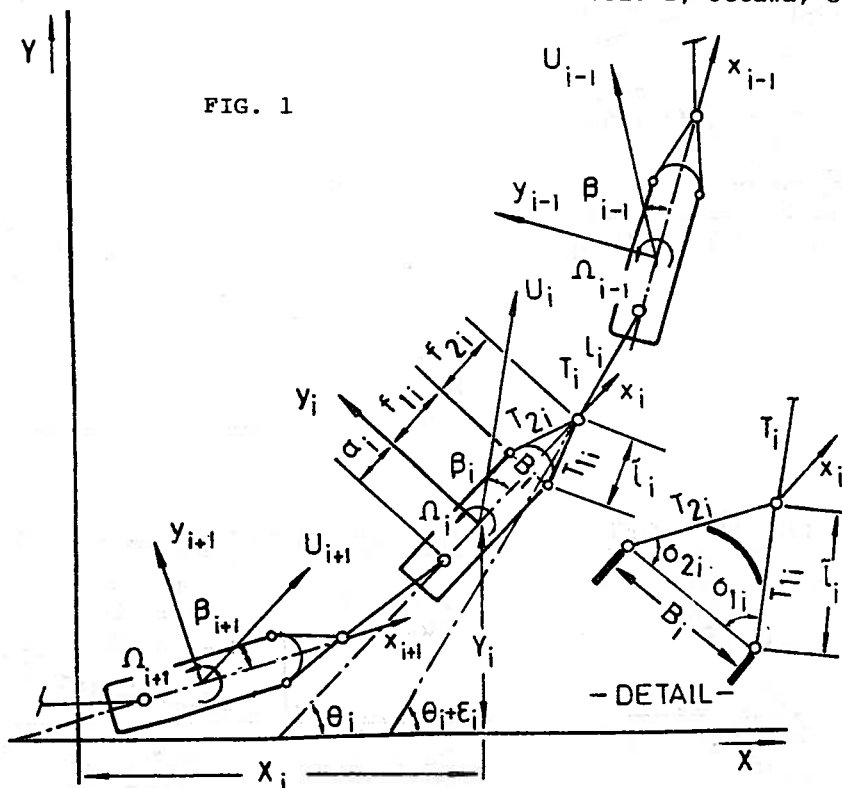
$\Delta_1 = 0.00671$
$\Delta_1 = -0.00758$
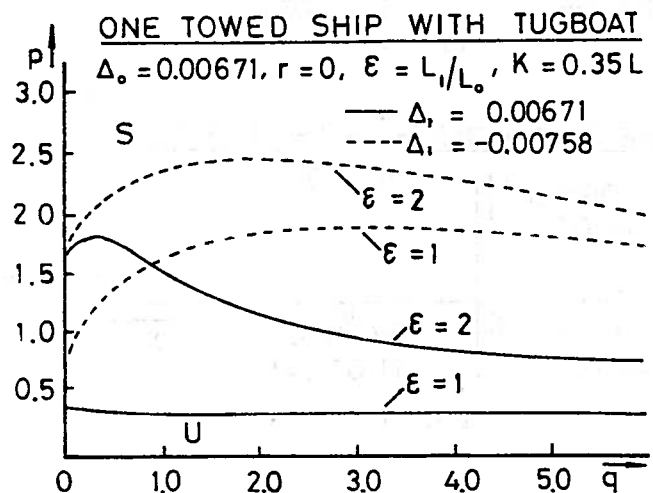
FIG. 3

Notation: $q = \ell/L$ where: L : Towed Vessel Length.
           $p = f/L$ ℓ : Length of towing rope between vessels.
                                      f : Distance from center of gravity to fore towing pt.

ONE TOWED SHIP WITH TUGBOAT

$\Delta_o = 0.00671, r = 0, K = $ Radius of Gyration

$\qquad$ $\Delta_1 = 0.00671$
----- $\Delta_1 = 0.00343$
-·-·- $\Delta_1 = -0.00758$

K = 0.50L

K = 0.35L

K = 0.24L

K = 0.24L - 0.35L
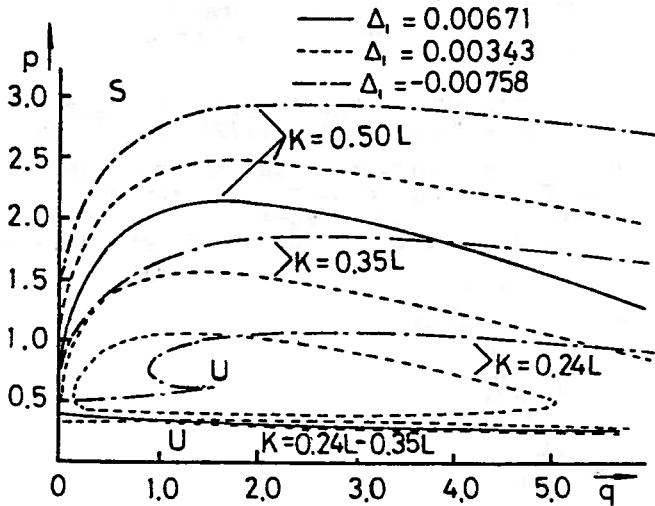
FIG. 4

(Notation as in Figs. 2 and 3)

MARINER SKEG

DEFORMED SKEG

SLOTTED FLAP SKEG WITH ROTOR
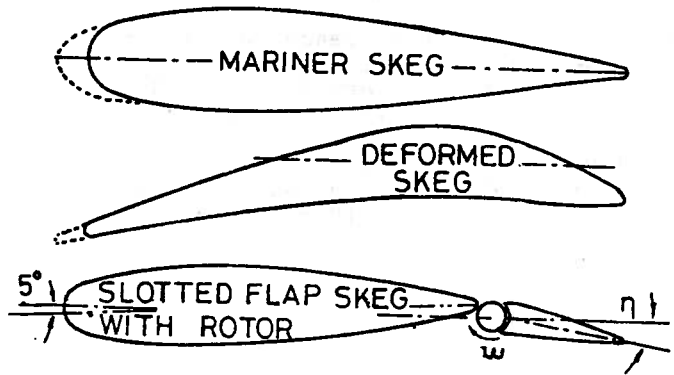
FIG. 5

FIG. 6

-ONE TOWED BARGE WITH TUGBOAT

WITHOUT SKEGS

$\qquad$ q = 2.0
----- q = 4.0
-·-·- q = 6.0

PATH OF BARGE
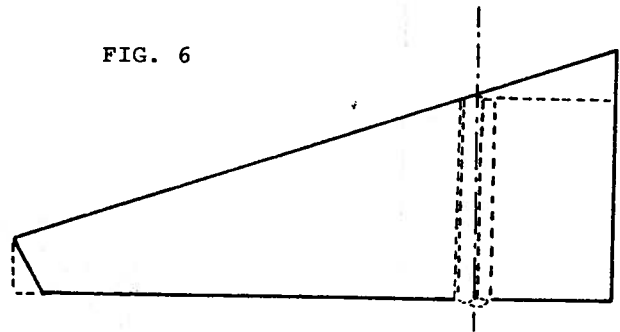
PATH OF TUGBOAT

FIG. 7

ONE TOWED BARGE WITH TUGBOAT

MARINER SKEGS

$\qquad$ q = 2.0
----- q = 4.0
-·-·- q = 6.0

PATH OF BARGE
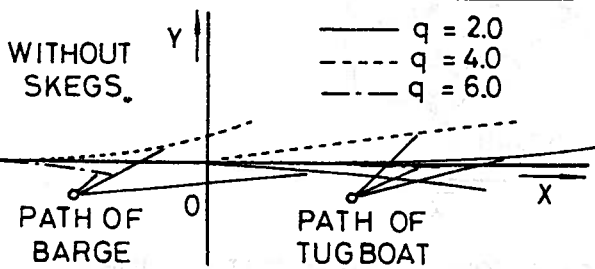
PATH OF TUGBOAT

FIG. 8

ONE TOWED BARGE WITH TUGBOAT

DEFORMED SKEGS

q > 1.0

PATH OF BARGE

PATH OF TUGBOAT

FIG. 9

ONE TOWED BARGE WITH TUGBOAT

SKEGS
$\qquad$ WITHOUT
----- MARINER
-·-·- DEFORMED

q = 2.0

PATH OF BARGE

PATH OF TUGBOAT

FIG. 10

FIG. 11

WITHOUT SKEG
MARINER SKEGS
DEFORMED SKEGS

FIG. 12

$\eta=70°$

50° 30°

10° 5° 0°

SLOTTED FLAP SKEGS

FIG. 13

$\eta=70°$ 10° 10° 0° 0°

$\omega=0$ r.p.m.
800

SLOTTED FLAP SKEGS
WITH ROTORS

FIG. 14

$\eta=15°-70°$ 10° 5° 0°

SLOTTED FLAP SKEGS
WITH ROTORS

$\omega=800$ r.p.m.

SLOTTED FLAP SKEGS
WITH ROTORS

X'

0.15

0.10

0.05

$\omega =$ 0 rpm
$\omega =$ 800 rpm
$\omega =$ 2000 rpm

0   20°   40°   60°   80° η

FIG 15

Y'

0.30

0.20

0.10

SKEGS
WITHOUT
MARINER
DEFORMED
SLOTTED FLAP
($\eta =0°$)

0   5°   10°   15°   20° β

FIG. 16

N'

SKEGS
WITHOUT   MARINER   DEFORMED

0.015

0.010

0.005

SLOTTED FLAP

0   5°   10°   15°   20° β

FIG. 17

-28-

FIG. 18

FIG. 19

FIG. 20

FIG 21

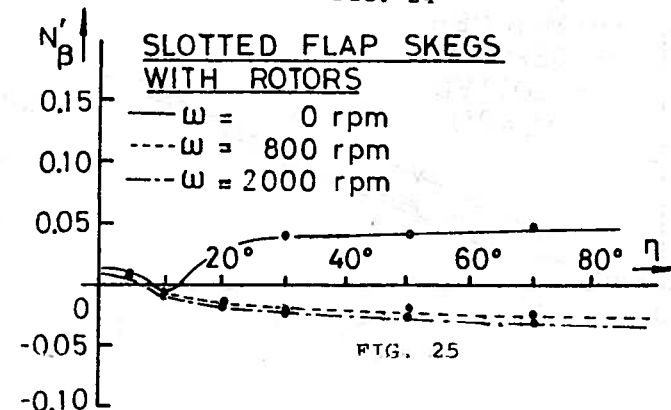

FIG. 22

FIG. 23

FIG. 24

FIG. 25

ON THE COURSE STABILITY OF A BARGE[1]

BY

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ABSTRACT

A barge has been used for elimination or reduction of labor in loading and unloading work, or for specialized purposes. Recently, their size is growing more and more, and they have a great variety. Because a barge generally is course unstable, various ways for stabilization have been considered. One of the well known ways is the skeg fitted on aft of the barge.

In this paper, the skeg effect on the course stability on a barge is investigated experimentally by measuring the lateral force and moment acting on hull and skeg. The types of skeg selected are the following:

(1) Mariner Type Skeg, (2) Deformed Type Skeg
(3) Slotted Flap Type Skeg.

From these results, we may conclude that the lateral force acting on a barge increases with greater distances between skegs but the moment decreases, and the difference due to skeg type is very remarkable. We see that a barge with skeg becomes more stable in order of Marine Type Skeg, Deformed Type Skeg and Slotted Flap Type Skeg. However, there is a remarkable difference in static derivatives in oblique towing tests, especially in the value of N_β'.

It is not necessary to consider the interference effect between skeg and ship's body on the hydrodynamic derivatives in the case of large separation distance between skegs.

The most desirable method for the course stabilization of the barge will be to select the Slotted Flap Type Skeg. However, it should be noted that there is a remarkable increase in the resistance of barge due to skegs. Therefore, when applying these results to the actual vessel we should have to consider the most suitable method for the course stabilization of a barge in connection with the lateral force and moment and the resistance increment of the barge to skegs.

TRANSLATION[3]

1. INTRODUCTION

From ancient times barge vessels have been considered as an economical and rational means of ocean borne transportation. Recently, in addition to transport, special barges are being built. The barge vessels used in transport work are of two major types; one is the towed barge and the other is the pushed barge. The towed barge hulls are basically simple hull forms and their course stability performance is poor. The barges are course unstable due to the hull form design which causes safety and navigation problems when towing.

The recently constructed large barge vessels typically have a spoon bow, a stern with the dead wood cut away, and they are basically very course unstable. Among the various methods considered for improving the course stability performance, the main one is to install skegs to the dead wood cutaway section of the stern.

While many types of skeg designs and types have been used on full scale vessels, there have been hardly any formal publications in research journals regarding the performance of the skegs.

The authors [1] previously conducted an investigation of the course stability performance of a barge vessel in a towed vessel system. In this present paper a discussion of the results of an experimental investigation of the influence of the skegs, their form and location on the course stability performance, and interaction of the hull and skegs is presented for a towed barge system.

2. FORM OF MODEL HULL AND SKEGS

The model used in the experiments is a barge hull whose main particulars and hull form are shown in Table 1 and Fig.1. The three types of skegs used in this report are shown in Table 2 and Fig.2. Each of these three skegs has a vertical projected area which is nearly the same profile as the deadwood cutaway from the barge. From their horizontal cross-section the skegs can be distinguished as: the Mariner type skeg (M.T.S.) with a symmetrical airfoil section, the Deformed type skeg (D.T.S.) with a cambered airfoil section and the slotted flap type skeg (S.F.T.S.). The slotted flap type skeg has the same amount of camber as the D.T.S. and has two slots at the two flap locations. The skegs are set in pairs symmetrically around the vessel's centerline near the vessel's sides.

L_{pp}	:	2.5m
B	:	0.60m
d	:	0.1417m
C_b	:	0.875
Δ	:	185.98kg

Table 1 Particulars of Barge Model

Length	:	0.25m
Area	:	$0.01635m^2$
Area Ratio	:	1/22
Aspect Ratio	:	0.524

Table 2 Particulars of Skegs

3. EXPERIMENTAL METHOD AND MEASUREMENT RESULTS

In determining the course stability performance of a barge with skegs, it is necessary to know the hydrodynamic forces acting on the vessel and its hull motion. Therefore, the hydrodynamic forces were measured in oblique towing tests and rotating arm turning tests of captive models.

In order to investigate the influence of the skeg forms and spacing, the hydrodynamic forces were measured for the following four cases in the experiments:

1. Hydrodynamic forces acting on the barge hull without skegs.

2. Hydrodynamic forces acting on the barge hull with skegs.

3. Hydrodynamic forces acting on the skeg fitted to the barge aft (stern).

4. Hydrodynamic forces acting on the skeg alone.

The model velocity was U=0.3 m/s which is scaled from the actual barge being towed at 5 knots. In the tests the coordinate system in Fig. 3 was adopted in order to measure the hydrodynamic forces as well as the skeg location.

The spacing between the skeg locations b_s is expressed in a non-dimensional parameter $\eta = bs/B$ using the vessels beam (B). In the tests four values of η were used: 0.33, 0.50, 0.67, 0.75. The maximum value of η was 0.75 which was determined from the hull form and the arrangement of the skegs on the model.

3.1 The Hydrodynamic Forces Acting on the Barge Hull With and Without the Skegs

The measurement results of the hull side force (Y) and the moment around midships (N) for the drift angle B are shown in non-dimensional form in Figs.4-10, where:

$$Y' = Y/1/2\rho LdU^2$$

$$N' = N/1/2\rho L^2 dU^2$$

It is clear that there is a remarkably large difference in the side force and moment for the vessel with and without the skegs. The hydrodynamic derivatives obtained from this data are shown in Figs. 11 and 12 and the center of pressure for these cases is shown in Fig. 13.

3.2 Hydrodynamic Forces Acting on the Skeg Behind the Hull.

The hydrodynamic forces acting on the skeg fitted behind the barge vessel were measured in the same manner as previously described. From these results the hydrodynamic derivatives of the side force and moment denoted by $_sY_\beta'$ and $_sN_\beta'$ were obtained and shown in Figs.14 and 15.

As for the value of $_sY_\beta'$, when the skeg spacing η changes there is a noticeable change with all three types of skegs.

3.3 Hydrodynamic Forces acting on the Skeg in Open Water

The static derivative of the side force and moment of a pair skeg denoted by $_oY_\beta'$ and $_oN_\beta'$ are shown in Figs. 16 and 17.

As for the value of $_oY_\beta'$ there are only small differences between the three types of skegs, M.T.S., D.T.S., and S.F.T.S. when the spacing of the skegs is small. However, when η becomes large the S.F.T.S., exhibits a large value. While there are small effects on $_oN_\beta'$ from varying η, there are remarkable differences between the S.F.T.S. values and those of the other skegs.

4. INFLUENCE OF SKEG SPACING

The previously described relationship of the side force and moment derivatives of the hull and skeg spacing are summarized in Table 3. From these results the value of Y_β' becomes larger in the following order: M.T.S., D.T.S. and S.F.T.S. while the value of N_β' becomes correspondingly smaller in that order. In addition the corresponding center of pressure moves back. This is especially evident in the case of S.F.T.S. where the center of pressure moves back from midships as the skeg spacing becomes wider. This represents improved stability from the viewpoint of the vessel's course stability performance. Consequently it is good to use skegs whose spacing is as wide as possible.

5. MUTUAL INTERACTION OF SKEG AND HULL

First, in order to study the effect of the hull on the skeg, comparisons are made between the ratio of the hydrodynamic derivatives $_sY_\beta'$ and $_sN_\beta'$ for the skegs behind the hull and the hydrodynamic derivatives $_oY_\beta'$ and $_oN_\beta'$ for the skegs in open water. These are shown in Figs. 18 and 19.

The influences of the hull on the MTS and DTS are evident irrespective of the value of η, and the ratio of the side force derivatives becomes

$$_sY_\beta' \ / \ _oY_\beta' \cong 1/2$$

while the hull effects on the SFTS skeg are not apparent except at large values of η. As for the moment around the skeg axis, the influence of the hull on each of the three types of skegs is present and it depends mainly on the value of η. It seems that this tendency is mainly caused by differences in flow around the stern

which are related to the deadwood cutaway from the stern.

Table 4 - Skeg Effect $(_{Bs}Y_\beta'/_BY_\beta'$ in %)

Type of Skeg	η: 0.75	0.67	0.50	0.33
Mariner	12.1	18.2	19.7	19.7
Deformed	16.4	22.8	18.0	16.5
Slotted Flap	30.0	33.6	39.8	35.3

Type of Skeg	η: 0.75	0.67	0.50	0.33
Mariner	0.95	1.04	0.89	1.48
Deformed	1.34	1.19	0.89	1.27
Slotted Flap	0.83	0.56	0.56	0.83

Table 5: Skeg Effect $(_{Bs}N_\beta'/_BN_\beta'$ in %)

Then in order to study the amount the skeg contributes to the barge hull, the hydrodynamic derivatives of the hull without the skegs $_BY_\beta'$ and $_BN_\beta'$ and the hydrodynamic derivatives of the hull with the skegs $_{Bs}Y_\beta'$ and $_{Bs}N_\beta'$ were obtained, and the values of $_{Bs}Y_\beta'/_BY_\beta'$ and $_{Bs}N_\beta'/_BN_\beta'$ are given in Tables 4 and 5. Here the S.F.T.S. has the largest contribution of 30% at $\eta = 0.7$. In addition, irrespective of the skeg type the effects are large when the values of η are small. It can be said that when η is small there is an effective interaction of the skegs with the stern flow pattern. While for the moment, irrespective of the skeg type, M.T.S. and D.T.S. or S.F.T.S. and value of η the contributions are small. Thus, it is possible to express the hydrodynamic derivatives of the hull motion with skegs by introducing the interaction coefficients of the hull and skegs as follows:

$$Y_\beta' = _BY_\beta' + C_1 \frac{As}{Ld} _oY_\beta' \qquad (1)$$

	η Type of Skeg	0.75	0.67	0.50	0.33	Without Skeg
Y_β'	Mariner	0.34	0.30	0.30	0.30	
	Deformed	0.47	0.37	0.34	0.33	0.26
	Slotted Flap	0.62	0.62	0.47	0.43	
N_β'	Mariner	0.098	0.103	0.114	0.128	
	Deformed	0.065	0.070	0.104	0.104	0.153
	Slotted Flap	-0.040	-0.028	0.031	0.047	
$\dfrac{N_\beta'}{Y_\beta'}$	Mariner	0.288	0.343	0.308	0.427	
	Deformed	0.138	0.189	0.306	0.315	0.588
	Slotted Flap	-0.065	-0.045	0.066	0.109	

Table 3 - Derivatives

$$N_\beta' = {}_B N_\beta' - C_2 \frac{A_s}{Ld} \left[\frac{\ell}{L} {}_o Y_\beta' + \frac{L_s}{L} {}_o N_\beta' \right]$$

(2)

where

${}_B Y_\beta'$: Y_β' for the hull without the skeg

${}_o Y_\beta'$, ${}_o N_\beta'$: Hydrodynamic Coefficients of Skeg in Open Water

ℓ : Distance from midships to the axis of the skeg moment

L_s : Skeg Length

A_s : Skeg Profile Area

C_1, C_2 : Interaction Coefficients

The interaction coefficients C_1 and C_2 are presented in Table 6. When the spacing between skegs becomes large, the values of C_1 and C_2 become nearly equal to 1.0 and the interaction with the hull can be ignored.

Table 6 - Values of C_1 and C_2

	Type of Skeg η:	0.75	0.67	0.50	0.33
C_1	Mariner	1.07	0.46	0.42	0 42
	Deformed	1.05	0.70	0.73	0.73
	Slotted Flap	0.97	0.87	0.58	0.58
C_2	Mariner	1.14	0.79	0.53	0.56
	Deformed	1.04	0.90	0.67	0.75
	Slotted Flap	1.03	0.82	0.64	0.67

6. SKEGS AND COURSE STABILITY PERFORMANCE

In the previous section the oblique towing test results were used to determine the towed vessel stability performance to a certain degree. In this section the effect of the skeg on the course stability is studied using the general expression for Δ to decide the course stability performance:

$$\Delta = Y_\beta' \cdot N_r' - (m_x - Y_r') \cdot N_\beta'$$

(3)

Type of Skeg	Y_β'	N_β'	$m_y - Y_r'$	N_r'	Δ
Without	0.26	0.146	0.449	0.014	-0.06919
Mariner	0.34	0.098	0.275	0.039	-0.0137
Deformed	0.47	0.065	0.235	0.052	0.0092
Slapped Flap	0.62	-0.040	0.285	0.044	0.0205

Table 7 - Course Stability (η = 0.75)

From the results of the investigations in the previous section, the skegs were most effective when $\eta = 0.75$ and for this condition rotating arm turning tests were made to obtain the values of Y_r' and N_r'. These results are shown in Table 7 and the following comparison of the course stability was obtained from this table.

When the M.T.S. is fitted to a hull whose course stability performance was originally unstable, the performance is slightly improved and the hull may be called slightly unstable. When the D.T.S. is fitted to this hull, the stability is further improved and the hull may be called slightly stable. The SFTS causes the hull to become extremely stable. Comparing the effect on the hull from D.T.S. and SFTS, it appears there is not a large variation in the turning derivatives, but there is a large difference in the static derivatives for oblique motion tests. This is especially obvious in the variation in N_β'.

Consequently, from considering only the course stability performance, it appears that the SFTS gives the maximum effect.

However, there is another problem that when the skegs are fitted the resistance increases. Assuming that the barge is usually towed at about 5 knots, the corresponding added resistance of the barge with each type of skeg are shown in Table 8. Examining this comparison of the barge hull resistance with-without skegs, when the SFTS are fitted there is an increase of 40% in the resistance. However, with the D.T.S. this increase is limited to 10%. In practice there are many cases in which it is necessary to fit skegs to the barge and since there is hardly any noticeable difference between the D.T.S. and the M.T.S. with symmetric airfoil section, if appears that since the D.T.S. contributes more stability from considerations of course stability the D.T.S. is the best skeg from resistance and course stability considerations.

Consequently, it is necessary to include consideration of not only the side force and moment caused by the skeg fitted to the towed barge but also the skegs' resistance. Consequently, when developing the optimum skeg design for actual vessels both of these aspects should be used in the evaluation.

Without Skeg	1.0	-
Mariner Type Skeg (MTS)	1.07	1.0
Deformed Type Skeg (DTS)	1.11	1.01
Slotted Flap Type Skeg (SFTS)	1.40	1.30

$\eta = 0.75$, $U = 5$ knots

Table 8   Rate of Resistance Increase

7. CONCLUSIONS

In the above investigation, the method of using skegs to improve the course stability performance of an unstable barge hull form was studied. The skegs caused different degrees of improvement in the course stability performance depending on the type of skeg and the position of its installation. For the three types of skegs examined, the course stabilty perfromance became increasingly better by using: Mariner Type Skegs (MTS), Deformed Type Skegs (DFS), and Slotted Flap Type Skegs (SFTS). It was also shown that the course stability performance would improve as the skeg spacing increased.

The SFTS caused the largest course stability improvement amoung the skegs examined. However, using this skeg type results in a large increase in the towing resistance. Consequently it can be concluded that it is worthwhile to to use the SFTS to improve the barge course stability when it is possible to accept the added resistance from these skegs. However, it is better to use the DFS in order to reduce the added resistance if it is possible to accept a smaller course stability.

Finally, giving consideration to the skeg performance (resistance and course stability improvement) is useful when studying the towed barge.

The authors would like to express their sincere thanks to Messrs. Murakami, Katsuno, and Nagiiri for their cooperation in making the experiments as well as to Mr. Todo who conducted these experiments as part of his graduation thesis.

LITERATURE

1. Inoue, S., et. al., "Some Studies of the Course Stability of Towed Ships Systems," TRANSACTIONS OF THE WEST JAPAN SOCIETY OF NAVAL ARCHITECTS, No. 50, August, 1975. pp. 65-74. [In Japanese] TRANSLATION: Improving Barge Towing Translation No. B-I-2, R. Latorre

LINES OF BARGE

FIG. 1   Plan of Model Barge

FIG. 2-1   Mariner Type Skeg
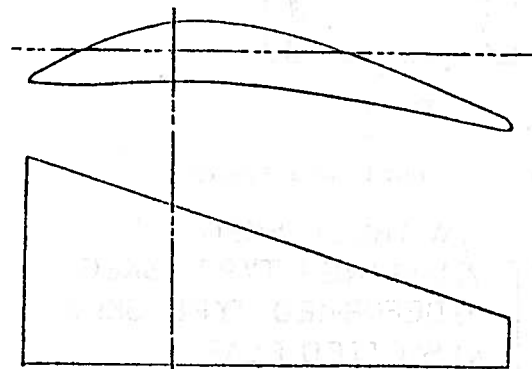
FIG. 2-2   Deformed Type Skeg

FIG. 2-3   Slotted Flap Type Skeg

$$\eta = \frac{b_s}{B}$$

FIG. 3 Coordinate System

1) WITHOUT SKEG
2) MARINER TYPE SKEG
3) DEFORMED TYPE SKEG
4) SLOTTED FLAP TYPE SKEG

$\eta = 0.67$

FIG. 5 Lateral Force Acting on Barge

1) WITHOUT SKEG
2) MARINER TYPE SKEG
3) DEFORMED SKEG
4) SLOTTED FLAP TYPE SKEG

$\eta = 0.50$

FIG. 6 Lateral Force Acting on Barge

1) WITHOUT SKEG
2) MARINER TYPE SKEG
3) DEFORMED TYPE SKEG
4) SLOTTED FLAP TYPE SKEG

$\eta = 0.75$
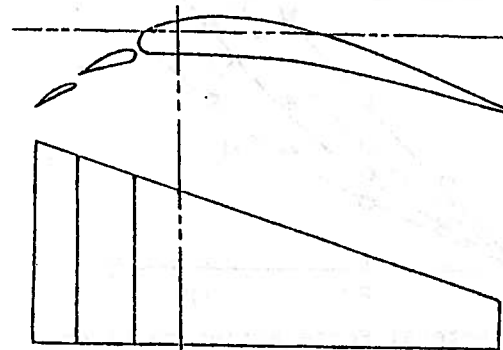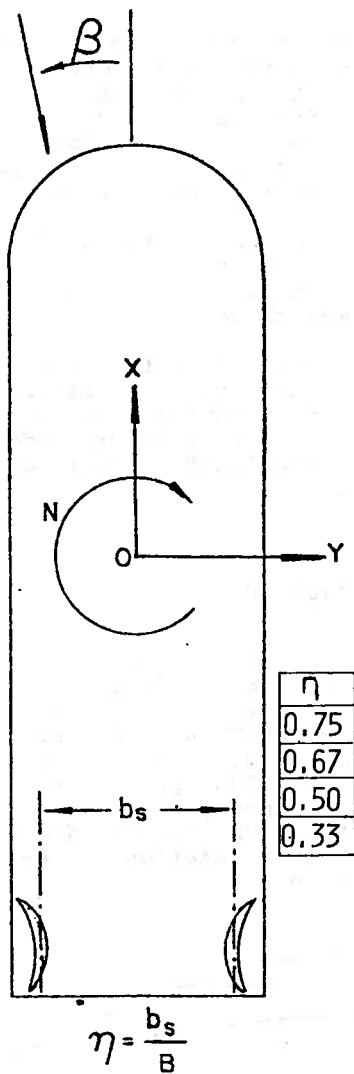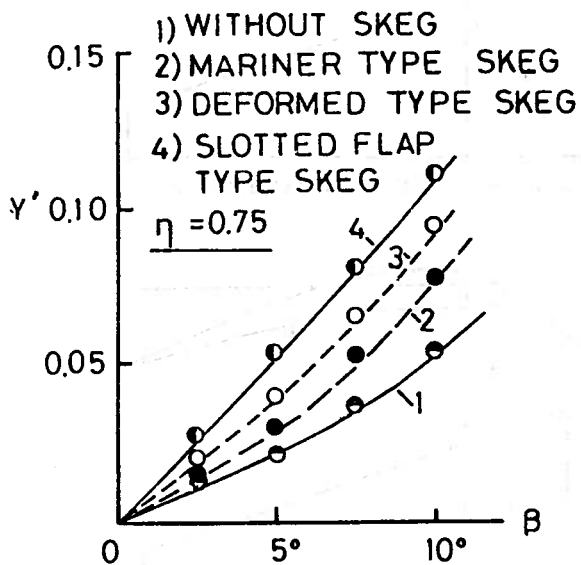
FIG. 4 Lateral Force Acting on Barge

1) WITHOUT SKEG
2) MARINER TYPE SKEG
3) DEFORMED TYPE SKEG
4) SLOTTED FLAP TYPE SKEG

$\eta = 0.33$
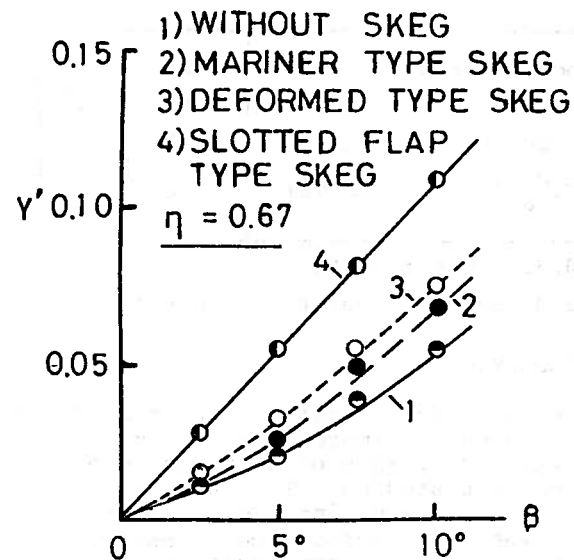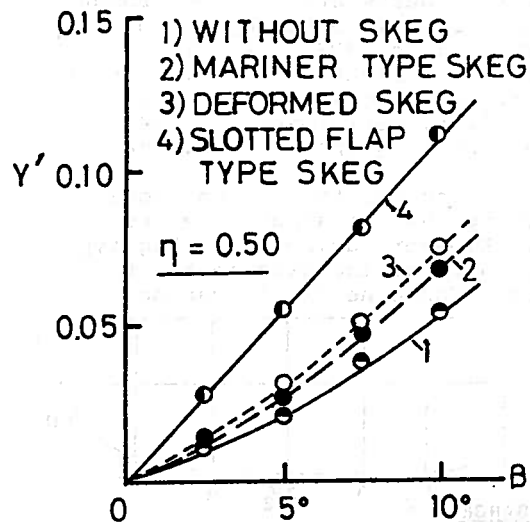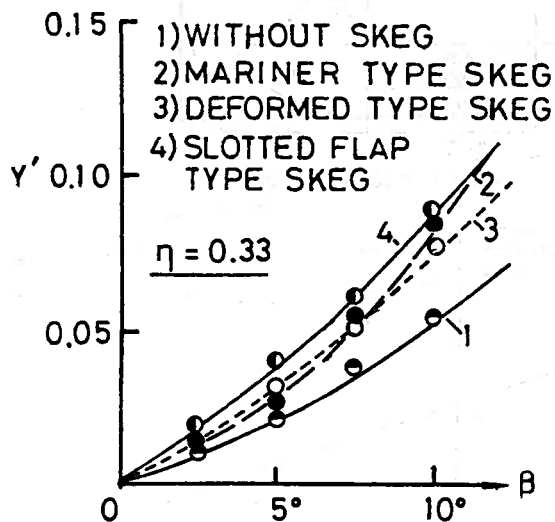
FIG. 7 Lateral Force Acting on Barge

-35-
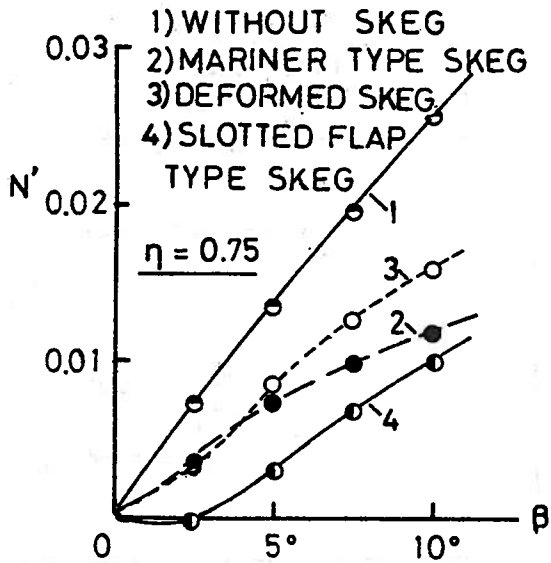
FIG. 8 Yawing Moment Acting on Barge
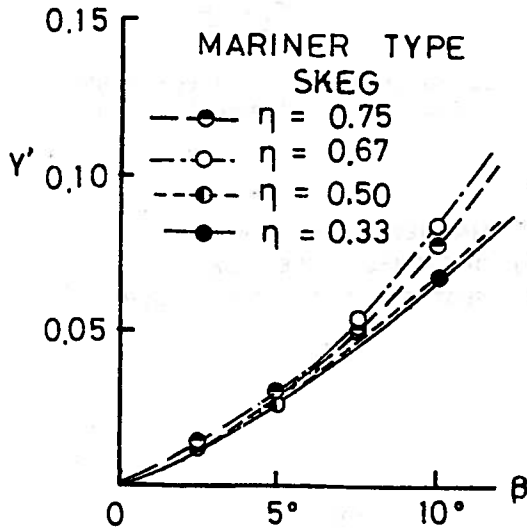
FIG. 9 Lateral Force Acting on Barge
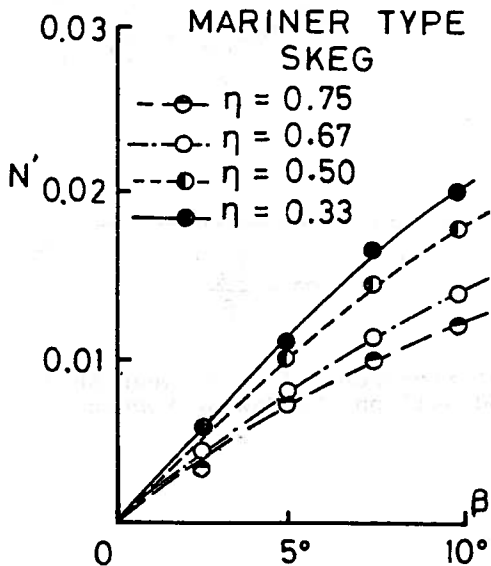
FIG. 10 Yawing Moment Acting on Barge
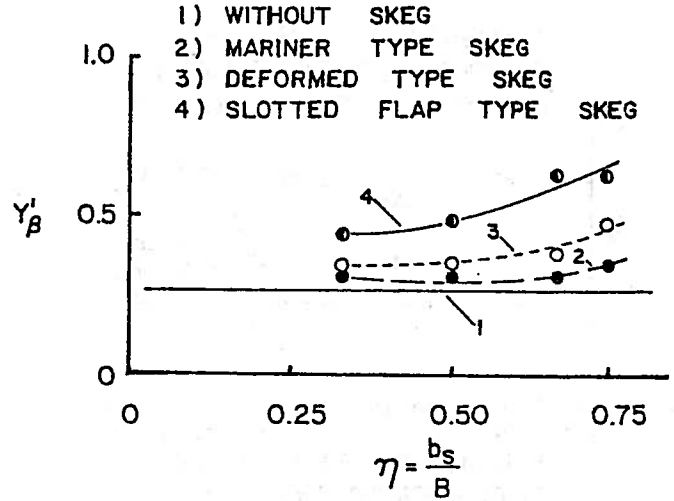
FIG. 11 Static Derivative of Lateral
Force Acting on Barge
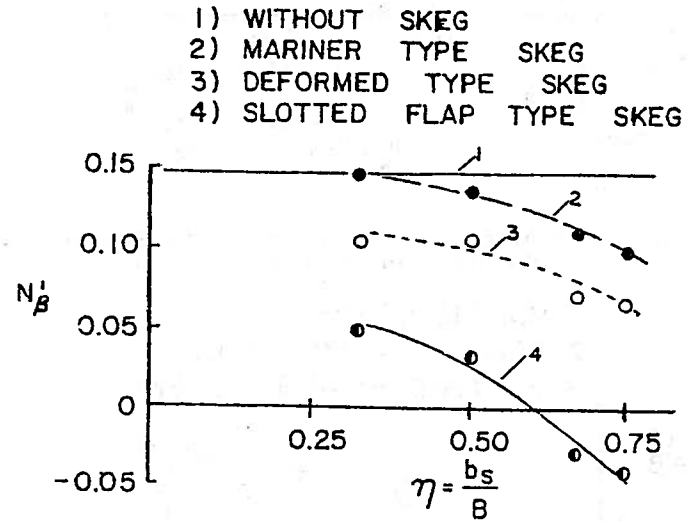
FIG. 12 Static Derivative of Yawing
Moment Acting on Barge
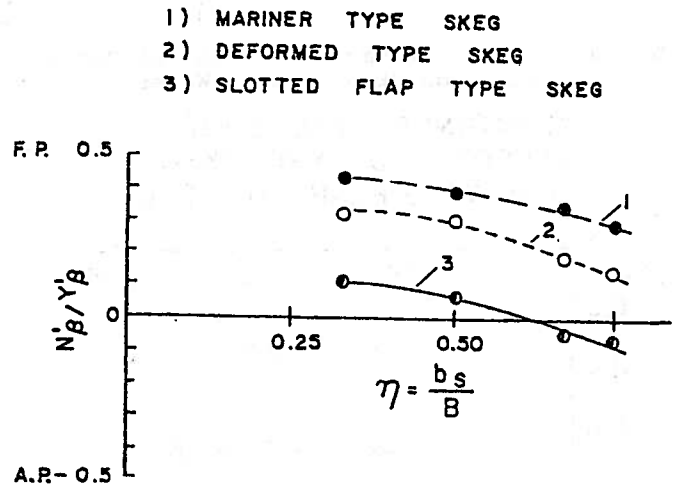
FIG. 13 Center of Action of Lateral Force

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1) MARINER TYPE SKEG
2) DEFORMED TYPE SKEG
3) SLOTTED FLAP TYPE SKEG
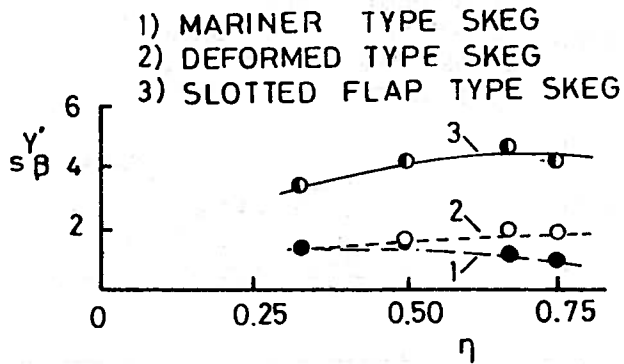
FIG. 14　Static Derivative of Lateral Force
Acting on Skeg Behind the Barge Hull

1) MARINER TYPE SKEG
2) DEFORMED TYPE SKEG
3) SLOTTED FLAP TYPE SKEG
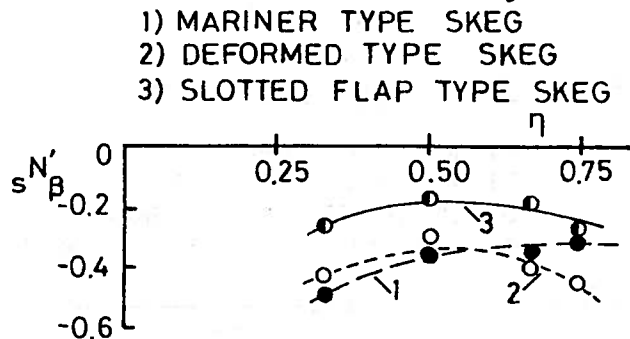
FIG. 15　Static Derivative of Yawing Moment
Acting on Skeg Behind the Barge Hull

1) MARINER TYPE SKEG
2) DEFORMED TYPE SKEG
3) SLOTTED FLAP TYPE SKEG
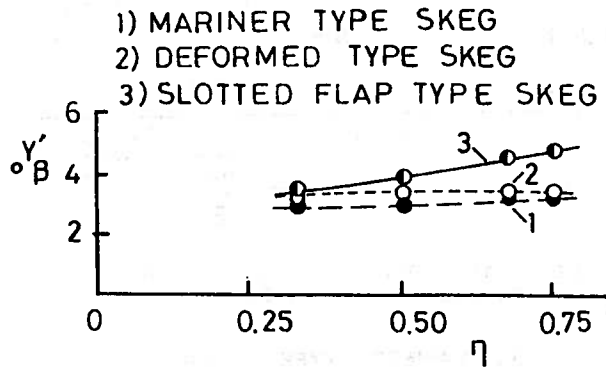
FIG. 16　Static Derivative of Lateral Force
Acting on Skeg in Open Water

1) MARINER TYPE SKEG
2) DEFORMED TYPE SKEG
3) SLOTTED FLAP TYPE SKEG
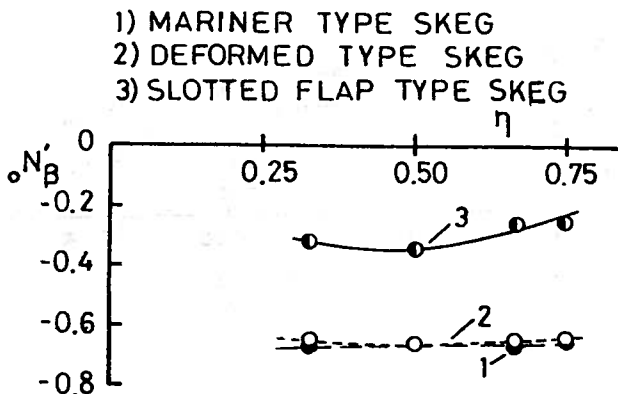
FIG. 17　Static Derivative of Yawing Moment
Acting on Skeg in Open Water

1) MARINER TYPE SKEG
2) DEFORMED TYPE SKEG
3) SLOTTED FLAP TYPE SKEG

$\eta = \dfrac{b_s}{B}$
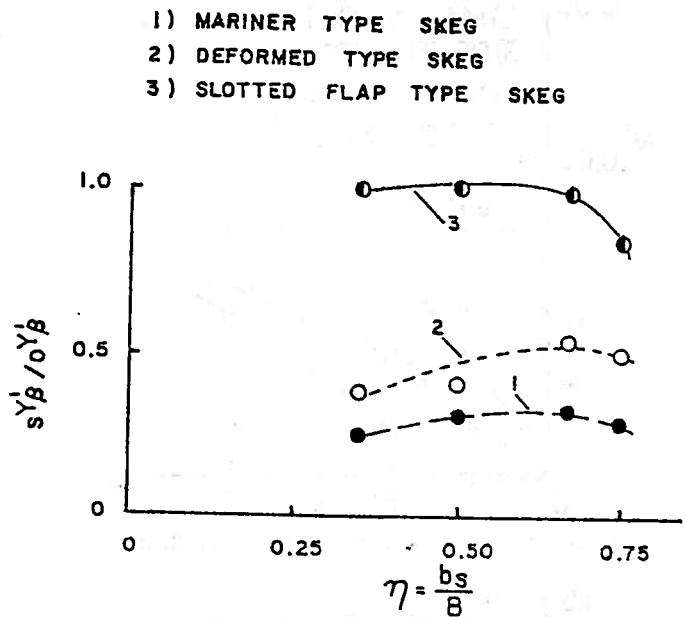
FIG. 18　Interference Effect Between Skeg
and Hull on the Lateral Force

1) MARINER TYPE SKEG
2) DEFORMED TYPE SKEG
3) SLOTTED FLAP TYPE SKEG

$\eta = \dfrac{b_s}{B}$
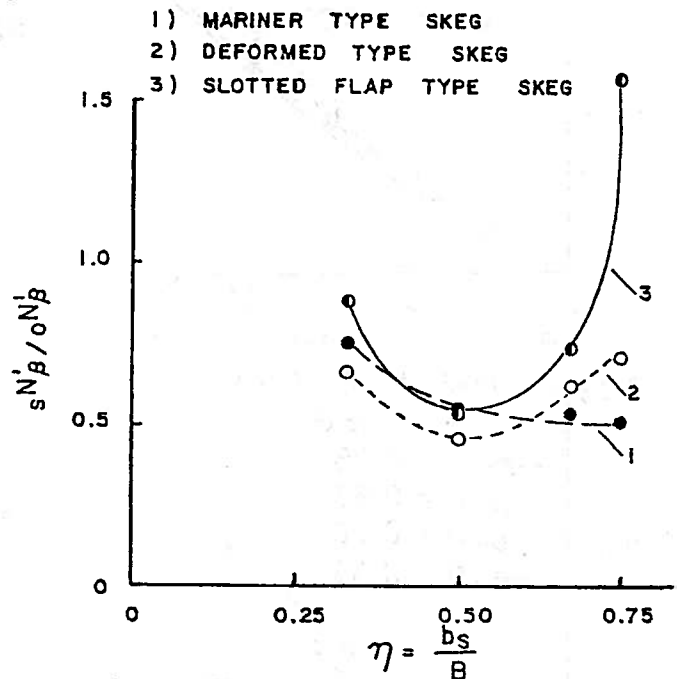
FIG. 19　Interference Effect Between Skeg
and Hull on the Yawing Moment

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EXPERIMENTAL STUDY ON THE COURSE STABILITY OF
A TOWED BARGE[1]

BY

M. TANAKA[2]

ABSTRACT

The ever-increasing demand for barges
presents an immediate problem due to the
fact that the barge towed by a tug often
swings (yaws). This problem is of crucial
importance when a towed barge negotiates
restricted or congested waters. However
the course stability of a towed barge
depends on many factors. The method of
towing as well as the respective stability
of the barge and tug play a significant
role. Further, since barges in general
have principal dimensions and hull-forms
which are different from conventional ships,
the estimate of the stability of barge
is a formidable problem.

In this paper experimental studies
on the course stability of a towed barge
are presented, covering two kinds of tests:
(i) The respective force and moment act-
ing on a barge and skegs were measured by
means of oblique towing test. i.e. the
influence of the type of skegs as well as
their locations on the stability was
clarified. & (ii) Towing tests with syste-
matically varied tow-line and bridle lengths
were carried out, and the barge motions
were investigated.

The following results relating to the
course stability of a towed barge were
obtained:

(1) The barge equipped with skegs is
more stable than without skegs. The two
Slot Skeg is more effective than any other
types tested.

(2) It is advisable for the distance
between skegs measured   with respect
to the center line of the barge to be as
wide as possible.

(3) The longer the tow-line and
bridle lengths are, the more stable the
towed barge becomes.

(4) Whether the barge is equipped
with skegs or not, the greater the course
stability of the barge itself, the more
stable the towed barge becomes.

(5) When the tow-line length is fixed,
the more stable the barge hull becomes, the
shorter the period of motion becomes.

(6) It is generally accepted that the
greater the contribution of skegs to the
stability is, the more the resistance in-
creases. However, since the skegs tested
cause no significant increase of resistance
(less than 7%), the use of skegs should be
considered from the viewpoint of course
stability.

TRANSLATION[3]

Recently there has been a phenomenal
increase in the demand for the transporta-
tion of ocean borne construction material
by towed barges. With this boom in barge
transportation the towing has become a
problem.

In the towed condition, there is a
chance for the towed vessel to have a
large amount of swaying to the right and
left. There are many risks that the
vessel might disrupt nearby navigation as
well as straying outside the limits imposed
by the navigation. In addition, the trans-
portation distance is increasing accom-
panied by barges of larger tonnage. There-
fore, the towed barge swaying to the right
and left results in meandering which cannot
be ignored when the towing distance increases.

The course stability of towed barge
vessels does not depend solely on the
course stability performance of each barge
or towed vessel. Rather it depends on the
stability performance of the entire towing
system and the method of towing.

In addition, since there are large
differences in the main dimensions, bow
and stern in comparison with conventional
ships, it is difficult to accurately de-
termine, even for individual vessels, the
course stability performance from
the previously accumulated information.

The author made oblique towing tests of a wide beam and shallow draft barge with a cutaway (beveled) bow and stern. This model was used to investigate the effect of fitting skegs to the stern have on the barge form's course stability. The following discusses the investigation of the barge motion as well as the effects observed in systematic tests which varied the skeg design, towrope length, and the bridle length.

2. EXPERIMENTAL MODEL HULL AND SKEGS

The model used in the experiments is shown in Fig.1. Its main particulars are summarized in Table 1. Typically, most barges are designed with a spoon bow and a beveled stern. However, since the speed of the barge used in this study is about 7 knots, which is a relatively low speed, the hull form was derived by simply beveling the bow and stern.

Two skegs used were the same design which Inoue [1] fitted to the stern of a "spoon bow" barge

Table 1   Principal Dimensions of Model Barge

L_{pp} (m)	2.500
B (m)	0.625
d (m)	0.044
C_b	0.97
Δ (kg)	67.5

The four types of skegs used in this investigation had vertical profiles which fitted into the stern cutaway. These skegs are distinguished by their cross sectional profile as follows: Two slot skeg (TSS) which has two slots. One slot skeg (OSS)which has one slot. Mariner Type skeg (MTS) which has a symmetric airfoil section and the Plate Skeg (P.S.) which has a thin flat plate section. The design of the skegs are shown in Figs. 2(a)-2(d). The main particulars of the skegs are summarized in Table 2.

Table 2   Principal Dimensions of Skegs.

Length (m)	0.10
Area (m²)	0.0024
Area Ratio	1/22.9
Aspect Ratio	0.24

3. TYPES OF EXPERIMENTS AND METHODS

3.1 Oblique Towing Test

The hydrodynamic forces were obtained by oblique towing tests of captive models. An investigation of the effect of skeg form and location on the course stability was made along with a study of the barge course stability without skegs.

The test parameters varied were:
1. Skeg Type, which including the case of no skeg represents five conditions.

2. Drift Angle β, β=15°∿0°∿15°

3. Location of Skegs, which consisted of two conditions (η= 0.84 and 0.42) Here when η = 0.84 the skegs are installed at the widest possible spacing.

The notation and coordinate system are shown in Fig. 3.

The barge hull drag force (X), side force (Y) and moment about its center (N) were simultaneoulsy measured with the forces acting on the skegs. These forces were non-dimensionalized using the water density ρ and hull velocity (U=0.38 m/s) as expressed below:

$$X' = \frac{X}{\frac{1}{2}\rho L_{pp} d U^2}$$

$$Y' = \frac{Y}{\frac{1}{2}\rho L_{pp} d U^2}$$

$$N' = \frac{N}{\frac{1}{2}\rho L_{pp}^2 d U^2}$$

3.2 Towing Test

In the towing tank tests of the models at an initial displacement,the barge model is restrained by a clamp at an approximate offset of 1.6xB. When the towing carriage reached a constant speed, the model was released and measurements were made of the towrope tension, angle,as well as the barge's turning angle and angular velocity. There was special concern to avoid having the measurement chord influence the motions of the model. Therefore,a small gyro and batteries were installed and during the tests the data was transmitted to the towing carriage by using radio telemeter. Thus the only connection between the carriage and models was the piano wire used for the towrope. These offset towing tests were made at a constant .38m/s speed while systematically varying the towrope length L and bridle length ℓ as described below:

1) Towrope length L, L=L_{pp}, 2L_{pp}, 3L_{pp}

2) Bridle length ℓ, ℓ=0, 0.1 L_{pp}, 0.2 L_{pp}, 0.3 L_{pp} where L and ℓ are defined as shown in Fig.4.

When the model is towed by a towing carriage, the tests correspond to towing with an extremely large towing vessel

		Y'_β	N'_β	N'_β/Y'_β	$\dfrac{A}{Lpp \cdot d}L'_\beta$
	Without Skegs	0.386	0.058	0.150	–
$\eta = 0.84$	Two Slot Type Skeg(TSS)	0.502	0.027	0.054	0.074
	One Slot Type Skeg(OSS)	0.478	0.041	0.086	0.025
	Mariner Type Skeg(MTS)	0.393	0.084	0.214	–
	Plate Type Skeg(PS)	0.421	0.068	0.162	–
$\eta = 0.42$	Two Slot Type Skeg(TSS)	0.449	0.047	0.105	0.025
	One Slot Type Skeg(OSS)	0.386	0.056	0.145	0.014
	Mariner Type Skeg(MTS)	0.295	0.072	0.244	–
	Plate Type Skeg(PS)	0.263	0.080	0.304	–

TABLE 3 Derivatives

There is a towing method [2] using a model towing ship which nearly corresponds to the actual towing conditions. However due to the difficulties in the measurements and the towing tank width limitations, this method was not used and the experiments were made using the towing carriage.

In these towing tests the spacing between the skegs was set at $\eta = 0.84$.

4. RESULTS OF EXPERIMENTS

4.1 Results of Oblique Towing Tests

The influence of the skegs on the barge hydrodynamic derivatives will be discussed. The notation adopted is mainly taken from the text of the Second Maneuvering Symposium [3]. The derivatives of the barge hull without the skegs are denoted by "o" and written as: $_oY'_\beta$, $_oN'_\beta$, $_oY'_r$, and $_oN'_r$.

Ignoring the interference between the hull and skegs, the derivatives of the barge hull with skegs are written:

$$Y'_\beta = {}_oY'_\beta + \frac{A}{Lpp \cdot d}L'_\beta$$

$$N'_\beta = {}_oN'_\beta - \frac{A}{Lpp \cdot d}\frac{\ell s}{Lpp}L'_\beta$$

$$Y'_r = {}_oY'_r + \frac{A}{Lpp \cdot d}\frac{\ell s}{Lpp}L'_\beta \qquad (1)$$

$$N'_r = {}_oN'_r - \frac{A}{Lpp \cdot d}\left(\frac{\ell s}{Lpp}\right)^2 L'_\beta$$

where:

ℓ_s : The distance from Φ to the skeg position
Here $\dfrac{\ell s}{Lpp} \doteqdot 0.5$

A : Projected area of the vertical skeg cross section

The second term on the right hand side of equation (1) expresses the influence of the skegs on the hydrodynamic derivatives. Considering that the term $\dfrac{A}{Lpp \cdot d}L'_\beta$ adds a value of 1 to Y'_β, the corresponding increase in the value of N'_β and Y'_r is 0.5, and the corresponding increase to N'_r is 0.25.

Consequently, there is a large effect from installing the skegs on the value of Y'_β followed by N'_β and Y'_r. This was also found in the previously reported results of Inoue [1] and Takekawa [4]. Therefore it is possible to determine the optimum skeg by oblique towing tests of captive models.

The course stability of a vessel is indicated by the Δ index.

$$\Delta = Y'_r - (m' + m'_x) N'_\beta - N'_r Y'_\beta \qquad (2)$$

Substituting (1) into (2), the effect of the skegs on the hull course stability is as follows:

$$\Delta \doteqdot \Delta_o + \frac{A}{Lpp \cdot d}L'_\beta \left[\frac{\ell s}{Lpp} {}_oN'_\beta - \frac{\ell s}{Lpp}({}_oY'_r - (m' + m'_x) - {}_oN'_r \right] \qquad (3)$$

Since the value of the terms inside the brackets [ ] is usually positive, there will be an improvement in the barge course stability when the skegs with a value of $L'_\beta > 0.0$ are installed behind the hull's center of gravity. There will be a phenomenal improvement when L'_β is large.

Comparing the derivative values for the barge with and without the skegs, when the skegs are present the Y'_β values increase while the value of N'_β decreases so N'_β/Y'_β, the location of the center of pressure moves backward.

While the course stability is improved by the four types of skegs, the best course stability is obtained by the TSS followed by the OSS. In comparison with these skegs the MTS and PS skegs cause only a small improvement. These experimental results are the same as those obtained by Inoue [1] for a barge model with a spoon bow.

It was found that while the course stability performance improves when $\eta = 0.84$, when $\eta = 0.42$, there is little difference in the course stabilty performance with and without the skegs. Usually when foils are arranged in a line their performance characteristic is different due to the influence

of adjacent foils. When skegs are fitted to a barge they are arranged symmetrically along the hull centerline. When the separation distance becomes small an effect similar to the one observed for foils in a straight line occurs and the resulting lift force of the skeg decreases.

Therefore a large improvement in the course stability performance can be obtained by installing skegs which generate high lift forces. The skeg locations should be as far as possible behind the center of gravity of the hull and the separation distance should be as wide as possible.

4.2 TOWING TEST RESULTS

The aft clamp was released when the towing carriage reached a constant speed. Then after an interval of time, the displacement of the barge from the tank centerline would decrease. However the barge motion was extremely large when its initial displacenment was about 1.6xB when the aft clamp was released.

In these tests while there was a decrease in the barge motion with time, during the towing tests the data indicated that the barge was course unstable for very small motions. It thus became clear that the decrease in the motion is due to the large initial displacement.

The following two coefficients were adopted to treat this decrease in motion:

The first is a dissipation coefficent. Here a technique of analysis similar to that used in free rolling experiments is utilized. Using the dissipation curve of the turning angle θ, the coefficient M is show in Figs. 5(a)-5(b) was obtained.

$$\Delta\theta = M \cdot \theta_m^2 \qquad (4)$$

The second is the coefficient of the time average of the turning angle $|\dot{\theta}|_{avg}$. This coefficient is related to the damping motion.

$$|\dot{\theta}|_{avg} = \frac{1}{\theta_o T} \int_o^T |\theta| dt \qquad (5)$$

where at $t = 0$, $\theta = \theta_c$
The time average coefficient is shown in Figs. 6(a)-6(b).

The following treats the damping of the motion during towing in order to study the influence of the skeg type, towrope length, and bridle length on the barge course stability. In other words, the course stability performance is good when there is a strong damping of the barge motion. This occurs when the coefficient M in expression (4) has a large value and when the value of $|\dot{\theta}|_{avg}$ in expression (5) is small.

Therefore in this discussion the turning angle analysis is presented using the coefficients described earlier. The other items measured in the tests will not be discussed except for the towrope tension.

4.2.1 Effect of Towrope Length, L

Figs. 5(a) and 6(a) show the trends in the coefficients M and $|\dot{\theta}|_{avg}$ with different lengths of towrope L . When the towrope length is large there is a strong damping of the motion and the course stability performance is good. The coefficients do not appear to have a linear relationship with the towrope length L. For example in Fig. 5(a) for $\ell/Lpp = 0$, the results for the two slot skeg (TSS) indicate that when the value of L = 3 Lpp, there is a drop off in the increase of M. In Fig. 6(a) for $\ell/Lpp = 0$, the value of $|\dot{\theta}|_{avg}$ for the two slot skeg (TSS) at L = 2Lpp indicates better course stability performance than at L = 3Lpp. It appears that a large improvement in the barge course stability will not be obtained when larger values of L are used.

This figure also indicates that when the original course stability performance is poor there will be hardly any improvement from increasing the value of L/Lpp when the value of ℓ/Lpp is small.

4.2.2 Effect of Towing Bridle Length, ℓ

With the exception of L = 3Lpp, it appears from examining Fig. 5(b) that there is a linear relationship of the coefficient M with the value of ℓ. In the case of L = 3Lpp the coefficient M has its largest value when $\ell = 0.1Lpp$. This tendency is especially pronounced for the two slot skeg (TSS) and one slot skeg (OSS) whose course stability performance indicated by M appear to be as good as the barge itself. It should be also noted that with these skegs (TSS and OSS) the course stability performance is better at $\ell = 0.1Lpp$ than at $\ell = 0.3Lpp$.

The barge course stability performance becomes extremely good when either the TSS or OSS skegs are fitted. This was previously observed in the inclined towing test results. Consequently even when towing at a towrope length of L = 3Lpp, the course stability performance of the barge with these skegs is very good. In these tests at L = 3Lpp when the clamp was released the motion was so damped as to become periodic. In this situation the expression for M becomes meaningless.

From the values of $|\dot{\theta}|_{avg}$ shown in Fig. 6(b) it appears that there is an inverse relationship between the bridle length ℓ and $|\dot{\theta}|_{avg}$. Thus $|\dot{\theta}|_{avg}$ becomes small when the towing bridle length ℓ becomes large and the course stability performance improves

The turning angle θ was also assumed to have the following governing equation:

$$\theta = \theta_o \cdot e^{-\frac{t}{T_D}} \qquad (6)$$

The time constant T_D is shown in Fig. 7. This constant exhibits the same trends as $|\dot{\theta}|_{avg}$.

4.2.3 Influence of Barge Hull on the Course Stability Performance

Figs. 5(a),5(b),6(a),6(b), and 7 indicate that the differences in the towing course stability seem to have the same trends as the barge without skegs. It can be expected that a course stable barge hull form will have good course stability when it is towed.

It has been pointed out that the course stability in towing improves when longer towrope and towing bridle lengths are used. However, since there are limits to these lengths imposed by towed barge's turning performance as well as its interaction with other vessels, it is not expected that there will be further improvements from using longer towrope and bridle lengths.

While it is desirable that the hull form design has good course stability performance, if the barge's course stability is in question then it is advisable to improve the course stability by using a suitable skeg design.

4.2.4 Period of Barge Motion

The relation between the towing bridle length ℓ and the average period of the barge motion T_m is shown in Fig. 8. The barge has good course stability performance when the value of ℓ increases and the value of T_m is small. Assuming that the towed barge system can be modeled as a pendulum the length of the pendulum's string corresponds to the towrope length L. The natural period of the system is then related to $\sqrt{L}$ and the damping force acting on the system is related to the skeg type and the towing bridle length ℓ. Thus the motion becomes small as L becomes large and the value of T_m increases.

4.2.5 Relationship of Skegs to Resistance

The average tension force during the towing tests is shown in Fig. 9 for L = 2Lpp and L = 3Lpp. For each type of skeg these values are constant irrespective of L and ℓ. The largest tension forces were measured for the two slot skegs (TSS) and the one slot skegs (OSS). The average value of tension force representing the hull resistance can be divided into two groups. It appears the skegs in addition to improving the barge course stability performance, cause a noticable increase in the barge's towing resistance.

Consequently it is necessary when designing the skeg to consider in addition to the improvement in course stability, the resulting overall resistance necessary to obtain a course stable barge. In this study there is not an exceptionally large increase in the resistance from installing the skegs. The maximum resistance increase is 7%. Thus the most suitable skeg for this barge design is the two slot skeg (TSS) since the course stability improvement is large and its resistance increase is small.

5. CONCLUSIONS

A course stability performance investigation was made for a wide beam, shallow draft barge model tested in a towing tank.

It was shown that oblique towing tests were useful in investigating the the effect of skegs on the barge hull's hydrodynamic derivatives. The influence of skeg type and skeg spacing on the barge course stability performance was investigated using four types of skegs and the oblique towing technique.

Systematic towing tank tests were also conducted to determine the influence of skeg type, towrope length, and towing bridle length on the barge course stability performance.

The phenomenon of the damping in the barge motion was characterized by two coefficients M and $|\dot{\theta}|_{avg}$. The course stability performance was considered good when the barge motion exhibited large damping.

The following summarizes the conclusions obtained in these studies.

1.) It is possible to improve the course stability performance by installing skegs. Among the skegs used in the tests, it was shown that the two slot skeg (TSS) was very effective.

2) It is best to use as wide a separation width between the skegs as possible.

3) Towing with longer towrope and towing bridle lengths resulted in improved course stability performance.

4) The better the barge course stability performance without skegs, the better its course stability will be in towing.

5) In the case of constant towrope length the course stability performance is good when the radius of the barge motion is small. Keeping the other parameters constant and letting the towrope length become longer, the course stability performance will improve and the period of the motion will be longer.

6) The larger the skegs contribution to improving the barge course stability performance, the stronger the damping of the barge motion and the larger the increase in the barge resistance. In the present

study the resistance increase was within 7% so consideration was given exclusively to the selection of skegs to improve the course stability performance.
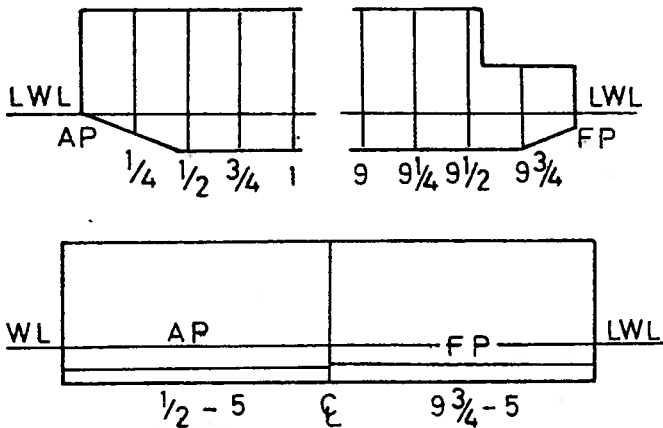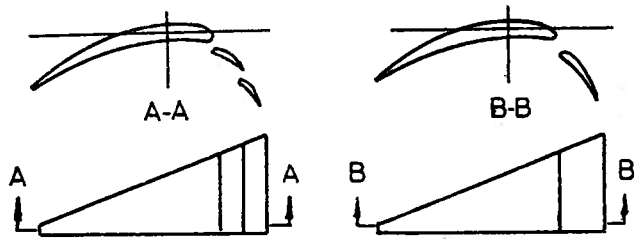
FIG. 1 Lines of Barge Model

FIG. 2(a) Two Slot Skeg (TSS)
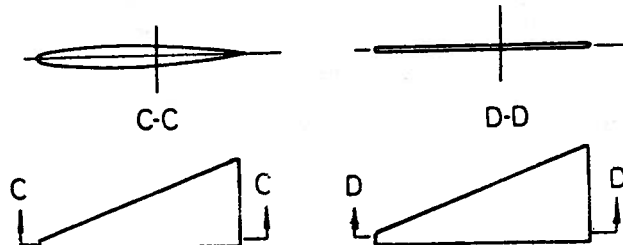
FIG. 2(b) One Slot Skeg (OSS)

FIG. 2(c) Mariner Type Skeg (MTS)

FIG. 2(d) Plate Skeg (PS)

LITERATURE

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2. Inoue, S., et. al., "Some Studies of Towed Ships System," TRANSACTIONS OF WEST-JAPAN SOCIETY OF NAVAL ARCHITECTS, No. 50, August, 1975, pp. 65-74. (In Japanese) TRANSLATION: Improving Barge Towing Translation No. B-I-2, R. Latorre.

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4. Takekawa, M., et. al., "Course Stability of Towed Large Barge," JOURNAL OF THE SOCIETY OF THE NAVAL ARCHITECTS OF JAPAN, Vol. 137, June, 1975, pp 186-195, (In Japanese)

ALSO: Nagamatsu, S. et. al., "Reports on Tank Tests and Full Scale Ship Experiment for Lash Feeder Barge," JAPAN SHIPBUILDING AND MARINE ENGINEERING, Vol. 11, No. 2, 1979.

FIG. 3 Coordinate System

$$\eta = \frac{b}{B}$$
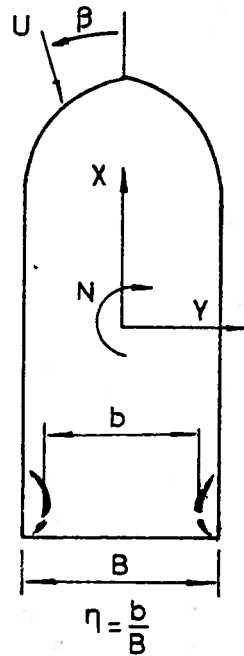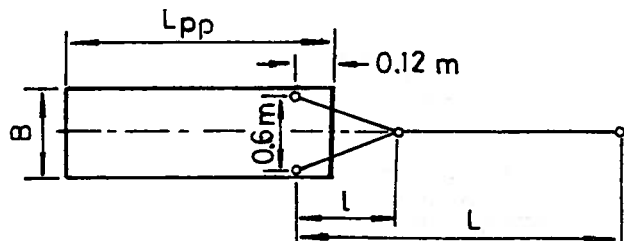
FIG. 4 Definition of T and ℓ

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KEY (FIGS. 5-9)

○——○ WITHOUT SKEGS
△------△ TWO SLOT SKEGS
□—·—□ ONE SLOT SKEGS
✕—··—✕ MAINER TYPE SKEGS
▽—··—▽ PLATE TYPE SKEGS
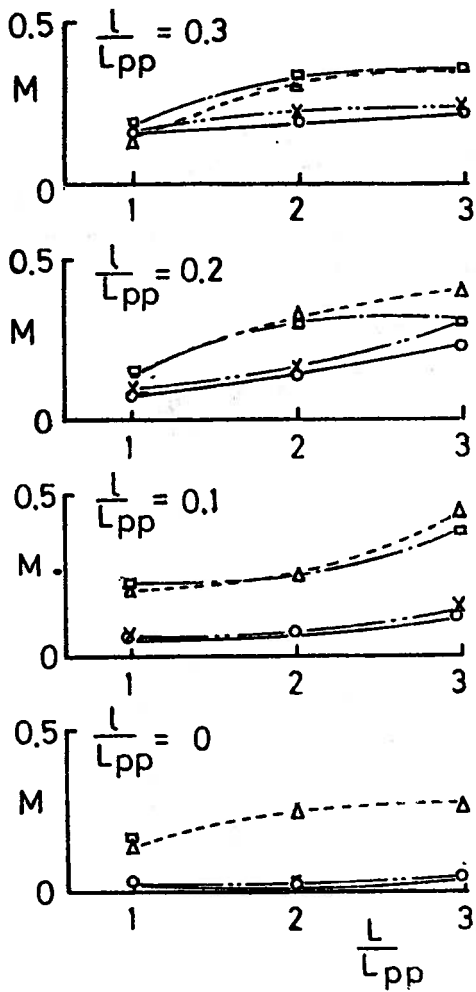
FIG. 5(a)   Extinction Coefficient
(M)

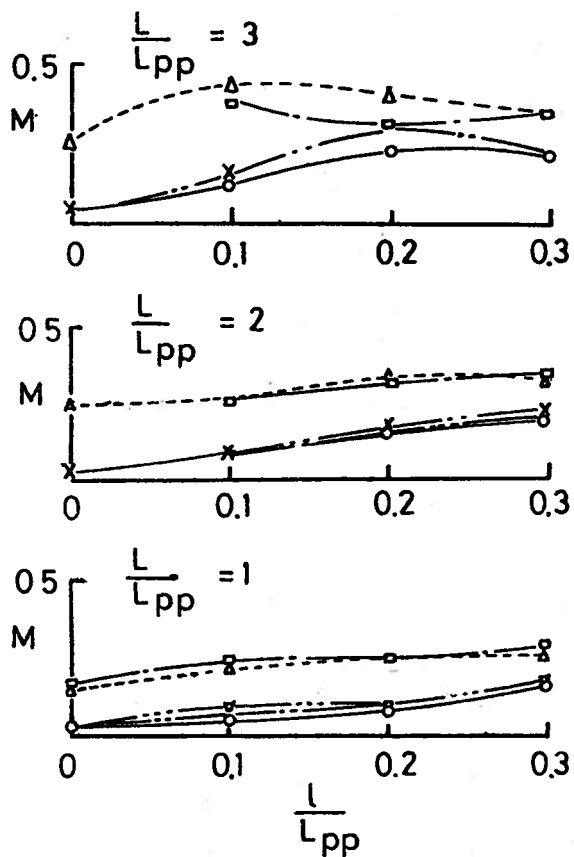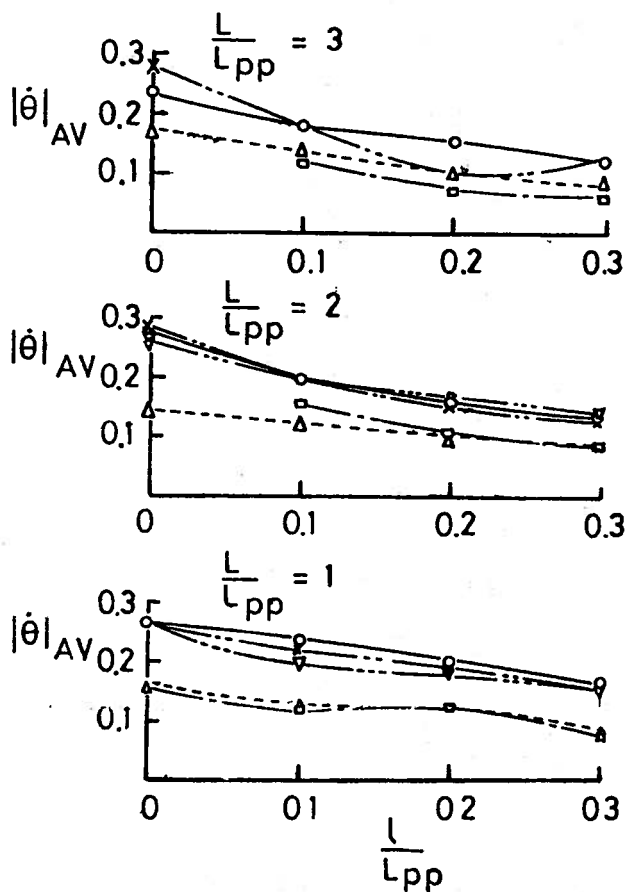FIG. 6(a)   Time Average $|\dot{\theta}|_{avg}$
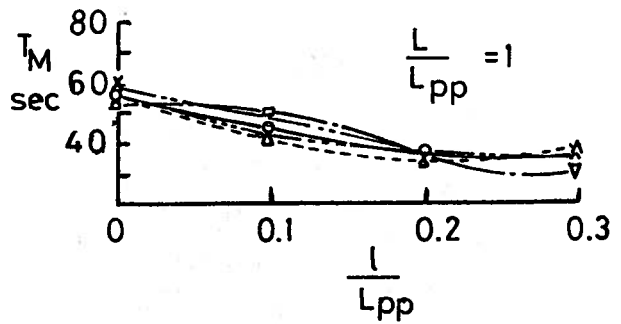
FIG. 5(b) Extinction Coefficient
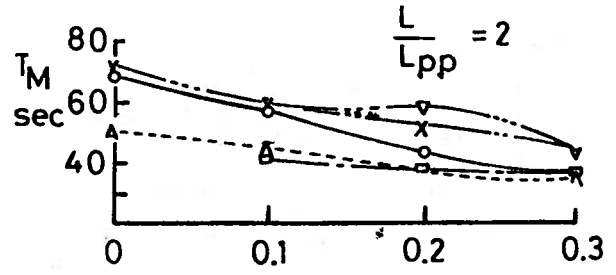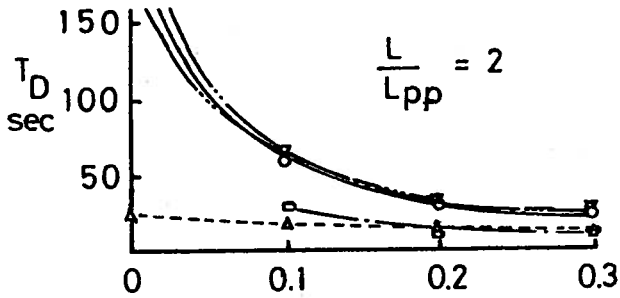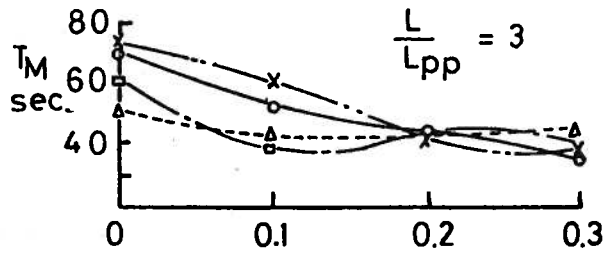(M)

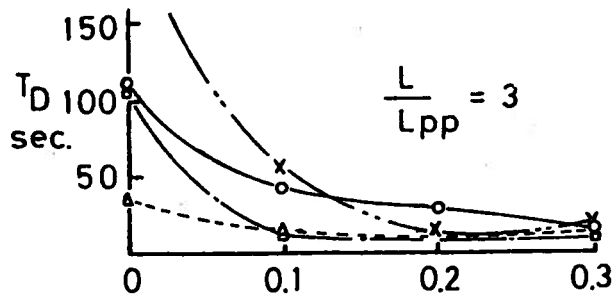FIG. 6(b) Time Average $|\dot{\theta}|_{avg}$
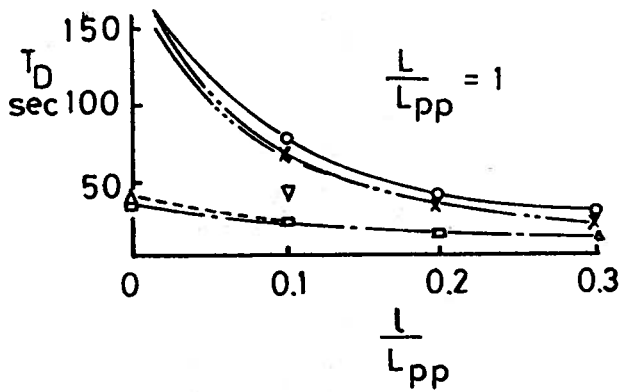
FIG. 7 T_D
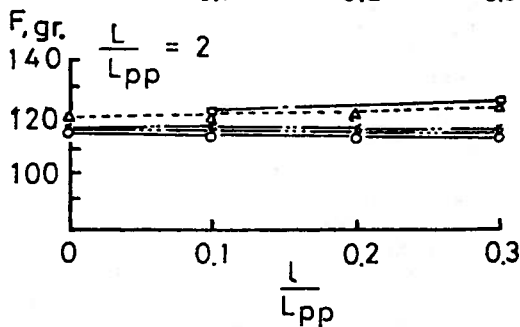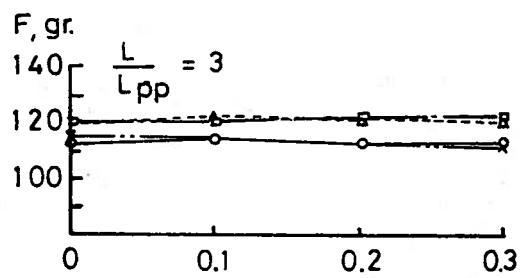

FIG. 8 Mean Period (T_m)

FIG. 9 Mean Towing Force (F)

ANALYSIS OF THE SPONTANEOUS
YAWING OF TOWED SHIPS [1]

BY

E.B. YUDEN
A.G. MIKOVSKY

SUMMARY

This paper presents a theoretical ana-
lysis of ship motion during towing. The
authors treat the phenomenon of spontan-
eous periodic yawing and the influence
the various design variables have on this
yawing. The equations of motion are re-
duced to a characteristic equation to
define the limits of stable towing. The
frequency of the oscillation and the effect
of the bridle on towing are also examined
[Translator].

TRANSLATION [2]

In ocean navigation there often arises
a period during which it is necessary to tow
a single ship. This problem and its re-
lated points have been dealt with for a
long time [1-6]. With the recent level
of theoretical development and experimen-
tal facilities, it is now possible to ob-
tain the data required for constructing
an accurate mathematical description of
the towing process for actual vessels. In
this case, the utilization of an electric
computer makes it possible to readily re-
produce the towing process itself and ana-
lyze each component.

Presently we are concerned with the
development of a general formulation
which includes special features of the
towing process to allow their inclusion
to obtain a practical solution. Among
the features not examined earlier
is the phenomenon observed during towing,
even in the absence of sea waves and wind,
of the spontaneous periodic yawing of
the towed ship having characteristic free
oscillations. Since in practice this os-
cillatory state should be avoided during
towing, it is necessary to understand
the conditions determining the onset of
the oscillations and to estimate the
dependence of their parameters (design)
on the conditioning factors. The investi-
gation indicates that the basic results
can be obtained within the limits of the
traditional assumptions used in character-
izing the response of the towing system.
These are, namely, that during towing the
distance measured as a straight line be-
tween the attachment points of the cables
connecting the towed ships remains con-
stant during towing and the motion takes
place in calm water. Furthermore,

assuming that the towing speed is constant,
the angle parameters determining the rela-
tive position of ships fluctuate in such
limits that the sine of the angle can be
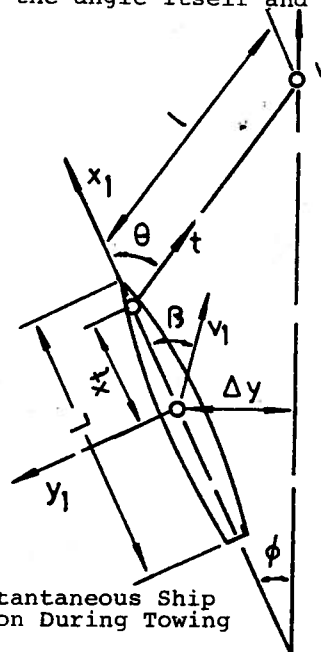replaced by the angle itself and its cosine
by unity.

Fig. 1 Instantaneous Ship
Position During Towing

Fig.1 shows the basic notation for the
geometrical and kinematic qualties en-
countered below. The following equality
can be obtained from considering this
figure:

$$\theta = \frac{\Delta y}{\ell} + \phi(1 + \frac{x_t}{\ell})$$

(1)

$$\frac{d\Delta y}{dt} = -v(\beta - \phi) \cong v(\phi - \beta)$$

Introducing the notation:

$$p = \frac{d}{dt} \qquad \text{Differential Operator}$$

$$\tau = t\frac{v}{L} \qquad \text{Non dimensional time}$$

$K = \dfrac{S \cdot L}{V}$ Scale Factor

S Area of the immersed part of the hull centerplane

V Volume of ship's displacement

$C_T = \dfrac{T}{\rho/2u^2 S} \cdot$ Thrust coefficient (Towing Force Coefficient)

C_1, C_2, m_1, m_2 Hydrodynamic Coefficients

k_{11}, k_{33}, k_{55} Coefficient of added mass

$\bar{r}$ Non-dimensional rotational inertia of ship relative to Z axis

$\bar{x}_t = \dfrac{x_t}{L}$, Relative values of the distance x_T and ℓ

$\bar{\ell} = \dfrac{\ell}{L}$

Following the work [7], the equation of motion for the towed ship taking into account the assumed notation and relation (1) with absence of disturbing forces will assume the form of:

$$2(1 + k_{33})p\beta - 2(1 + k_{11})p\phi + K(C_1\beta + C_2 |\beta|\beta)$$
$$= KC_T\theta$$

$$-2\bar{r}^2(1 + k_{55}^{\bullet})p^2\phi - km_3p\phi + km_1\beta = k\bar{x} \cdot C_T\theta \qquad (2)$$

$$p\theta = \dfrac{1}{\bar{\ell}}(\phi - \beta) + (1 + \dfrac{\bar{x}_t}{\bar{\ell}})p\phi$$

Following reference [7], we introduce the additional notation:

$$q_{21} = \dfrac{KC_1}{2(1+k_{33})} \quad ; \quad q_{31} = \dfrac{Km_1}{2\bar{r}^2(1+k_{55})}$$

$$h_1 = \dfrac{KC_2}{2(1+k_{33})} \quad ; \quad r_{31} = \dfrac{Km_3}{2\bar{r}^2(1+k_{55})}$$

$$r_{21} = \dfrac{1+k_{11}}{1+k_{33}} \quad ; \quad s_{31} = \dfrac{K\bar{x}_t \, C_T}{2\bar{r}^2(1+k_{55})} \qquad (3)$$

$$s_{21} = \dfrac{KC_T}{2(1 + k_{33})}$$

$$q = q_{21}r_{31} - q_{31} r_{21}$$

$\bar{s} = q_{21}r_{31} - q_{31}s_{21}$

$s = r_{21}s_{31} - r_{31}s_{21}$

$\bar{x}_\beta = \dfrac{m_1\beta}{C_1\beta + C_2|\beta|\beta}$

$\bar{x}_\beta' = \dfrac{m_1}{C_1 + 2C_2|\beta|}$

$\bar{x}_{p\phi} = \dfrac{Km_3p\phi}{2(1+k_{11})p\phi}$

where $\bar{x}_\beta$ Position of the relative arm of the hydrodynamic force (center of pressure)

$\bar{x}_\beta'$ Ratio of derivative of drift angle position moment to position hydrodynamic force

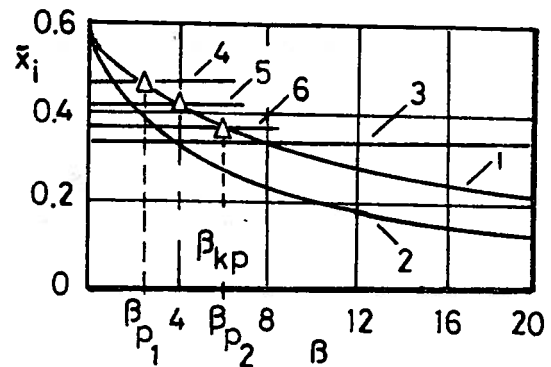$\bar{x}_{p\phi}$ Relative arm of damping force.

FIG. 2 Characteristic Relation of Hydrodynamic Arm with Drift Angle

Key: 1. $\bar{x}_\beta$, 2. $\bar{x}_\beta'$, 3. $\bar{x}_{p\phi}$

4. $\bar{x}_t$ corresponding to unstable equilibrium

5. $\bar{x}_t = \bar{x}_{t2}$

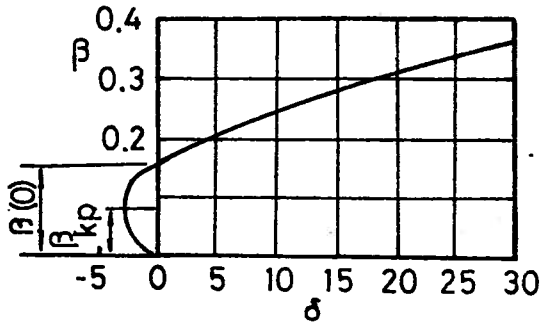6. $\bar{x}_\beta'$ corresponding to stable equilibrium

Fig.3 Dependance of Ship Drift Angle in Steady Turn by Moving the Rudder

$$\beta(0) = -\frac{9}{n_1 r_{31}} \quad ; \quad \beta_{kp} = -\frac{q}{2h_1 r_{31}}$$

The characteristic dependence of the values of X_β, $\bar{X}_\beta'$ and $\bar{X}_{p\phi}$ on the drift angle for an unstable ship·is illustrated in Fig.2. The maneuvering diagram characteristic for this example ship is presented in Fig.3.

The initial consideration is given to the possible equilibrium state of the system described by·equation (2). Assuming that p=0 and designating the equilibrium values of β as β_p, we obtain the following equation for determining the equilibrium from the basic equations (2), (3) and (4)

$$(\bar{s} + s_{31}h_1 \beta_p)\beta_p = 0$$

Here two cases are possible (see Fig.4 and curve 1 in Fig.5)

1) $\bar{s} > 0$; $\beta_p = 0$

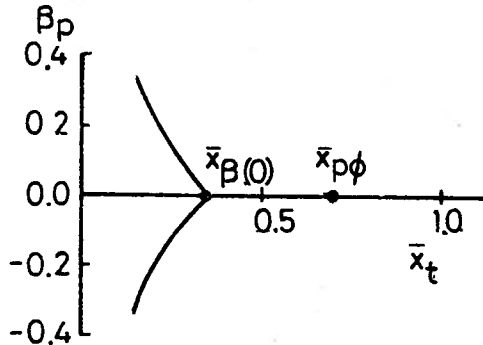2) $\bar{s} < 0$; $(\bar{s} + \bar{s}_{31}h_1\beta_p) = 0$; $\beta_p = -\frac{\bar{s}}{s_{31}h_1}$

Fig.4 Dependence of the drift angle in equilibrium position on the location of towing cable fastening point for a nonasymptotically stable ship.
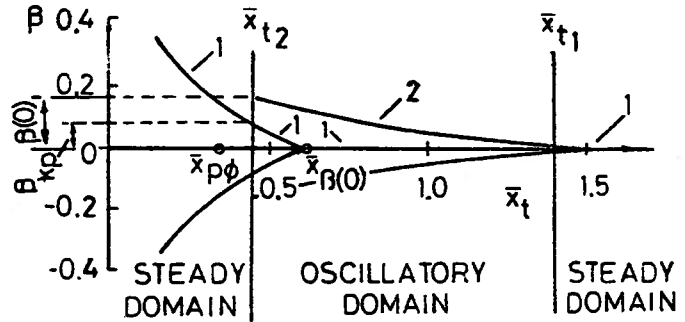

Fig.5 Possible drift angle values for towing unstable ship

1 β_p Characteristic equilibrium position

2 β_o Amplitude of oscillation

The first case corresponds to the conditions $\bar{X}_{,t} > \bar{X}_\beta(0)$. This means that the cable fastening point is displaced towards the bow from the center of pressure acting on the hull at zero drift angle. The second case corresponds to the conditions of $\bar{X}_t < X_\beta(0)$. In this case $\bar{X}_\beta(\beta_p) = \bar{X}_t$, that is the center of pressure for $\beta = \beta_p$ coincides with the $\bar{X}_t$ abscissa·

Returning now to the analysis of stability we consider the equilibrium position and use the equation of motions describing a disturbance relative to the equilibrium position.

Analysis indicates that after the disturbance the motion will be stable or unstable depending firstly if:

$$\bar{X}_{p\phi} + \bar{X}_t \frac{s_{21}}{r_{21}(q_{21} + 2h_1\beta_p)} - \bar{X}_\beta'(\beta_p) > 0 \qquad (7)$$

To find the value of the arm $\bar{X}_t$, which enables the inequality (7) to be transformed to the equation:

$$\bar{X}_{p\phi} + \bar{X}_t \frac{s_{21}}{r_{21}(q_{21} + 2h_1\beta_p)} = \bar{X}_\beta'(\beta_p) \qquad (8)$$

Here again we have two cases:

$$\beta_p = 0 \qquad (9)$$

$$\beta_p = -\frac{\bar{s}}{s_{31}h_1} = -\frac{q_{21}}{\bar{X}_t h_1}[\bar{X}_t - \bar{X}_\beta(0)]$$

-49-

In the first case, taking into account that $\bar{x}_\beta'(0) = \bar{x}_\beta(0)$ results in

$$\bar{x}_t = \frac{r_{21}q_{21}}{s_{21}} [\bar{x}_\beta(0) - \bar{x}_{\rho\phi}]$$

and for the second case:

$$\bar{x}_t = \frac{2\bar{x}_\beta(0)\bar{x}_{\rho\phi}}{\bar{x}_\beta(0) + \bar{x}_{\rho\phi}}$$

In this manner, it can be confirmed that the necessary conditions for stable motion occurs when:

$$\bar{x}_t < \bar{x}_{t_2}$$

$$\bar{x}_t < \bar{x}_{t_1}$$

and that in the range $\bar{x}_{t_1} < \bar{x}_t \leq \bar{x}_{t_2}$

the disturbing motion will become unstable (lead to unstable motion).

In the latter case, the parameter β is nonlinear due to the oscillation. Assuming that in the state of oscillation, $\beta = \beta_o \sin \bar{v}t$ and adopting harmonic linearization of the nonlinear component $h_1|\beta|\beta$ results in:

$$h_1|\beta|\beta \cong \frac{8}{3\pi} h_1\beta_o\beta = h_1^*\beta_o\beta \qquad (10)$$

Taking into account equations (3),(4) and (10), the system of equations (2) can be reduced to the following differential equation relative to the variable β:

$$\left\{ p^4 + (r_{31}+q_{21}+h_1^*\beta_o)p^3 + \left[q+r_{31}h_1^*\beta_o + s_{31}+ \frac{1}{\ell}(s_{21}+\bar{x}\,s_{31})\right]p^2 + \left[(1+\frac{x_t}{\ell})\bar{s} + s_{31}h_1^*\beta_o + \frac{1}{\ell}(s_{31}-s)\right]p + \frac{1}{\ell}(\bar{s}+s_{31}h_1^*\beta_o) \right\} \beta = 0$$

$$(11)$$

The characteristic equation corresponding to expression (11) has the form

$$p^4 + a_1p^3 + a_2p^2 + a_3p + a_4 = 0 \qquad (12)$$

where the values of the coefficients a_1, a_2, a_3 and a_4 are clear from comparing

equations (11) and (12). Assuming $p=i\bar{v}$ and substituting it into equations (12) results in an equation for determining the amplitude β_o and frequency $\bar{v}$.

Numerical analysis indicates that for cable lengths $\ell>3$ and values of $x_t< 0.5$, the terms containing the factor $1/\ell$ become insignificant. This permits investigating a simplified equation instead of equation (11) and the solution gives the following results.

$$\beta_o = -\frac{q}{h_1^* r_{31}}$$

If we compare this result with the derivation in reference [7] of the formula for determining the drift angle of a steady turning ship with zero rudder angle (see Fig.3), it has the form

$$\beta(0) = \frac{-q}{h_1 r_{31}}$$

It is easily observed that the value of the oscillation amplitude β_o differs from the value of the drift angle in steady turning $\beta(0)$ by the linear harmonic coefficient term h_1^*. Otherwise the amplitude β_o for towing an unstable ship for values $\bar{x}_t < 0.5$ are nearly the same as the value of drift angle of the ship (considered as self-propelled) in a steady turn for zero rudder angle (see Fig.5).

Having determined the amplitude, β_o it is easy to estimate the frequency
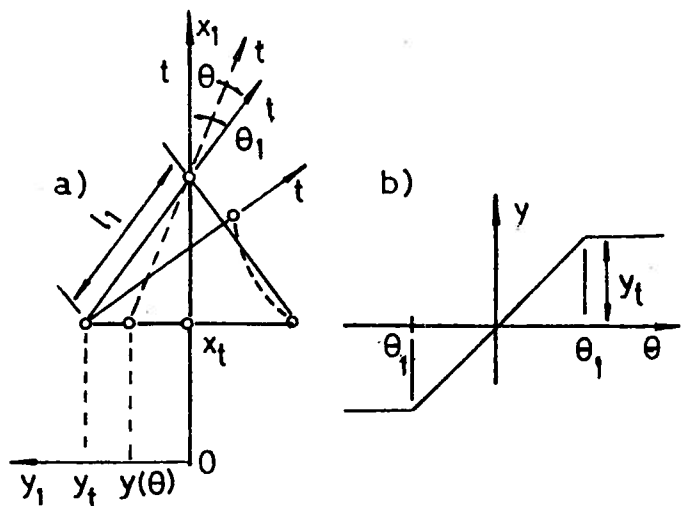
$$\bar{v} = v\frac{v}{L}$$

Figure 6 Diagram of harnessing bridle in towing (a) and dependence of ordinate Y on angle v

Here
$$\vartheta = \frac{v}{L}\sqrt{\frac{\bar{s} + \bar{s}_{31}h^{*}_{1}\beta_o}{r_{31} + q_{21} + h^{*}_{1}\beta_o}}$$

This produced the systematic results in Figs. 4 and 5 which give the values for the drift angle β_p corresponding to states of stable and unstable equilibrium. These figures also give the amplitude of the drift angle in oscillatory behavior for the point where the towing cable is fastened for stable $(\bar{x}_{p\phi} > \bar{X}_{\beta}(0)$ and unstable $(\bar{x}_{p\phi} < \bar{x}_{\beta}(0))$ ships.

In conclusion it should be noted that it is perfectly fine to use a bridle during towing. The difference is that instead of the value of $\bar{X}_t$ the value of $\bar{X}^{*}_t$ is used when the towing bridle is adopted. It is defined as follows:

$$\bar{X}^{*}_t = \bar{X}_t + \ell_1$$

where ℓ_1 is related to the length of one leg of the towing bridle (Fig. 6).

To understand this relation consider the moment M_T and tow force T from the towing bridle relative to the ship's center of gravity.

$$M_t = X_t T\sin\theta + yT\cos\theta = T(X_t\theta + y)$$

For $y = y(\theta)$ where $y(\theta)$ is a nonlinear expression (Fig. 6-b)

In the limit $|\theta| < |\theta_1|y \doteq \ell_1\theta$, this results in the following expression:

$$M_t = T(X_t + \ell_1)\theta = T \cdot X^{*}_t\theta$$

This corresponds to the case where the towrope attachment point is shifted forward along the vessel's centerplane.

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