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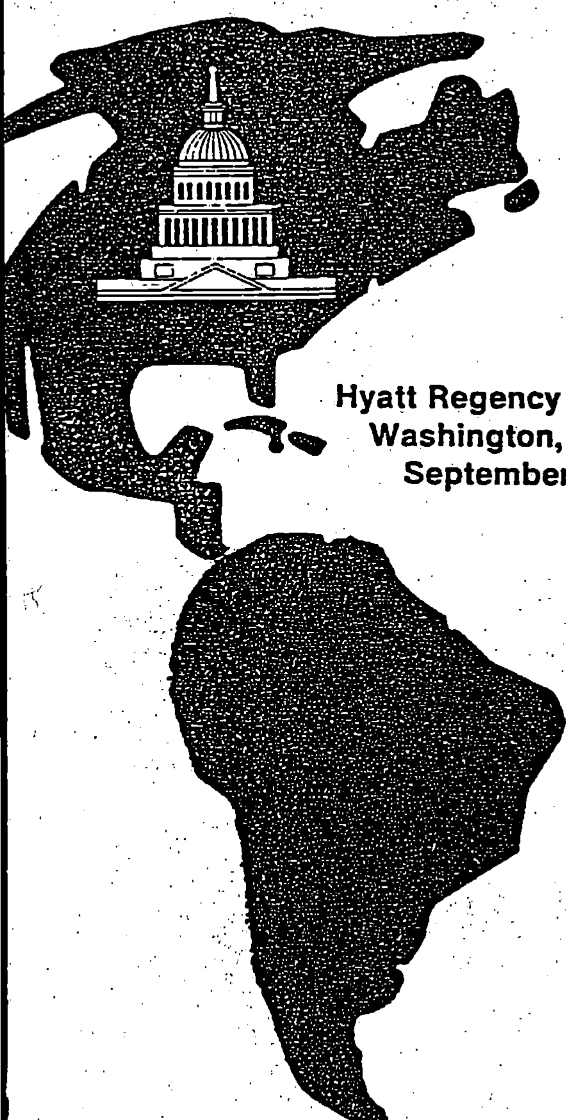

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INLAND VESSELS - GENERAL CONSIDERATIONS AND RECENT DEVELOPMENTS RELATED TO THE DESIGN AND OPERATION

by

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INTRODUCTION

The drastic increase in oil price has led to an urgent need for fuel economy and inland shipping, while being highly competitive mode of transportation, has not been spared this impact. This applies alike to countries with traditionally high share of transportation by waterways in movement of goods and those in the process of modernizing or expanding such systems, which characteristically fall under two main catagories, viz. self-driven vessels and pushed barge tows.

Fuel economy as such was recognized even long before the oil price escalation as an important design factor and has been one of the primary objectives underlying the work of the VBD, i. e. the Duisburg Research Laboratory for Inland Shipbuilding and Shallow Water Hydrodynamics in development and implementation of design of inland vessels and improvement in their performance. Thus with the closely co-ordinated effort between research laboratory, waterway authorities, ship builders and shipping companies the inland fleet on German and West European waterways with their high traffic density and rigid operational demands is today not only matured but also has the necessary prerequisites to cope with the situation. However, with more stringent emphasis on energy saving, endeavours are being made to reduce fuel consumption further by improving the design with respect to resistance and propulsive efficiency; widespread standardization; and cost-oriented operations.

For bulk commodities the classical pushed tows of tug and pushed barges offer a number of benefits. Large convoys are superior in resistance, particularly when the barge size and form are standardized, as is the case in Western Europe, because the unit resistance of such tows of barges whose principal particulars including draft do not differ considerably is decidedly lower than that of a single barge. Scope for improvements here is partially discussed in refs. [1,2], whereas improvements in propulsive efficiency over and above the present status may involve more sophisticated changes. Some aspects of propulsion in pusher tug are dealt with by another paper in this session.

Parallel to the pushed tows the role of the self-driven vessel or GMS (Goods Motor Ship), particularly for general and liquid cargoes as also for varying freight market remains equally important, although container and other special inland vessels following the pattern known from sea ships are not seldom any more. With emphasis on economy the trend on West European waterways is towards larger GMS with capacity to pushtow upto three standard barges in pushed convoy thus giving it the same flexibility as that of a classical tug tow. A similar dual concept with its inherent scope for change over to pushtug tows at any stage of growth is apparantly also a suitable alternative for other areas with variable cargoes or expanding inland transportation.

The paper is mainly directed towards this development, but most of the considerations put forward here also apply to inland vessels in general.

SIZE CONSIDERATIONS

The economy of size rests on several factors. Most of these are known from sea ships, where increasing size for example brings many benefits, particulary in resistance and powering area. While the latter is also true for inland vessels, their size is restricted by the given physical conditions in way of waterdepth, channel width, radius of bends and size of locks. Among these little is known on the relationship between length and beam of a vessel or a convoy on one side and the required channel width to negotiate a bend of given radius on the other side. The width of path is considerably bigger than the ships beam because of the side-slip or angle of drift between the heading angle and tangent to the actual path in a turn, which is influenced, among others by the ships length, speed, waterdepth and radius of turning circle or bend.

Fig. 1 shows the width of path as a function of the radius of mid-channel or path in bends for a single vessel or a single row pushed tow loaded to a draft of 2,5 m in waterdepths corresponding to 5 to 7 m. The curves shown here are for a ship speed of 14 km/h through the water in zero and two further current velocities both in upstream and downstream runs. These have been derived from full scale measurements on a large number of vessels of different size and in different formations. For evaluation of results it was necessary to convert them to correspond to a common size. The curves presented here apply to a standard length of 185 m as is permitted on the Rhine, and to zero beam. To calculate the actual width of path Bp the breadth of the vessel or tow has to be added to the plotted Bp* value.

Fig. 2 shows similar universal curves for twin row tows of standard length of 185 m. for the conversion to standard length in both cases the tactical turning centre of ship has been assumed to be located at one-third of length from forward perpendicular.

Fig. 3 shows on the left the geometrical definitions used in the conversion, which can be applied to calculate the width of path from these plots for other lengths. Equally the maximum feasible lengths of various units for given channel width for minimizing resistance can be calculated. The diagram on the right shows a schematic diagram to estimate the required total channel width in a bend for two lane traffic.

SHIP FORM

The draft restriction imposed by shallow depth and the desire to keep the size as big as possible lead to big $L/_B$ and $B/_T$ ratios. With the long parallel middle body a relatively high block coefficient can be realized. But here too there are limits arising from the need to keep both ends fine for lower resistance, but above all for more free access for the water entering the propeller. Before coming to the design aspects of bow and stern and to the current trend a brief review of the past development may illustrate some considerations influencing the design of inland vessels.

From among a large variety of ships with respect to size, hull form and power during the rebuilding phase of German inland fleet since after the second world war 5 types as recommended for standardization by the Central Association of German Inland Shipping are dominantly representative [4]. These differ in size to keep them compatible with the size-oriented classification of waterways. However, except for the two bigger types i. e.

"Gustav Koenigs" LxBxT = 67x8.2x2.5 m, 940 tdw
and
"Johann Welker" LxBxT = 80x9.5x2.5 m, 1290 tdw,

which also found widespread employment on the Rhine, the others have not been built in any large number. The next step towards realization of an efficient fleet was the introduction of the so-called "Europa"-size-GMS classified as "Otto Most" with LxBxT = 85x9.5x2.8 m and about 1500 tdw. It differs from "Johann Welker" by having simplified form with conventional tunnel stern, but pontoon type fore body [5].

Fig. 4 shows a comparison of body plan of both last-mentioned vessels. Both are designed as single-screw vessels with either an open propeller or a nozzle arrangement, although a few twin-screw versions were also built. As a single-screw vessel for single unit operation the "Europa" class GMS has an average engine rating of between 900 to 1000 hp. This also represents approximately the upper limit of power for single-screw vessels without undue high risk of cavitation and excitation of vibrations. Moreover the increased thrust loading above this figure leads to a poorer propulsive efficiency.

The conclusion from the foregoing making it obvious to take as big a propeller diameter as possible in order to keep the thrust loading low cannot be put to pratice because of the fact that the propeller in tunnel should not emerge by more than 30 % of its diameter above the waterline at ballast draft, so that it can draw water adequately, especially when reversing.

Moreover for inland vessels whose length exceeds 86 m good stopping quality in downstream runs is a prerequisite for license from the authorities in Germany. This requirement additionally limits the propeller diameter because the reversed propeller can draw water from the stern only when the tunnel is hooked back to end below the waterline at the transom. Otherwise the propeller will draw air and become less effective in delivering the needed backward thrust.

Thus the upper limit of a propeller in an inland vessel on the Rhine (excepting tugs and push-boats) can be taken to be as approximately 1.7 to 1.8 m. Larger diameters can be realized only if a bigger standard ballasting is tolerated. The respective value for maximum power transmitted to the shaft amounts to about 1000 hp for a nozzle-less and about 1200 hp for a nozzle propeller.

In practice higher rated engines have often been installed but they show that they invariably lead to the accompanying disadvantages in way of cavitation, propeller excited vibrations in after ship and a drop in propeller efficiency or even in ships speed. Although it is possible to combat these effects for crews comfort by mounting the deck-houses on elastic bedding, primary aim should be to eliminate the cause as far as possible by adequate after body design and by keeping the propeller thrust loading within limits.

Under these circumstances it is understandable that the change-over to large GMS is based on twin screw version. The bigger beam in the large GMS is also an argument in favour of the twin screw arrangement. Moreover the benefits in improved manoeuvrability and with that in traffic safety speak for themselves.

The principal particulars and the installed engine rating for operation on the Rhine are as a rule dimensioned such that it is possible to run them as single units or in convoy with one, and lately, even with three barges. Since the permissible length of convoys on the Rhine generally amounts to 185 m - in the Dutch sector it is even strictly restricted to this value -, and the length of the standard "EUROPA II" barge with 11.4 m beam is fixed at 76.5 m, the upper limit for the size of large GMS currently results to be

$LxB = 108.5 \times 11.4$ m

At a draft of 3.0 m such a ship has a carrying capacity of about 2500 tonnes of dry bulk cargo, which increases to about 4450 tonnes for a convoy with one standard barge at same draft and to 8300 tonnes with three such barges.

Fig. 5 shows the body plan of one typical example of such a vessel. The installed engine rating amounts approximately to 2 x 840 hp. However, this type is not yet representative of large GMS, which includes all vessels upwards of 90 m length, and various units deviating from this size and form are currently in operation. At the same time further development, particularly of the after body is in progress. Whereas improvements in bow and stern design as also in propeller arrangement have been made and may still be possible, it appears that in river transport only very little further gains with respect to unit resistance per tonne displacement can be obtained by increasing the length much beyond some 125 m, i. e. $L/_{B} > 11$. At the same time bigger lengths lead to rapidly increasing steel weight because of strength reason so that the economic benefit in terms of carrying capacity is still smaller.

BOW FORM

The bow form has been the subject of extended investigations aimed at modifications at the VBD. Starting with the conventional bow with ship-shaped sections in standard vessels on the one hand and pontoon bow of simplified design in barges on the other hand, many further derivatives have been developed and tested for both types of vessels to meet the varying functional demands such as their operation as single units or as coupled tows and sea-kindliness in slight to moderate seas in coastal and estuarial waters.

Basically it can be said that for a vessel predominantly plying as a single unit, the ship-shaped bow is better suited, the more so the more it approaches an ellipsoid shape. For vessels engaged in push towing as also for barges which are mostly coupled together, the pontoon type of bow is in general adequate, although it has scope for improvement. If the latter is retained, it is essential that the flatplate stem rising at about 30^{o} inclination does not end at the bottom with a hard knuckle but is faired into the same with a large transitional radius. Similarly it is important that the bilge in this area has a sloped cut away so that the water meeting the bow can go round the side.

Elaborating on these remarks it may be mentioned that with elliptical type of bow in single unit operation about 10 to 15 % reduction in resistance and propulsive power as compared to other forms can be obtained. Fig. 6 shows such a bow. This benefit, however, reduces when such vessels are coupled together, especially in abreast formation because of the open wedge piece formed between the bows. Moreover the building costs are disproportionately high because of the form being relatively complicated, made more so due to the sophisticated construction necessary for fitting pushing knees.

On the other hand the pontoon type bow in a vessel or barge has the disadvantage that because of a big beam, becoming even bigger when two units are coupled together, most of the water meeting the bow connot go around the sides and must flow under the bottom. Shallow water conditions force a part of the flow to the sides but also accelerate the underflow generating a high bow wave and causing the forward unit or units in convoy to trim excessively by the head. Thus with diminishing bed clearance not only the resistance increases rapidly but also the risk of bows becoming awash or hitting bottom grows seriously.

An alteration to the pontoon or wedge bow giving it V-section shape and a parabolic run of stem has been found to be effective in reducing resistance while retaining or even overbidding other prerequisites in the way of simple construction and deadweight capacity, the latter being about 30 to 50 tonnes bigger depending upon the draft. Significant is that the resistance in spite of bigger displacement is not only lower but also decreases substantially with decreasing water depth.

Fig. 7 shows the V-section bow along with the lines of the after body discussed under the next heading.

STERN FORM

The stern form and propulsion of a ship designed for service on inland waterways presents special problems because of water depth limiting the draft. Due to shallow draft it is not possible to increase the size of propellers in step with the increasing size of the vessels, even when these are arranged in sophisticated tunnels. This leads, particularly in the case of larger vessels and those engaged in push-towing, to the difficulty that the required high propulsive power has to be delivered at relatively small propellers operating under unfavourable flow conditions. In turn the propeller thrust loading is extremely high with the disadvantage of lower propulsive efficiency and increased danger of cavitation and vibration excitation. The problem is aggravated further by fluctuating depth and width of waterway and variable speed of ship due to changing operating conditions.

The propulsive efficiency, as is generally known, can be positively influenced primarily by two factors, viz. by keeping the propeller disc area as big and with that the thrust loading as small as possible and by endeavouring to achieve a uniform and undisturbed flow to the propellers, which in turn has beneficial effects on thrust deduction and wake fraction and with that on hull efficiency.

The latter can only be realized by a careful design with due regard to the size and arrangement of propellers and steering devices. Generally speaking, it is important that the rising slope of tunnel top towards the propeller is kept within 15 to 20 degrees and its transition to the flat bottom as also to the highest point above or preferably in front of the propeller is faired with a big radius of curvature. Since suction of air must be prevented both while running ahead and while reversing for stopping or going astern, the tunnel top must be hooked back towards the sides and behind the propeller to keep the lower edge of side skirts and transom underneath the waterline.

Based on test data supported by full scale investigations general guide lines for the design of tunnel have been compiled and are available at the VBD. With tunnel top above the waterline for accommodating big diameter the propeller as a special type of axial pump must have a minimum immersion in order that it can pump the tunnel full and keep it that way.

Fig. 8 shows the tunnel geometry illustrating the above. Whereas this sets a limit on the propeller diameter, its size in turn determines the maximum power which can be delivered to it without exceeding the acceptable thrust loading.

For heavily loaded propellers working under such poorer suction conditions thrust loading as measure for dimensioning the propeller can be substituted by specific power loading as quotient of delivered power to propeller disc area.

For relatively slow large GMS following upper limits appear to be acceptable:

Thrust loading coeff. $C_{Th\ max} = 35$

specific power loading $q_{PA\ max} = 380\ hp/m^2$ for propellers without nozzles

and $q_{PA\ max} = 460\ hp/m^2$ for propellers in nozzles.

Combining both the criteria one can determine the design data for propeller diameter and whether twin or multiple screw arrangement is necessary.

Fig. 9 shows a worked out example, which illustrates that for the given engine rating and ballast draft single screw arrangement is not suitable. Such calculations serve in the first instance to obtain first-hand estimates and naturally cannot substitute the usual calculations to check cavitation risk etc. in propeller design.

Considera~~l~~ble advantages can be obtained by fitting heavily loaded propellers in nozzles. With the same total thrust from the system propeller plus nozzle in such cases the required power can be reduced significantly against that of non-ducted propeller and the thrust loading favourably influenced. Further advantages are that the nozzle aids in making the wake more uniform, whereas the flow entering the propeller is less sensitive to fluctuations in thrust loading arising from variations in ship's speed.

Fig. 10 shows the attainable saving in power from using nozzles in the case of a large GMS as a function of depth to keel clearance ratio. The advantage of nozzle grows continuously with increasing thrust loading, i. e. with increasing power or with decreasing keel clearance.

Another feature not to be neglected in GMS is that the breadth at the transom should be reduced to avoid undue separation and eddy building, which otherwise can be expected to occur with certainty because of the beam being much bigger than the span of the propellers. It is better to draw in the waterlines towards the stern gradually without forming a pronounced knuckle at the sides. Some advantage here can also be gained by arranging the shafts closer together or at a slight convergence towards aft.

Fig. 11 shows the propulsion data from model tests of three different designs of large GMS as presented here in Figs. 5 to 7. The comparison shows considerable reduction in propulsive power for the design with V-section bow and narrow transom. About 10 - 12 % power gain is due to the change over from pontoon to V-section bow and another 14 - 15 % power reduction is due to the modification in stern design.

BOW THRUSTER

In larger motor vessels bow thrusters invariably need to be installed for manoeuvrability reasons, particularly in waterways with high traffic density as on the Rhine. They represent a significant aid to steering while running at ballast in strong side winds but at service speed these are in general scarcely used for transverse force and can be advantageously employed to generate a part of the forward thrust for propulsion thus reducing the thrust loading of the after main propellers.

There are many types available and if one has to be installed, the alternative having the possibility to give side or forward thrust as required, might be a better solution unless other considerations overweigh. Diverting about 25 to 30 % of propulsive power to a hull adopted bow thruster and about 70 to 75 % to the main propulsive unit can lead to a considerable reduction in total power especially when the main propulsive unit is working under heavy thrust load condition.

APPENDAGES

Appendages like skegs, bossings, shaft brackets, etc. tend to have a bigger significance on shallower depths than in deep water. In twin or triple screw vessels it is better to avoid bossings or skegs in shaft line of side or wing propellers. In shallower depths the propellers draw water more and more from the sides. Such bossings or skegs interfere with the flow entering the propeller and thus cause a higher thrust loss. Skegs, if needed, should preferably be fitted on the centre line.

Similarly "I" brackets are usually better. For "V" and "Y" brackets care must be taken that the opening angle corresponds to the number of blades of propeller. In any case they should be kept in streamlined form as far as possible.

If flanking rudders are necessary as steering aid for stopping or going astern, their disadvantage with respect to forward propulsion can be kept to a minimum when these have a short blade length and when the after end of the rudder blades does not extend into the nozzle opening. The optimum zero-position adjustment of the flanking rudders for forward performance should be carefully fixed and can lead to a main engine power saving of up to 5 % [8].

ECONOMICS OF SPEED

As is known from sea ships, determining economic speed is rather intricate and involves various factors such as fuel costs on the one hand and crew costs, building costs, rates of interest, insurance, management etc. on the other hand. If the former increases the economic speed will be lower than the starting point and vice versa, if the latter costs increase.

Generally a reduction in speed brings an immediate and proportionately large reduction in required power because of the sharply rising curve of power versus speed. In inland vessels this curve is rather steep and becomes more so the shallower the depth is. A good measure for determining a limit here is the Froude depth number, which relates the ships speed to that of the wave of translation on a given depth.

Although sufficient performance data on large GMS in convoy with one or three barges is not yet available, such formations are essentially comparable with classical tows of tug and standard pushed barges, whose resistance and power curves for various water depths are known. The VBD has evaluated these for suitable range of Froude depth number corresponding to speeds, which may be termed as economical under hydrodynamic aspects. In most cases these must be regarded in conjunction with other relevant considerations such as current velocities, operational requirements, user needs and flow speed of goods on the whole.

Another drawback is that these economic speeds are relative to water and in rivers with currents their continuous surveillance is not easy. However, modern equipment, as for example Doppler effect speed logs and bottom scans are becoming available and permit relatively inexpensive monitoring of speed both through the water and over the ground as also the water depth. Thus with a board computer the reference speed from Froude depth number for given water depth could be instantly calculated and compared with the speed relative to water and over ground - the latter as a further key value in order that the shipmaster can run the ship more rationally while adhering to a given time-table. Another aid here can be the continuous monitoring of fuel consumption. Flow meters for this purpose are being developed and tested.

SUMMARY

The subject of fuel economy has attained a position of major importance, and as elsewhere, endeavours are being made to reduce fuel consumption in inland water transportation, too.

The aim of this paper is to show that there are a number of possible economy measures available within the design aspects of inland vessels, which can lead to remarkable success.

The treatment of the subject is confined to self-driven vessels operating as single units or in convoy with barges. Classical pushed tows of tug and barges have not been included in the considerations reviewed here.

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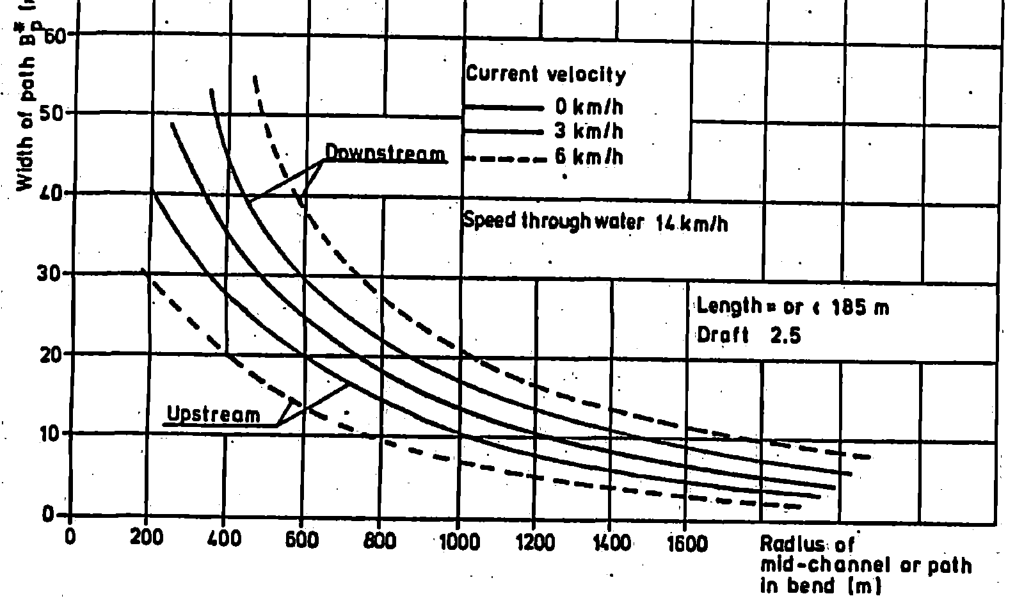


FIG. 1
UNIVERSAL CURVES FOR DETERMINATION OF WIDTH OF PATH
SINGLE VESSELS AND SINGLE ROW TOWS.

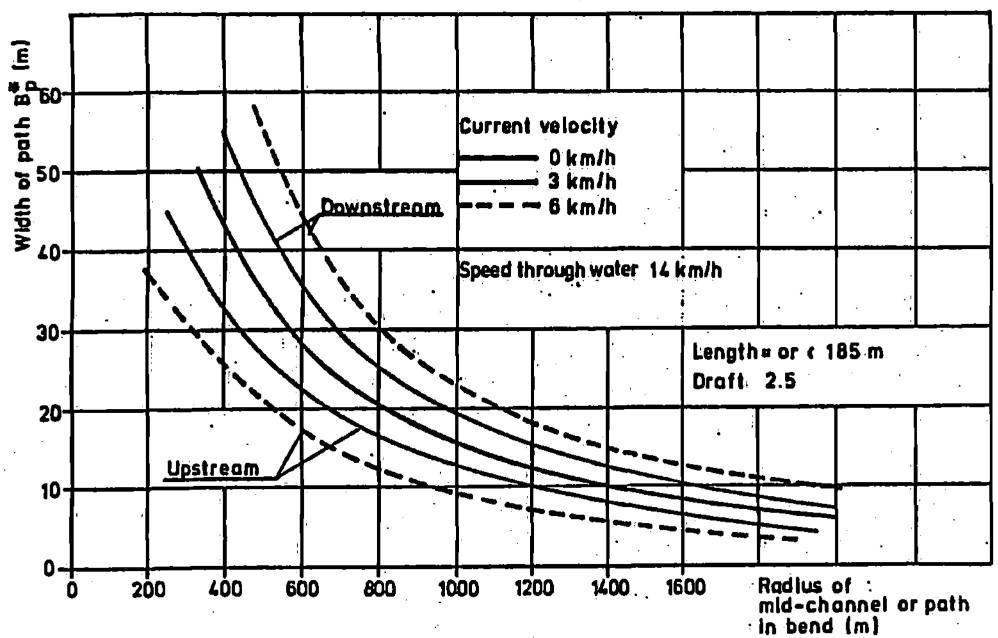
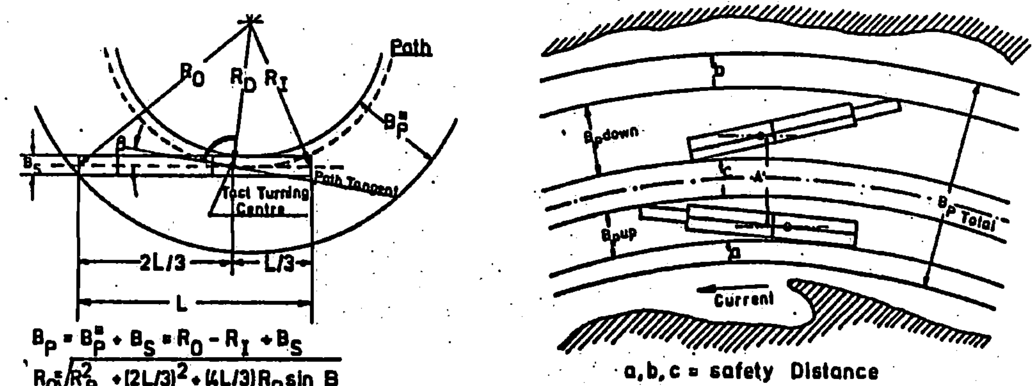


FIG. 2
UNIVERSAL CURVES FOR DETERMINATION OF WIDTH OF PATH
DOUBLE ROW TOWS



$$B_P = B_P^* + B_S = R_O - R_I + B_S$$

$$R_O = \sqrt{R_P^2 + (2L/3)^2 + (4L/3)R_P \sin\beta}$$

$$R_I = \sqrt{R_P^2 + (L/3)^2 - (2L/3)R_P \sin\beta}$$

FIG. 3
GEOMETRICAL DEFINITION OF WIDTH OF PATH

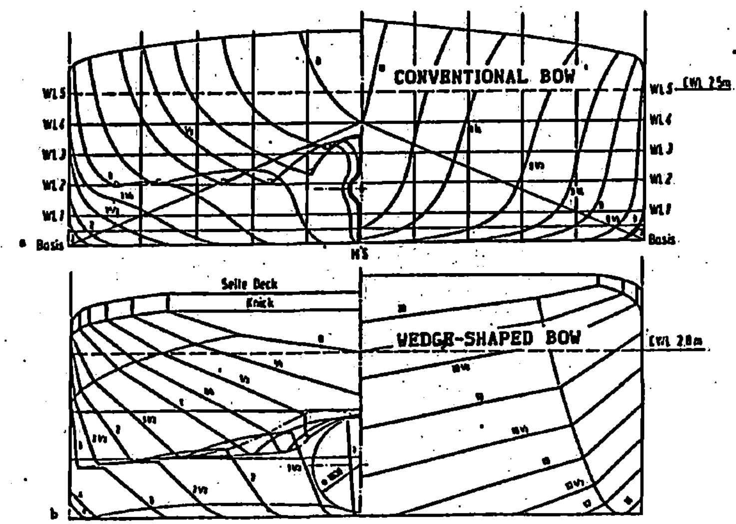


FIG.4 TYPE MOTOR VESSEL
a) JOHANN WELKER 80x9.5m, 1290 tdw
b) EUROPA-Size 85x9.5m, 1500 tdw

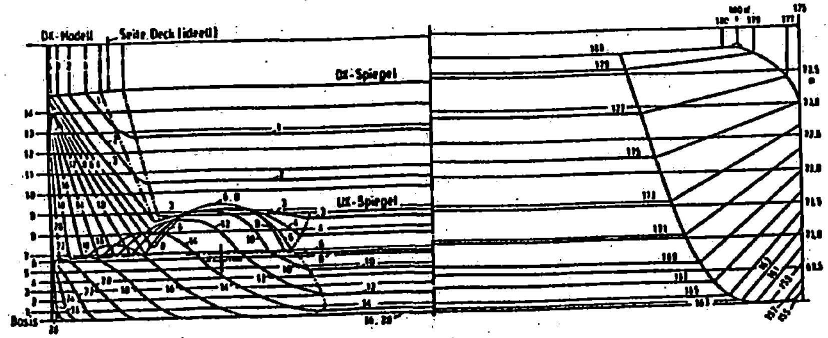


FIG.5 TWIN-SCREW MOTOR VESSEL

LxBxT = 108.5x11.4x3.3m, 2900 tdw

PONTOON BOW

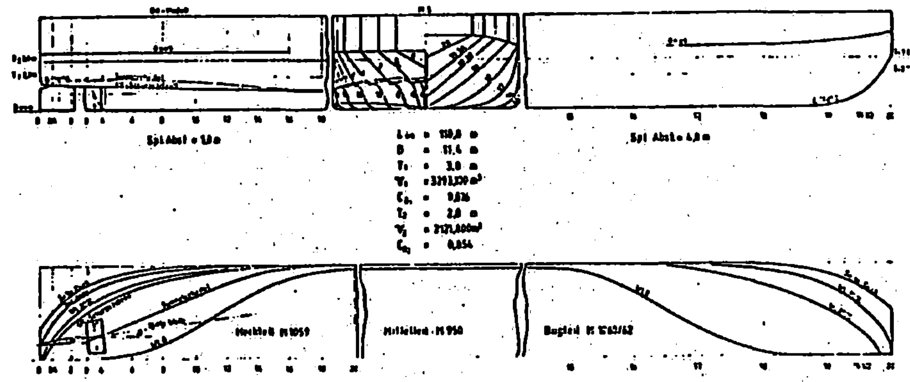


FIG.7 TWIN-SCREW MOTOR VESSEL

V-SECTION BOW

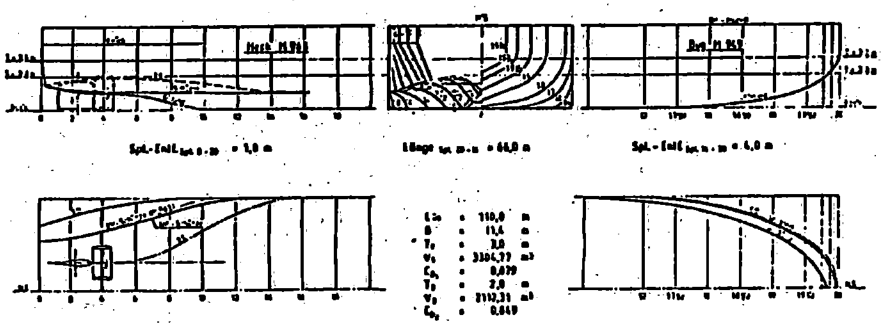
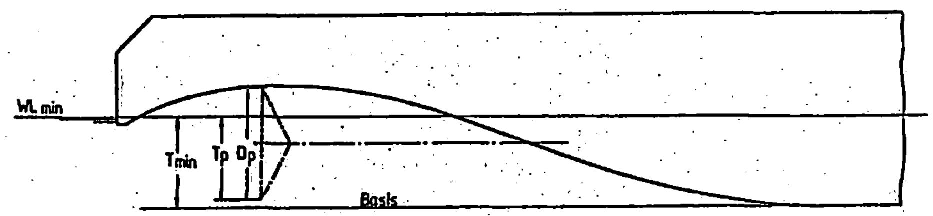


FIG.6 TWIN-SCREW MOTOR VESSEL

ELLIPTICAL BOW



PROPELLER WITHOUT NOZZLE TP/DP =or> 0.60

PROPELLER WITH NOZZLE TP/DP =or> 0.70

FIG.8 TUNNEL GEOMETRY

PROPELLER WITHOUT NOZZLE IN TUNNEL

Ballast draft at propeller location	T min	= 1.20m
Propeller immersion thereby	TP	= 1.15m
Engine rating	PB	= 1800 hp
Delivered shaft power	PD	= 1700 hp

1) Max. propeller diameter considering tunnel geometry (TP/DP =or> 0.60) — DPmax = 1.9m

2) Min. propeller diameter considering acceptable power loading PD/AO=380 hp/m*2

a) as single-screw vessel — DPmin = 2.38m

b) as twin-screw vessel — DPmin = 1.70m

RESULT: Projected ship cannot be built as single-screw vessel.
Minimum number of propellers : two

FIG.9

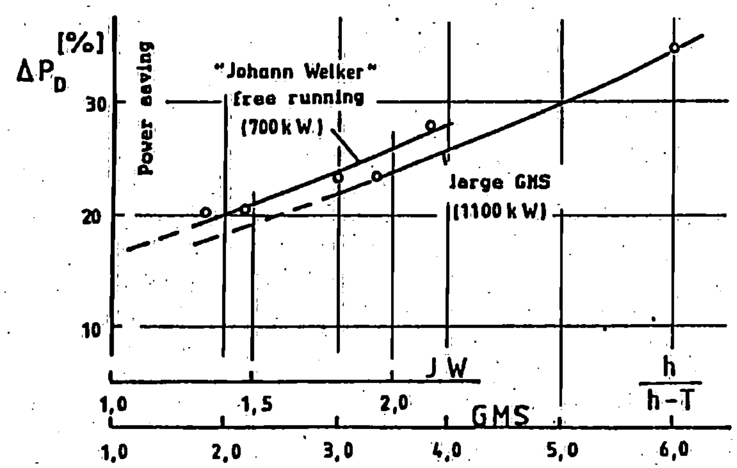


FIG.10 POWER REDUCTION FROM FITTING NOZZLES
a) JOHANN WELKER
b) LARGE GMS

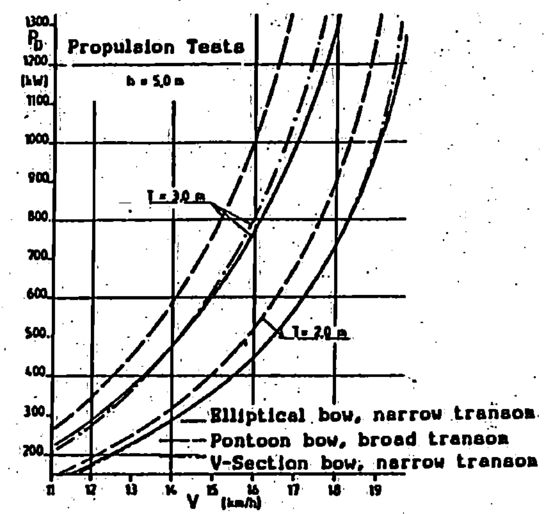


FIG.11 PROPULSION
TWIN SCREW LARGE GMS