Conference No. 84910
Time: 4 6 May 1988
Venue: t lotel Caledonien, Kristiansand

High-Speed Marine Craft
Technology update and market potentials

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HIGH SPEED MARINE CRAFT

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K JÆR THOMAS	NAVAL ARCH I TFCT	BATSERVICE VERFT A/S'	4500	MANDAL
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HIGH SPEED MARINE CRAFT

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NORWEGIAN SOCIETY OF CHARTERED ENGINEERS

Conference on High Speed Marine Craft

4-6 May 1988

Market Needs and Potential for High Speed Marine Transport

E.G. Tattersall
Director & General Manager
Hovermarine International Ltd.

Date: 23rd April, 1988.

Introduction

Nobody can claim a sole or unique formula to present or predict markets or potentials in the high speed craft area although there are a number of sound principles which cover this important matter. However there are so many exceptions to the rule - so many one off situations - that it is by some luck as well as a lot of judgement that successful market penetration and continuing market sustenance is brought about.

As an engineer I find this subject so inexact a science and suffering from a considerable amount of subjective preference that I always am trying to find ways to rationalise it in terms of at least 'cause and effect'. Right at the start of the Hovermarine Company we identified three major variables to consider for any one application and by at least being conscious of these we have exported over one hundred craft worldwide. These variables are common for all types of marine craft and can be universally used as test considerations for any new operational application.

What are these parameters? They are that the craft has to have the right combination of size, speed and cost and answer the three fundamental questions - HOW BIG? HOW FAST? HOW MUCH?

These are like three unknowns to an algebraic simultaneous equation and although some small latitude may be accepted around the optimal value of each parameter you cannot solve the equation by satisfying only one or two - all three have to be simultaneously compatible.

Also when discussing the market needs - who is the customer? As far as the designer/manufacturer is concerned the immediate customer is the operator but in turn the operator's market is the traffic and the individual passenger is his customer. The designer is therefore inexorably linked to the passenger and has to accept his requirements.

The Major Considerations

When considering any transport problem that is to identify and specify the most appropriate vehicle the stages in i s solution can be illustrated by Fig.1 and are the cardinal considerations in the process.

In the first instance there may be physical limitations or barriers that have to be considered e.g. it may be wholly appropriate that an amphibious capability might reduce the route length significantly or overcome icing problems. A reduced draught craft might have significant advantage over those that are relatively deep draught. From these initial considerations it may be possible then to define the most appropriate type or types of marine vehicle. Over water we have to assess the frequency of wind and wave conditions to allow a reliable service without undue cancellation because of passenger discomfort. These considerations will indicate that a certain physical size of craft is required and that it would be unwise to fall much below it.

Undoubtedly there arc ways in which the size effect can bc artificially augmented such as the very effective controlled foil system on the Boeing Jetfoil, roll stabilisers on various other craft, air pressure cycle attenuation systems on hovercraft and surface effect ships but these in the main can only improve ride quality within the craft's seacapacity. The seacapacity of the craft to counter the roughest conditions it is expected to face is still a f'urction of its physical size and leading dimensions. Power and system failures may occur under such severe conditions and you may be faced with a low speed platform where size and survival are directly linked.

Having determined the approximate size of craft appropriate to the route on the basis of safety and comfort when this size is combined with the traffic potential and service frequency this then should determine the required speed.

Too often speed is defined as a fixed parameter too early in the project stage and bearing in mind that fixed costs as well as running costs are speed dependent it can sometimes be an expensive decision.

Having determined the target speed the job should then be left to the designer and the manufacturer to produce the right size capable of performing at the right speed at an acceptable cost. What in fact happens of course is that the prospective customer/operator scans all available data on existing craft and hopefully finds one that most suits his needs. The problem for the designer/manufacturer is that when taking into account the considerable design and development costs in putting a new 'model' onto the market where can he concentrate his efforts to attract most sales?

Some Facts and Figures

Having established some basis of a logical process in order to choose the most appropriate high speed marine craft what can be deduced from the size of the parameters involved?

Looking at wind strength frequencies in various parts of the world throughout the year would suggest that in order to cater for a reasonable percentage of comfortable trips, say 75%, the craft design should be appropriate to wave conditions when at least a 20 Kt. wind is blowing. See Fig.2. If now we compare the average wave lengths likely on such ferry routes in a head sea condition and compare them with actual craft lengths employed we see from Fig.3 that a large proportion of the sample are more appropriate to operate in wind strengths of less than 30 Kts. I am assuming to the first order that the craft length should be at least larger than the wavelength in which it is operating - to minimise response the craft length probably should be more than 50% more than the wavelength. In such wave conditions of course the 'comfortable' craft should not experience undue impacting and should have the seacapacity to contain the corresponding waveheight. For coastal waters Darbyshire suggested that the maximum waveheight likely to be experienced in the fully developed sea would vary as in the following table:-

Wind Speed

Route Length Equal to Fetch n.m.	10 Kt.	20 Kt.	30 Kt
10	0.3m	1m	2m
20	0.6m	2m	2.8m
50	0.9m	2.6m	4.3m
100	1.1m	3m	5m

Therefore as an example if the craft was to be chosen to cater for say, a route length of typically 50 miles, in which it could operate reasonably comfortably for a large proportion of the time it should be at least 50 metres in length and have a hard structure clearance above the undisturbed water level of at least 2.6 metres. This would be the clearance of a hydrofoil craft hull above the water, the cushion depth of a hovercraft or S.E.S. or the wet deck clearance on the bridging structure of a catamaran.

In looking at the distribution of potential route application Fig.4 illustrates an analysis of a sample of world wide ferry routes (excluding river and sheltered coastal waterways). It can be seen that 50% of the sample were for routes of less than 25 miles and 75% were not more than 70 miles in length the sample contained 85% of the passengers carried.

People will cross 5 miles of water to shop, 10 miles or so to commute to work enjoy week-end breaks over 25 miles but only once a year cross 100 miles or more of water. to journey on holiday. Generally frequent journeys that would take longer than 2½ hours by sea are more appropriate to air travel unless it is necessary to carry cars.

Now we can begin to combine the size and potential traffic requirements and take into account numbers of craft per route, load factors etc. On this basis we can produce curves such as A & B in Fig.5. Note how sharply the speed requirement falls as ferry route length increases on this basis. The other considerationofcourse must be that the craft should be operated for as high a utilization as possible bearing in mind the proportion of journey time to total time including boarding and disembarking etc. As an example one could consider the 'comfortable' craft and assuming a 70% utilization factor produce a curve such as C. This of course is quite the reverse of the size and traffic limited bourdary curves as the advantage to utilization of speed can best be realized as stage or route length increases, A base line D in Fig.4 chosen to represent approximately 50% above what speed could be expected from slower large ferries completes the boundaries of the market needs.

The speed boundaries limited by seastate and traffic indicate an approximate envelope for a large sample of routes and within the triangular area are distributed each individual route requirements. The 'market ccntre' on this basis is in fact around 25 miles route length at a speed of 30-40 Kts. Craft of around 35-40 metres in length with an impact wave clearance of more than 2 metres should therefore appear to have considerable 'potential'. Of course a Lot of shorter, dense traffic routes with a high frequency service can readily apply much smaller craft e.g. the HM200 series of Hovermarine which started as a 60 passenger 16 metre craft and has evolved through an 84 passenger 18 metre version to now the 120 passenger 21 metre version all with the same basic production tooling.

The Growth of High Speed Passenger Ferries

Historically since 1970 the average high speed ferry craft (i.e. over 28 Kt. carrying more than 50 passengers) has increased in capacity from 90 seats to in excess of 180. In 1970 there were 160 such high speed craft in operation - today the figure is in excess of 600.

New craft sales and craft entering service each year since 1980 is as follows:-

1980	1981	1982	1983	1984	1985	1986	1987
21	38	51	23	22	23	32	30+

These figures are approximate and interpret reports based on Janes "Surface Skimmers" and the High Speed Surface Craft but are of the right order. The present growth area appears to be in seat capacities in excess of 200 and with larger craft in service with well established operators most on already established routes. Whereas in the early 1970's the markets scemedto bedominated by the smaller hydrofoil craft the present 'vogue' is for catamarans and surface effect ships. The now long lived large SRN4 amphibious hovercrafthave considerably exploited the English channel routes taking up to about 30% of the available traffic but it would appear that this size, speed, cost combination was not attractive on a worldwide market basis.

However the tendency is gradually establishing itself towards larger craft providing 5 more comfortable ride but the lower first costs appears to compromise the growth of speed even though in certain instances through life costing estimates of some higher performers need not bc any greater. What the future holds depends on the competitive edge operators will want to establish in their market sectors. The passenger will always want to travel faster provided the cost premium is not too great and he has a reasonably comfortable ride. A lot of new passengers and operators have been introduced to' the benefits of a higher speed with a modest step into catamarans - this may indeed have provided a useful stepping stone to higher speed craft.

Other Commercial Applications

With regard to cargo or frciqht carrying with hiqh speed craft the cargo price has to benefit from speed. Westamarin with their 5000 series open sea catamaran have made a bold step into this area whore perishable food is concerned. Certainly there are other areas in the world where such applications would bc worthwhile where fish, meat, and high valued agricultural produce would benefit from shorter sea journey times to markot with-benefits of freshness and premium prices.

Coastal or interisland fast distribution of containers from a main port should be attractive as an 'express' service where quick deliveries can justify premium rates. The light package 'express' service delivery business is growing and craft able to accommodate light vans or lorries would be worthwhile for overnight short haul routes.

In the past we have looked at riverine fast transport. In fact the very first application enquiry for an S.E.S. craft in 1969 was to take cargo of tea by river from the Assam valley in India down the Brahmaputra to Calcutta. Very simple unsophisticated high speed barges may evolve for certain areas (see Fig.61 but it is obviously difficult to compete in the main with efficient rail and road transport if such facilities are available.

High speed craft are by definition relatively light loaded vessels and are best suited to cargos which have a loading density no more than the base loading of the craft. Even at this density only about 40% of the deck space can be used since it is unlikely apart from rivertype vessels that the payload fraction could be higher than 40% of the all up weight of the craft.

Perhaps the most appropriate 'cargo' fcr high speed craft is that directly associated with passenger; i.e. his personal transportation module. i.e. the car. It is difficult however to charge on a weight basis the same fare for cars as passengers and therefore only on routes that have dense traffic and preferably on an all year round basis will such applications be viable.

The English channel is one application as previously mentioned that has at least sustained such a service with the amphibious hovercraft. Other now more cost effective options now offer themselves. (- and I'm not referring to the tunnel). One possibility is to use the deep cushioned 60 metre 54 Kt. S.E.S. craft as proposed by Hovermarine in the HM760 design. See Fig.7.

A large proportion of international ferry routes depend heavily on having a captive market on board for the purchase of duty free goods. On some routes this is such an important factor that they would not be commercially viable without such facilities.

The size of the manufacturers' market at todays costings is somewhere in the range of 60 - 120 million U.S. dollars per year. There are up to sixteen manufacturers able to seriously compete for this. In order to sustain this industry other ways and outlets will have to be found. One way of expanding the market is of course to promote licence construction perhaps in areas where lower cost labour is available. In certain developing countries water transport may be the only practical alternative to costly road construction which requires huge sums of capital investment. In such areas river transport could generate a large sector- in the market.

One thing is certain and after over twenty years in this business I have no hesitation in saying that one has to MAKE one's market in high speed marine transport. There is no natural hidden craving for our products just waiting to be discovered - the hard facts are that commercial and govermental inertia, prejudice, already vested interests all continue to resist change and investment in this exciting transport area. In spite of this and through costly development and demonstration programmes some of us, some of the time come through with gleaming success. Inevitably such success is cyclical and to sustain such enterprise the companies involved must have other products or services to sell.

As requested my remarks have been limited to commercial craft as I believe other speakers have been asked to talk of potential military applications. However I believe the market so far developed for commercial applications is just the tip of the iceberg - the hidden potential for large vessels will be military and quasi military and in value an order higher.

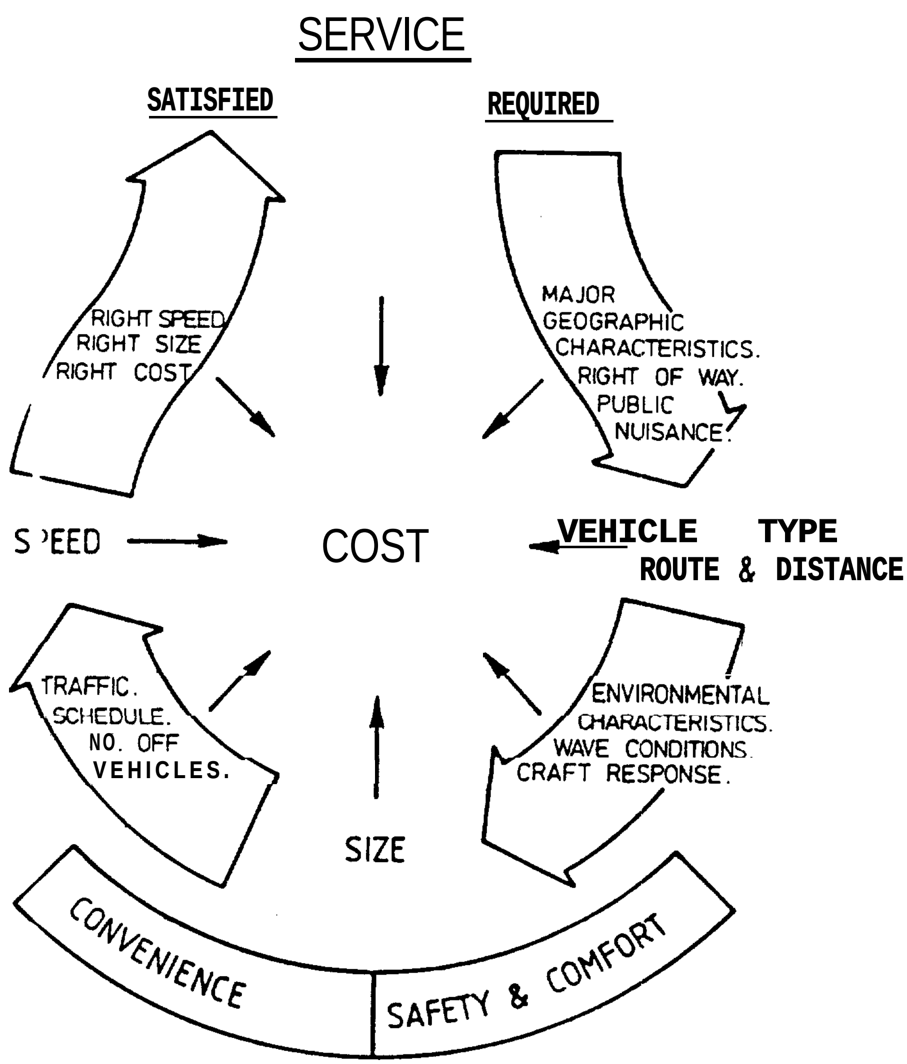

SERVICE
SATISFIED
REQUIRED
RIGHT SPEED
RIGHT SIZE
RIGHT COST
MAJOR
GEOGRAPHIC
CHARACTERISTICS.
RIGHT OF WAY.
PUBLIC
NUISANCE.
SPEED
COST
VEHICLE TYPE
ROUTE & DISTANCE
TRAFFIC.
SCHEDULE.
NO. OFF
VEHICLES.
ENVIRONMENTAL
CHARACTERISTICS.
WAVE CONDITIONS.
CRAFT RESPONSE.
SIZE
CONVENIENCE
SAFETY & COMFORT

FIG. 1

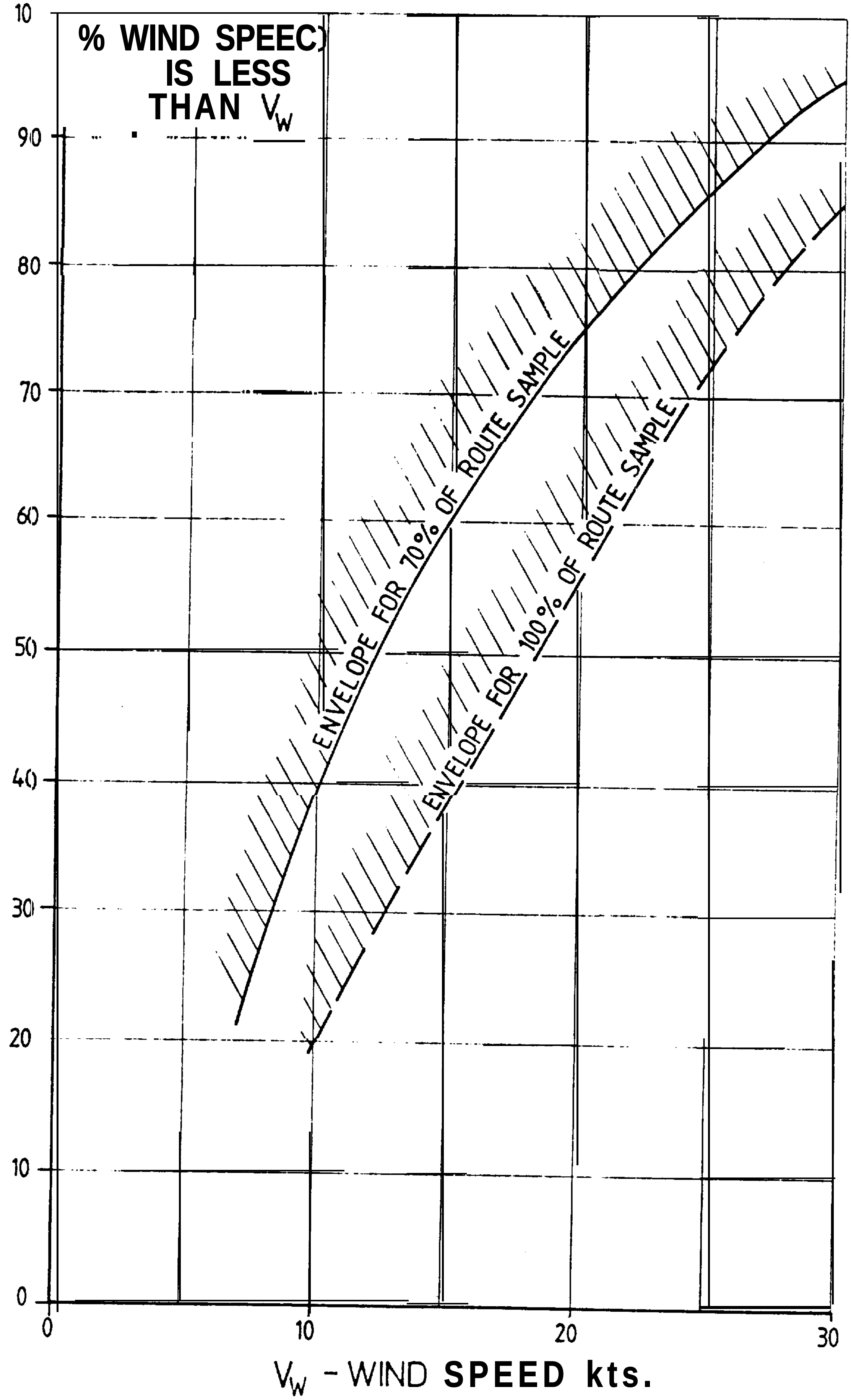
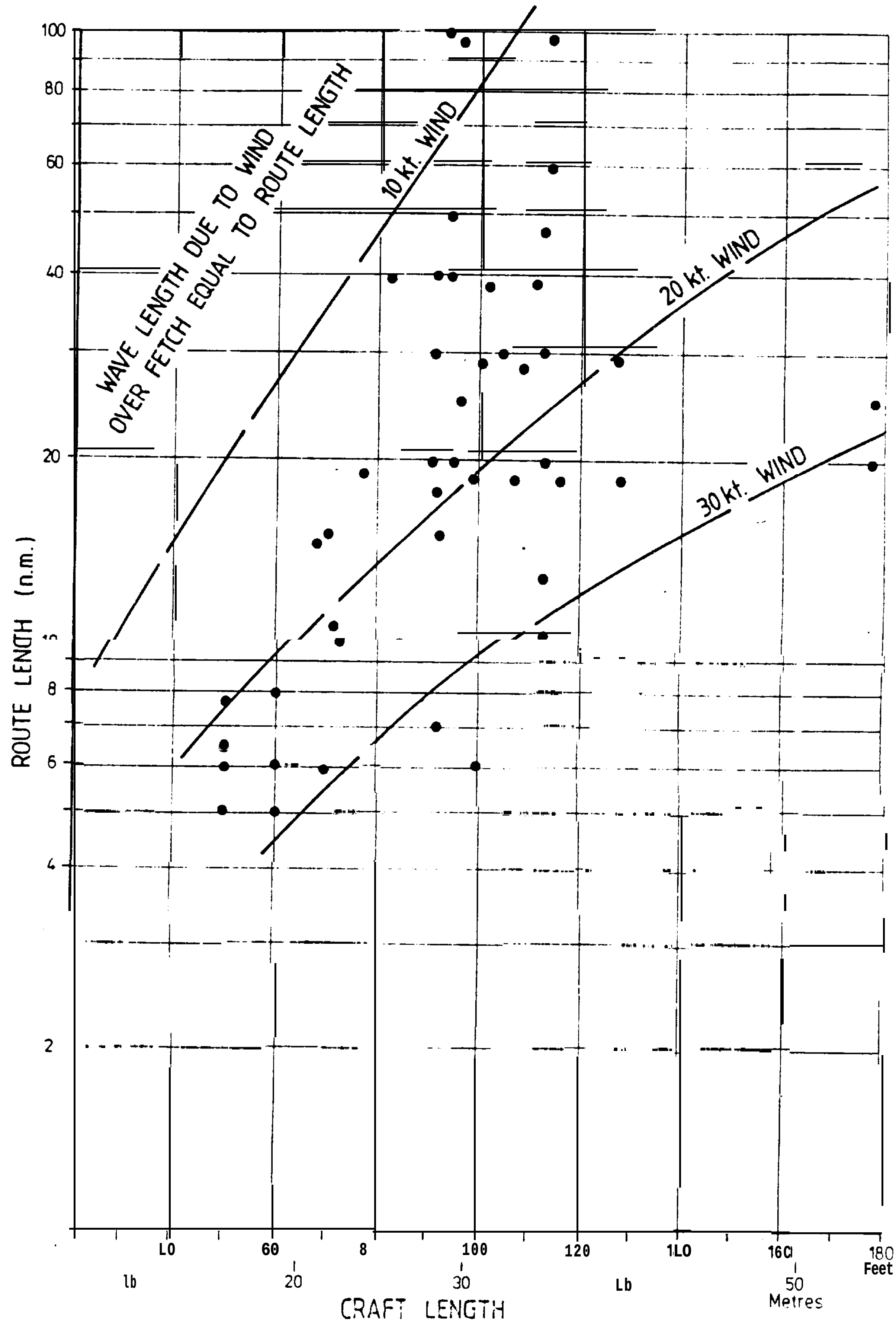
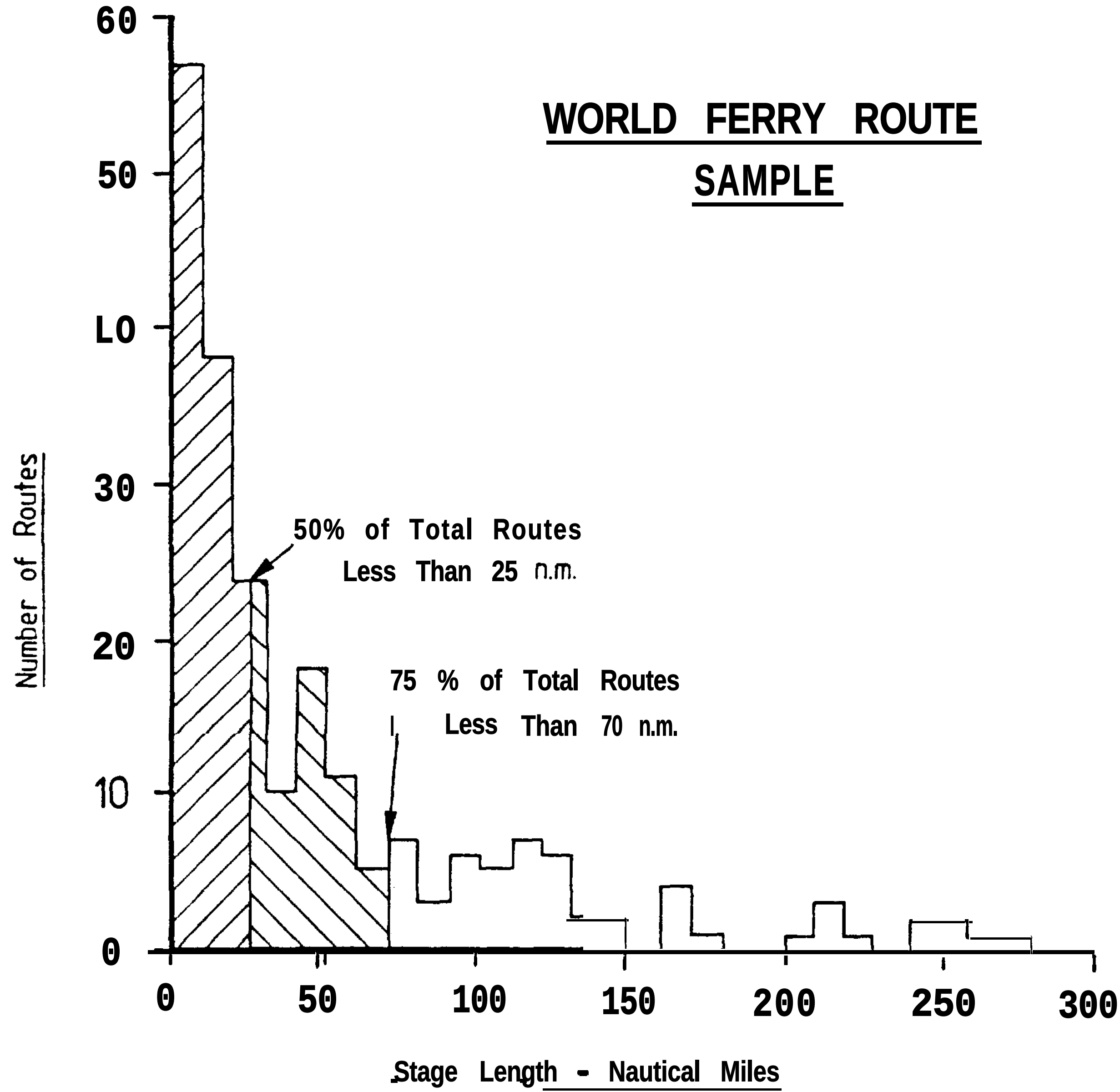

% WIND SPEED IS LESS THAN V_W
ENVELOPE FOR 70% OF ROUTE SAMPLE
ENVELOPE FOR 100% OF ROUTE SAMPLE
0
10
20
30
40
50
60
70
80
90
10
V_W - WIND SPEED kts.

FIG.2



ROUTE LENGTH (n.m.)
100
80
60
40
20
8
6
4
2
WAVE LENGTH DUE TO WIND OVER FETCH EQUAL TO ROUTE LENGTH
10 kt. WIND
20 kt. WIND
30 kt. WIND
60
8
100
120
180
Feet
20
30
50
Metres
CRAFT LENGTH

FIG.3



WORLD FERRY ROUTE
SAMPLE
60
50
LO
30
20
10
0
Number of Routes
50% of Total Routes
Less Than 25 n.m.
75 % of Total Routes
Less Than 70 n.m.
0
50
100
150
200
250
300
Stage Length - Nautical Miles

FIG. 4

16

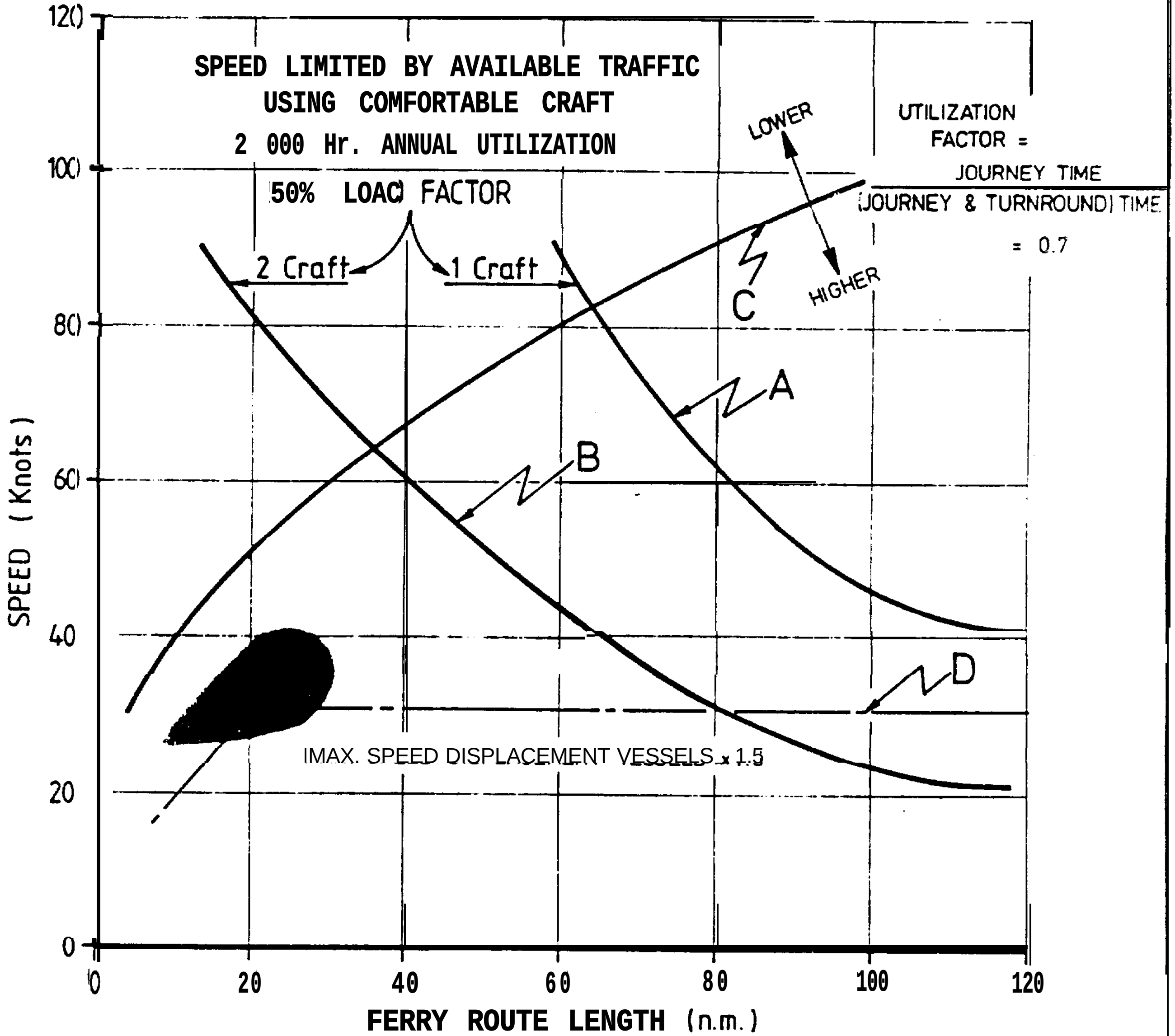

SPEED LIMITED BY AVAILABLE TRAFFIC
USING COMFORTABLE CRAFT
2 000 Hr. ANNUAL UTILIZATION
50% LOAD FACTOR
2 Craft
1 Craft
LOWER
HIGHER
UTILIZATION FACTOR = JOURNEY TIME / (JOURNEY & TURNROUND) TIME = 0.7
A
B
C
D
MAX. SPEED DISPLACEMENT VESSELS x 1.5
SPEED (Knots)
0
20
40
60
80
100
120
FERRY ROUTE LENGTH (n.m.)

FIG.5

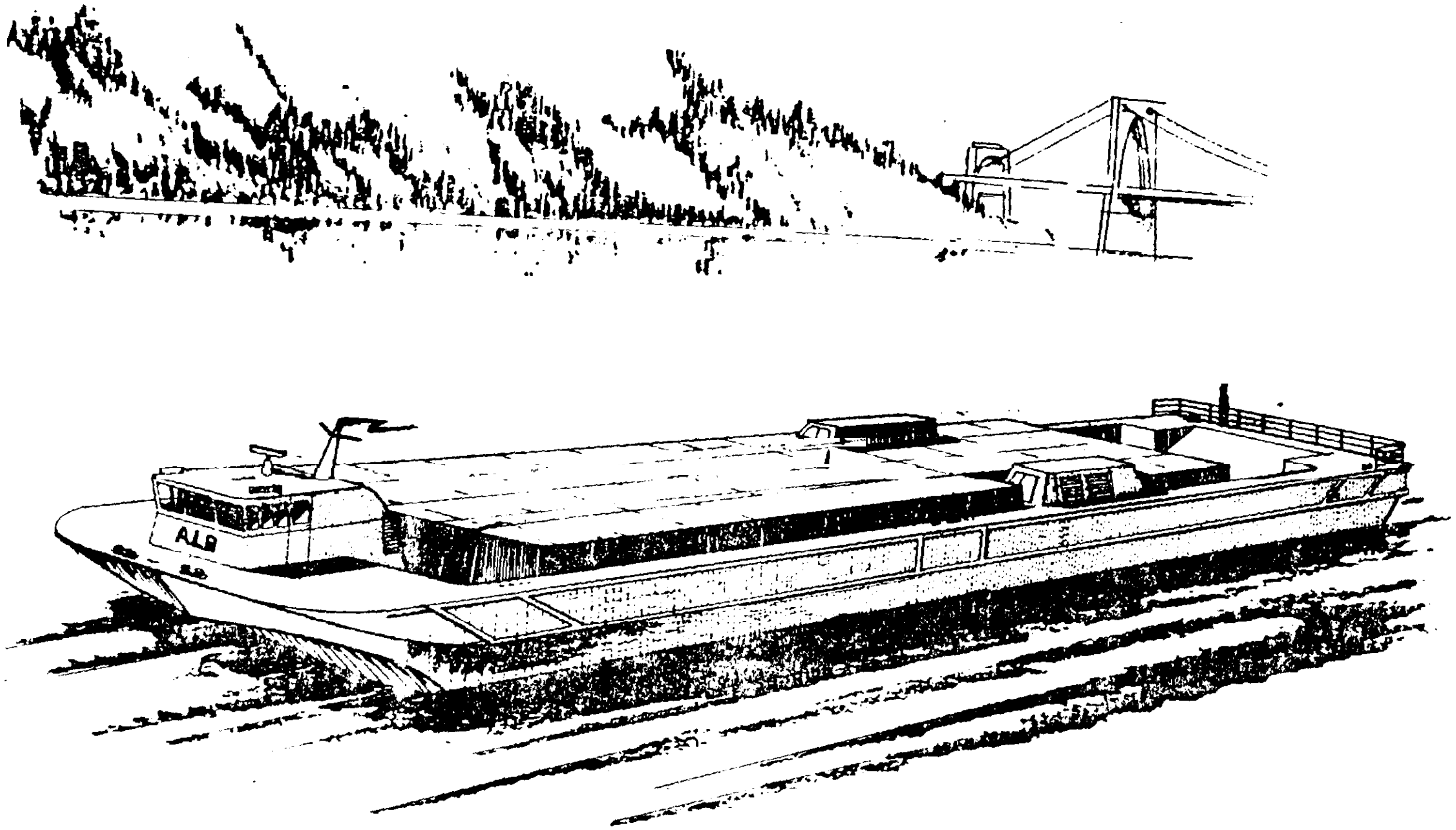
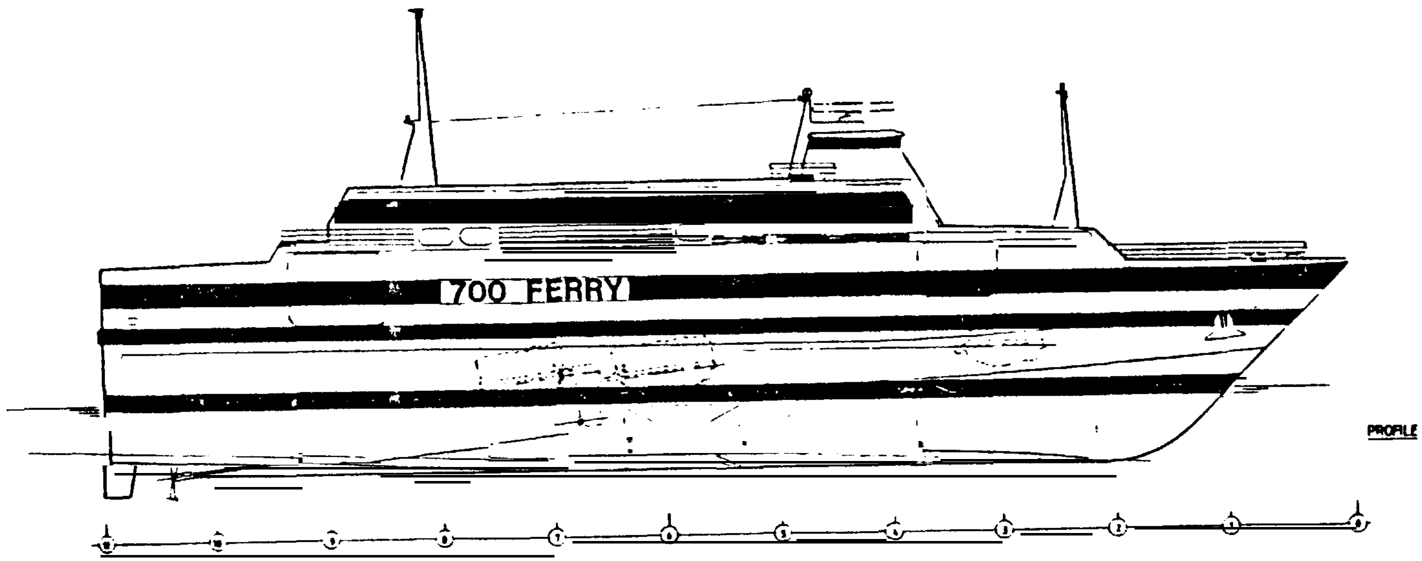


FIG.6

100 Ton PAYLOAD
HOVERFREIGHTER



The HM760 - 60 metre Passenger/Car Surface Effect Ship

Payload: 500 passengers and 76 cars

Maximum Speed: 54 Kts.

Speeds up to 40 Kt. in 3-4 metre waves.

HIGH SPEED MARINE CRAFT

CONFERENCE 1988

Technology update and market potentials.

NORWEGIAN SOCIETY OF CHARTERED ENGINEERS

TODAY'S AND TOMORROW'S MATERIALS FOR HIGH SPEED MARINE CRAFT

A. MAR-
Director : CETEC A/S
: Marchant Filer Dixon
Consulting Group

SUMMARY

All high speed marine craft rely on the efficient use of materials and as greater efficiencies are sought, the more important becomes the need to apply materials to minimise weight whilst maintaining an acceptable performance. This paper reviews the current materials available and looks ahead to how these materials can be applied to the benefit of high speed craft.

1. INTRODUCTION

There is no one material that answers all the design criteria for high speed marine craft, simply because the craft vary in size, speed, operational environment, quantity, cost restraints and function.

There is undoubtedly though one very important factor to be considered by any designer for high speed craft and that is how to produce a light, yet robust structure at minimum first cost and minimum through life cost.

This paper will address the possible candidate materials for primary and secondary structure with this aspect of cost effectiveness in mind and with a particular emphasis on the need to keep weight to a minimum and yet maintain a sufficient level of robustness.

The choice of materials is extensive between the range of suitable metals and composites (fibre reinforced polymers). However, given a clear set of criteria embodying both the structural performance required, the manufacturing facilities and the operational requirements, the choice can be realistically quantified.

It is suggested that if the structural engineer, materials technologist and naval architect were to work together at the conceptual design stage and onwards through to production, then a more radical approach to the overall design of fast marine craft could he taken leading to improved performance.

2. CANDIDATE MATERIALS

High speed marine craft generally require a light yet robust structure and if they are to operate in a commercial environment, the structure must be cost effective. Military vessels may accept a higher structural cost for the gain in performance and therefore the more expensive materials could be considered.

The choice of material lies between metals and the non-metallic materials. Metals can be split between steel alloys and aluminium alloys.

The non-metallic materials include wood and fibre reinforced polymers (composites). The use of wood for primary structure has decreased dramatically since the introduction of composites, but plywood still enjoys a high usage in internal and secondary structure.

Attention here will be given to high strength steel alloys, aluminium alloys and fibre reinforced polymers.

Comparing materials is a difficult task and can lead to a great deal of misunderstanding and confusion. To properly quantify the benefit from one material to another, it is necessary to design structures and have them costed. This is time consuming and expensive. However, comparing materials by their mechanical properties and raw costs is only indicative but nevertheless a useful task as it does put into perspective the various materials, particularly if specific values are compared.

Table 1 provides a comparison of the basic mechanical properties of the various materials discussed. The material raw cost will of course depend on quantity purchased, source and exchange rates.

* STEELS

The most common carbon steels (mild steel) have been used extensively in ship building as they are inexpensive, easy to weld and therefore low in fabrication cost. For high speed craft however, where weight becomes more critical, they are naturally heavy, which added to through life problems of corrosion Leading to increased maintenance costs, reduces their acceptability.

MATERIAL	Matrix	Fibre Weight Fraction	Density Laminate (g/cm^3)	Tensile Strength (N/mm^2)	Tensile Modulus (KN/mm^2)	Compressive Strength (N/mm^2)	Specific Tensile Strength	Specific Tensile Modulus	Material Cost £/kg
Mild Steel BS4360-43C			7.8	130-540	207	255	62	2-1	0.35
Steel Cor-Ten A			7.8	480	207	340	62	27	0.40
HSLA			7.8	620	207	550	80	27	0.40
Al. Alloy 5083-0(N8)			2.67	312	71	140	117	27	2.10
Al. Alloy 6082-TF(H30)			2.70	310	69	270	115	25	2.40
'E' Glass Random Mat	Polyester	0.33	1.44	80-130	7.3-9.3	140-150	73	6	1.60
'E' Glass Woven Roving	Polyester	0.50	1.63	210-300	12-21	I SO-270	156	10	1.80
'S' Glass Woven Roving	Polyester	0.50	1.64	440	20	210	26a	12	5.50
Aramid K49 Woven	Polyester	0.44	I. 31	430	26	115	328	20	17.10
Carbon Fibre Woven	Polyester	0.40	1.40	460	30		330	11	35.60
Aramid K49 Woven	Cold Cure Epoxy	0.55	1.31	450	30	172	344	23	28.50
Carbon Fibre Woven	Cold Cure Epoxy	0.59	1.47	550	55	360	374	37	37.30
PEEK/CF APC-2	PEEK	0.67	1.66	2130	134	1100	1283	al	150.00 122 $/#

TABLE 1 : COMPARATIVE PROPERTIES OF METALS AND COMPOSITES.

However, the advantage of low material costs has led to the use of such steels as the weathering steel Cor-Ten A and more recently, HSLA. The attraction being improved properties leading to reduced structural weight whilst maintaining a relatively low fabrication cost compared to aluminium alloys and composites.

High strength low alloy steels (HSLA) have been researched for use in naval vessels(1) where benefits of reduction in cost of welded ship structures when compared to HY-80 steel, have been shown to be in the range $0.40 to $0.90 per pound (£0.18 to £0.14 per kilogram), which would lead to significant savings for a large naval vessel. HY-80 is a difficult material to weld requiring a pre-heat requirement as part of the welding process. HSLA does not require this and it is claimed an easy material to weld.

Despite the good tensile strength properties of steel and high stiffness, the attendant high density of the material reduces the specific tensile strength to the lowest of the materials presented in Table 1, with a good comparitive specific stiffness. As strength is normally the limiting criteria, as opposed to stiffness, then the use of steel will not produce the lightest structure.

Specialist steels such as stainless steel and Monel will continue to be used for ancillary components, but because of their high cost and difficulty of fabrication, they are unlikely to be used for primary structure.

* **ALUMINIUM ALLOY**

Like steel, this material has been used extensively for small vessel construction and for the superstructures to large vessels. The materials used in the marine industry are well understood and favoured for high speed craft because of their improved weight performance over steel and their ease of limited quantity fabrication when compared to composites.

The mechanical strength of the 5083 and 6082 alloys are not particularly high, but when looked at specifically, they compare well with other materials - even on stiffness.

The material is easy to weld if undertaken properly, but the weld reduces considerably the strength of the parent metal resulting in increased weight over a rivetted or bonded 'structure. Rivetted structures are however much more expensive due to the higher labour content.

The bonciing of aluminium alloy (and of steel) is receiving attention in the automotive industry as a means of improving structural efficiency. The technology of adhesives exists to join metals - for instance aircraft structures have been Redux bonded for several years, but the usage of bonding primary structure of aluminium alloy for marine vessels requires research and production experience.

Development of aluminium alloys has been primarily aimed at the aircraft and automotive industries by the introduction of aluminium lithium alloys giving a reduction in material density for lower weight aircraft structure and superform aluminium alloy for low cost complex shaped components. Neither of these alloys would be efficiently used in the marine environment due to their poor corrosion resistance.

*** COMPOSITES**

Glass reinforced polyester was introduced into the marine industry about 40 years ago and is now a commonly used material. The material is favoured for its good environmental resistance, good formability and as grp, requires only semi skilled labour in production.

The common E glass chopped fibre reinforced polyester has only modest strength and low stiffness but due to the material's lower density than metal, exhibits an acceptable specific tensile strength but a low specific modulus. Using E glass continuous fibre as woven rovings improves the specific strength value but still leaves a low specific modulus.

E glass fibre reinforced polyester remains the most commonly used materials for composites, primarily because of their low cost and ease of handling in production. Designing for stiffness though results in the need of sandwich type materials or increased mass compared to metals.

Improved glass fibres such as S and A glass, have not been used extensively in Europe, despite their improved mechanical properties. This may be because of the much higher cost than E glass fibres.

The introduction of aramid fibre (Kevlar 49) has provided the opportunity to dramatically increase the properties of thermosetting resins such as polyester and epoxy and as the fibre has a density nearly half that of glass fihre, the specific strength and stiffness values are significantly improved. From Table 1 it can be seen that the specific tensile strength of woven Kevlar 49 is about twice that of woven E glass fibre and three times that of aluminium alloy with a specific stiffness 20% less. Kevlar however, does have a relatively low compression stress which inhibits its use in certain structural areas.

Carbon fibre, the most expensive of the currently available reinforcing fibres, has marginally improved tensile and stiffness properties, but much improved compression properties compared to Kevlar fibre. However, being a conductive material, its usage in a marine environment requires care in detail design to avoid corrosion where metals are used.

The marine industry has yet to use the more expensive thermoplastic materials which are being extensively researched and used in the aircraft industry, where weight reduction is of a much higher priority and comparative structural costs are also much higher. Table 1 includes the properties of APC-2 continuous carbon fibre reinforced PEEK (polyetheretherketone) thermoplastic in order to put into perspective the properties and costs. It is perhaps worth noting that the specific tensile strength of APC-2 is some twenty times that of steel and ten times aluminium alloy with a specific stiffness three times that of metal. However, the raw material cost is also to be noted as over four hundred times that of steel giving the obvious reason for its limited use.

3. MATERIAL SELECTION FOR PRIMARY STRUCTURE

Material selection for primary structure, i.e. the hull and superstructure, decks and bulkheads, should be made against a clear set of parameters, not by some arbitary choice based on say, what materials and production facilities are available.

For high speed marine craft, the parameters are undoubtedly going to include minimum weight and cost. Minimum weight should always be a driving parameter because excess weight simply means excess money, increasing capital cost and through life cost.

But weight has also an effect on performance as the greater the weight, the slower the speed or increased engine power to maintain the design speed.

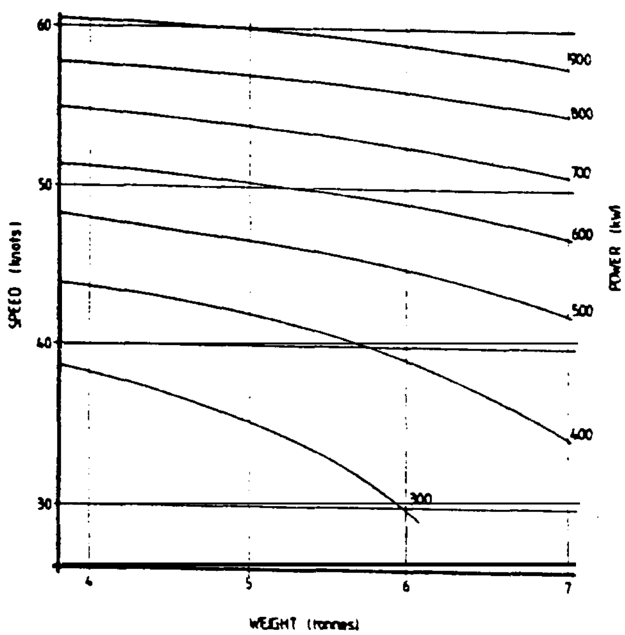


Figure 1 : the effect of weight on speed for a 9 metre planing craft.

To illustrate the effect of weight on performance, consider as an example a 9 metre high speed planing boat with a displacement of approximately 5 tonnes. Constructed in aluminium alloy to normal standards, the structure would be approximately 2 tonnes. Figure 1(2) plots speed versus weight for various power ratings and indicates the fall off or increase of speed for an increased or decreased weight from the 5 tonne displacement considered.

For instance for a power of 300kW (400hp), a reduction in weight of 1 tonne increases the speed from 35 knots to 38 knots and an increase in weight of 1 tonne decreases the speed to 30 knots. The effect of weight reduces as the speed increases, but ferries operating at between 20 and 35 knots are currently the norm, so weight control and design for minimum weight are important. Increased weight will also effect through life cost as if more power is required then increased running costs will be incurred etc.

However, weight reduction must be executed cost effectively whilst maintaining an adequate level of robustness to avoid through life damage. It is up to the structural designer to select his materials to suit the cost parameters set, but in order to do so, he must work closely with the naval architect to quantify the weight/cost/performance cycle.

As a general rule, steel structure will be the cheapest in cost per kilogram fabricated, but also the heaviest. Because of the significant increase in mass over aluminium alloy or composites, the resulting total structural cost may be higher than the lighter materials. Choosing steel, even the higher strength alloy steels, will, if properly welded, give a good life as fatigue is unlikely to be a problem. There is insufficient published evidence to provide quantifiable data on cost of such structures for high speed craft as it is little used. The material has been favoured by German and American builders of surface effect ships, but probably more from tradition rather than performance.

For minimum weight, the choice is left between aluminium alloy and composites. It can be readily shown, and has been by a number of designers, [3][4], that welded aluminium alloy cannot compete with properly engineered composites on weight. However, if it is one-off construction, the cost of the composite tooling results in a total cost in excess of the aluminium alloy construction, although this cost difference can often be eroded by the use of sandwich construction for composites and the cost of finishing aluminium alloy structure to an acceptable external aesthetic level.

The use of rivetted aluminium alloy, as used for instance by Rodriquez Cantiere Navale for their hydrofoils, generally increases the cost of manufacture but reduces the weight, due to increased mechanical properties of the material over welded structures. The very lightweight rivetted construction as used by the British Hovercraft Corporation SRN4 hovercraft, undoubtedly produces a light weight structure, but at a high cost.

Where vessels are to be constructed in quantities of more than three or four (depending on size), then composites will be both the cheapest and lightest. Properly engineered, they will also give added benefit in through life costs due to their improved environmental resistance over metals and their much improved fatigue life over welded aluminium alloy.

The current difficulties of composite structure are more subjective than objective. Due to many problems in the past of poor design and construction, and to some extent from the lack of good design standards, there is a reluctance on behalf of some naval architects to specify composites. Perhaps also the vast range of fibres and resins inhibits the inexperienced designer in the use of composites. It is, however, this range which provides the designer with the means of taking weight reduction to an optimum level as the fibres can be selected and stragetically used to provide local strength or stiffness.

4. MATERIAL SELECTION FOR SECONDARY STRUCTURE & COMPONENTS

Insufficient attention is given to material selection for secondary structure (propellor shafts, rudders, superstructure) and to components (internal finishing, deck fittings, seats etc.). For high speed craft, more attention should be paid to this area as weight can be readily reduced often without incurring a cost penalty.

Some interesting work has been carried out on propellor shafting, comparing composites with conventional steel shafting for both warships and smaller vessels ([5]). For smaller vessels the researchers have indicated that for an 8 metre vessel, the shafting weight of 227 kgs in copper alloy can be reduced to 68 kgs in filament wound composites, all for a surprising 508 cost reduction. The cost reduction is assisted by the reduced number of support bearings as a result of reduced mass effecting shaft whirling levels. Composite shafting is now being used in the automotive industry for the same reasons.

Where vessels have been designed for minimum weight primary structure, it is dissapointing to see weight being thrown away on the fit-out, particularly in furniture, doors and trim. A philosophy similar to that used in aircraft design should be adopted - that of minimum weight throughout, but at a quantified cost level. For instance, solid plywood doors can be replaced by sandwich construction of plywood and lightweight cores to dramatically reduce weight at no additional cost. Aluminium alloy skinned aluminium honeycomb sandwich construction is a good choice for decks and bulkheads, leading to reduced weight and increased stiffness.

Stainless steel is a traditional material for deck fittings and finishing, but it is becoming expensive and is, of course, heavy. There are several non-metallic alternatives, which if carefully selected, can provide increased performance without a reduction in aesthetic value.

Flexible materials are used in hovercraft for skirts and for inflatable craft. Nylon reinforced neoprene and natural rubber being the choice for hovercraft skirts with Kevlar added for higher strength and wear resistance for inflatable craft.

5. THE POSSIBILITIES AND CHALLENGES

Today's challenge for high speed marine surface transportation must be the need for reliable and cost effective craft. Materials are at the heart of this challenge.

For progress to be made though, experience and data must be accummulated. This requires money, but above all, courage on behalf of the designers and operators to explore the possibilities. A great deal of progress has been made with hydrodynamics, and even aerodynamics, to push the speed of vessels upwards whilst maintaining efficiencies of power and cost. Less progress has been made with the application of materials for the benefit of structural and component efficiency.

To reduce the cost of reasearch and development, the marine industry can turn to other industries, such as the aircraft and automotive industries, where the experience in lightweight structures can be found in the former and production techniques in the latter. This has certainly happened in the hovercraft industry where designers have been drawn from the aircraft industry to produce both large and small hovercraft. A disadvantage of this approach is the attendant high costs of some manufacturers who have built such craft in an aircraft manufacturing environment. It is unlikely that the design skills will be found in conventional shipbuilding which could handle with experience the non-metallic higher performance composites. It is therefore suggested that the high speed craft industry, one which is already justifiably finding its own market niche, should create its own breed of design and manufacturing skills.

An alternative, or indeed a parallel approach to accummulating data, is by collaborative research. A recent programme of work reports a collaborative joint venture development programme in the UK betwen Du Pont de Nemours, Cougar Holdings Ltd., Scott Bader & Co. Ltd. and design consultants(6). This programme was concerned with the application of high performance fibres (Kevlar 49) and new types of matrices (modified acrylic polymers) to produce light, yet robust, structures for high speed power craft and to investigate the induced forces and craft motions by direct measurement. Material suppliers in conjunction with power craft builders and designers collaborated to research jointly their own particular requirement. More such programmes are needed to provide the seeds of experience in materials on which the high speed craft industry can grow.

The metals currently used by the industry have to a great extent reached a plateau of development. Certainly, higher performance alloy steels have been introduced, but these are not ideal materials for craft seeking minimum weight.

It is unlikely that aluminium alloy can be improved significantly for marine use. Most vessels employing this material use welding as the manufacturing method, which can lead to poor service performance due to fatigue cracking. The material does not perform well in the presence of fire, which is causing concern amongst some end users. To reduce the problems associated with welding, development must lie in adhesive bonding as rivetting is a labour intensive activity.

Composite materials have established themselves as the main production material with glass reinforced polyester (grp) the most extensively used material for small to medium leisure craft. It is interesting to note that E glass and polyester resin are cheaper in raw material cost than aluminium alloy, so that provided the inherent tooling costs of composites can be amortised by quantity production, grp will remain as the most widely used material because it is cost effective for both first cost and through life cost due to much improved environmental performance.

Materials that are high in raw material cost, do not necessarily mean they are expensive in product cost as the manufacturing cost per tonne of material is now the major cost of the finished product. Care must therefore be exercised by designers to quantify material cost by equating finished structure or component cost and better still, but comparing through life cost.

A challenge for tomorrow with respect for composite materials lies in this factor - the conversion cost of the raw material into the finished product.

Probably the best example of an efficient conversion process is extruded aluminium alloy. The process requires only minimal tooling cost and the associated labour cost is low, resulting in only a small mark-up on the material cost to the finished product.

Composites require tooling to make the product and if that product is a 30 metre vessel, the tooling can cost as much as the first structure. This factor inhibits the use of composites for one-off construction. Various cheaper alternatives to a large female tool exist such as the use of sandwich construction on a timber frame mould or the use of pultruded composites to provide the hull shape and hold the foam for the sandwich core (7).

To make a major breakthrough in composite processing requires a new approach to tooling. One such approach would be to have a variable surface tool that could be changed in shape to make a variety of components or portions of a large hull which would be subsequently assembled to make the complete hull. The tool would be computer driven using design co-ordinates generated in the drawing office, therefore providing a true CAD/CAM system dedicated to making cost effective and highly efficient light weight structures for high speed craft.

Another major challenge lies ahead though if materials are to be exploited to the benefit of performance of high speed craft and that is the challenge to get the naval architects to work with the material technologists, the structural designers, the interior designers and the material processors at the conceptual stage of any project and to remain as an integrated team dedicated to the task of optimum performance right through to production.

Too often an architectural design is complete before the problems of materials and structure are addressed, leaving the structural engineer with the very difficult task of meeting a target weight and having to contend with extremely complicated hydrodynamic or aerodynamic shapes.

Good design is a compromise between the many facets that make up the total design. Starting with an integrated team is of paramount importance.

If such freedom of thought was allowed by the materials engineer and the structural designer, the process of thought must be to find the right material which is fit for the purpose, cost effective and acceptable for the given parameters. There will not be one material which will be suitable for everything. As an example, consider a large one-off SWATH vessel, where such a process of thought could lead to the use of geodesic structure incorporating tubes of suitable material to meet the particular loading mode, and where also no tooling would be required. However, because a geodesic structure is to be considered, the naval architect must compromise hydrodynamically.

Such freedom of thought from the conceptual stage of design is essential if today's materials are going to benefit tomorrow's high speed craft, and indeed such freedom of thought, combined with the courage of designers and operators to explore all possibilities, is essential if tomorrow's materials are going to be used.

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HIGH-SPEED MARINE CRAFT
KRISTIANSAND 4-6 MAY 1988

PROPULSION SYSTEMS

FOR

HIGH SPEED MARINE CRAFT

BY

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CONTENT

1. ABSTRACT

Fundamental problems related to propulsion of high speed crafts below 55 knots are discussed. The discussion has been limited to waterjet propulsion and to propulsion with conventional and partly submerged propellers. Results from model tests with a propeller operating in fromt of a z - drive are presented. A comparison is made between the propulsive efficiency of a ship propelled alternatively with conventional propellers, a z - drive and a waterjet .

Finally propulsion in a seaway is discussed.

2. CONVENTIONAL PROPULSION AND PROPULSION WITH A Z DRIVE

High speed crafts propelled with conventional propellers do normally have a shaft arrangement as shown in fig.1. This system works sufficiently for moderate speeds but for higher speeds the the appendix resistance will become to high. Rudder, shafts and brackets may at 20 knots give a resistance which is approximately 2% of the total resistance. At 40 knots this resistance may be as high as 10% of the total resistane as shown in the figures 2, 3 and 4. At speeds above 50 knots such arrangements are difficult to use due to cavitation and ventilation.

The resistance of a rudder or a strut is:

$$R_e = C_{Dr} \cdot \rho/2 \cdot Vs^2 \cdot \cdot$$

where

V_s = speed of the craft

$Rn = \dfrac{V \cdot c}{\upsilon}$ (Reynolds number of rudder or strut)

S = wetted surface of the rudder (or strut)

$$C_{D_r} = \frac{0.075 \cdot (1+2 \cdot t/c)}{(\log Rn - 2)^2}$$

c = chord length of the rudder

t/c = thickness chord ratio of the rudder

The resistance of the shaft or the fairing is according to Hadler (6):

$$R_S = \rho/2 \cdot Vs^2 \cdot 1 \cdot d \; (1.1 \cdot \sin^3\varepsilon + \pi \cdot C_F)$$

where

$$C_F = \frac{0.075}{(\log R_{ns} - 2)^2}$$

$R_{ns} = \dfrac{v \cdot 1}{\upsilon}$

ε = shaft inclination angle

d = shaft diameter

1 = length of shaft

If this equation is applied for the barrel, corresponding length and diameter are used. For the system shown in fig 1 the added resistance is given in the figures 2, 3 and 4. The added power due to appendix drag is presented as a fraction of HP. In the calculation of the resistance the equations given above were applied. For speeds above 40 knots it is evident that this resistance becomes a problem. Model tests confirm that the order of magnitude of the results are correct. Still cavitation and ventilation have not been considered. As a consequence of these circumstances it is natural to look for alternatives.

The waterjet has been proposed as an alternative to conventional propulsion. An other alternative is the Z drive which was introduced many years ago. See figures 5 and 6. Financed by NTNF Marintek tested several units of this type since 1980.

With the propeller in front of strut and body the inflow to the propeller becomes homogenious, which will reduce the cavitation induced noise. When the strut is equipped with a flap as indicated in fig. 5 the follqwing advantages are obtained:

1. Increased submergence.

2. The strut and the flap acts as an effective rudder. (See fig. 14 from [11])

3. Improved thrust/torque ratio and thereby total efficiency due to the interference between propeller, strut and body.

When the propeller is placed in front of the strut and the body the static pressure behind the propeller will increase leading to higher thrust and torque. This effect is discussed for excample in van Manens dr. thesis (10). We may very roughly express the torque and the thrust of an equivalent section of the propeller by:

$$dT_p = \rho\, 2\pi r\, [(\omega r + \Delta\omega r) - U_T/2]\, .\, [U_T + \Delta U_T]\, dr$$

and

$$dQ = \rho\, 2\pi r^2\, (V_A + U_a/2)\, (U_T + \Delta U_T)\, dr$$

where

U_T = tangential velocity far behind the propeller without the body and the strut.

ΔU_T = change in tangential velocity far behind the body due to strut and body.

$\Delta\omega r$ = change in inflow velocity to the propeller due to the presence of body and strut.

In this way it is possible to understand the marked increase in propeller thrust and the more moderate increase in torque resulting in an increase in efficiency. The drag of the strut will only to a small extent be compensated by the forward tilted component of the side force induced by the rotation of the propeller stream.

The z drive has resistance due to the speed of advance and due to the induced velocity of the propeller. If the propeller and the strut are close to each other the increase in propeller thrust will give an additional resistance on the strut as indicated in fig. 7. (ΔT). The corresponding increase in torque (ΔQ) is also indicated. If cavitation occurs on the strut and the body there may be a marked increase in drag as shown on fig. 9

This increase in drag will reduce the efficiency of the unit dramatically. Neglecting cavitation and propeller induced drag the viscous drag of the z drive can be estimated be dividing the unit in 3 parts as indicated on fig. 7.:

I : The body
II : The strut within the slip stream
III: The strut outside the slip stream.

Referring to fig 7 the drag og the body is:

$$R_I = (V_0 + C_a)^2 \cdot \rho/2 \cdot S_I \cdot (1 + \& \cdot D^*/L) \cdot C_F$$

$$Rn = \frac{(V_0 + C_a) \cdot L}{\upsilon} , \quad C_F = \frac{0.075}{(\log Rn - 2)^2}$$

where:

L = length of the body

S_I = wetted surface of the body
D^* = equivalent diameter of the body
k = form factor

The additional axial velocity Ca induced by the propeller is estimated as a function of the propeller loading:

$$C_T = \frac{K_T}{8\pi} \cdot J_A^2$$

$$C_a = V_A (\sqrt{1+C_T} - 1).$$

The resistance of part II is estimated from:

$$R_{II} = (V_0 + C_a)^2 \rho/2 \cdot S_{II} \cdot (1 + 2\ t/c) \cdot C_F$$

$$R_n = \frac{(V_0 + C_a) \cdot c}{\nu}, \quad C_F = \frac{0.0075}{(\log R_n - 2)^2}$$

where:

t/c = thickness cordlength ratio of the strut.

The resistance of part III is estimated with the same formulaes but using $C_a = 0$. We have used these formulaes for full scale and model scale for the unit shown in fig.5. As seen on fig. 8 this gives a considerable scale effect: If such units are applied one should therefore be very carefull in securing that the surface is as smooth as possible..

The unit shown in fig. 5 was designed by A.M. Liaaen and tested in the cavitation tunnel of Marintek. Drag, propeller thrust, torque and side force were measured for different flap angles and cavitation numbers as shown on figs. 9 and 10. The propeller was designed by Marintek and had the following characteristics:

Diameter model:	D	250
Number of blades:	Z	4
Expanded blade area ratio	EAR	0.74
Hub/diameter ratio:	d/D	0.326
Thickness/chord ratio:	t/c	0.021
Chord/diam. ratio:	c/D	0.507

The propeller was tested at P/D = 0.90, 1.2 and 1.45 at different cavitation numbers.

The efficiency of the unit is given by:

$$\eta = \frac{K_T}{K_Q} \cdot \frac{J_A}{2\pi}$$

where

$$KT = \frac{-D}{\rho n^2 D^4} + \frac{T_p}{\rho n^2 D^4}$$

D = drag of the unit

TP = propeller thrust

K_Q = propeller torque coefficient.

Fig. 10 gives the side force in Newtons as a function of the flap angle. For comparison fig. 14 may be studied. Based on the tests we get:

V_S (knots)	35	42	53
D (m)	1.250	1.250	1.250
K_Q (model)	0.0245	0.0250	0.0265
K_p (model)	0.0940	0.0823	0.0656
Kd (model)	0.016	0.018	0.0225
Kd (ship)	0.0097	0.01091	0.01029
J	0.839	1.066	1.235
P/D	0.985	1.180	1.390
HP	3293	2774	3589
η (model)	0.600	0.682	0.563
η (ship)	0.634	0.731	0.662

The difference in drag between full scale and model scale was calculated applying the procedure described above. The propeller was also tested without the z drive at different pitches and cavitation numbers. These tests give the following results:

V_S (knots)	35	42	53
D (m)	1.255	1.100	1.180
RPM	1000	1000	1000
P/D	1.25	1.43	1.60
HP	3293	2774	3589
η	0.620	0.69	0.675

We have assumed that the propeller operates in a wake equal to w = 0.035. We may assume that the thrust deduction can vary from t = 0.015 to t = 0.020 and that the relative rotative efficiency η_r = 0.97 - 1.01 depending on the ship. Applying data from the figures 2, 3 and 4 for the appendix resistance we get:

V_S	35	42	53
η	0.620	0.690	0.675
η_r	0.97-1.01	0.97-1.01	0.97-1.01
η_h	1.015-1.021	1.015-1.021	1.015-1.021
η_{app}	0.045	0.060	0.120
$(1-\eta_{app})\eta \cdot \eta_r \cdot \eta_h$	0.583-0.611	0.638-0.669	0.585-0.613

This shows that the z drive will require approx. 5-10% less power than a conventional propeller with inclined shaft strut and rudder. The propulsive effect of the rudder is included in η_r and t.

If strut and body are not carefully designed cavitation may restrict the application of the z drive. The most critical part seems to be the choice of strut profile and its thickness ratio. If cavitation occurs on the profile, this will have an effect both on the sideforce and the drag This is clear from fig. 9 and fig. 10 as well as from fig 11. The loss of side force is easy to understand from fig. 12 from [11]. It is seen that a suction peak will be present near the leading edge of the flap. At high speeds and large flap angles this will increase the cavitation on the suction side of the foil and gave a loss of side force as observed during the tests. However at high speeds the need for large side force is limited. In addition the required force can be obtained by adjusting the flap angle sligthly. The cavitation on the strut started at 53 knots at approx. 60% of the chord length. The thickness chord length ratio was 13,57% and the profile a NACA 16 profile. The pressure distribution for such a profile is given in fig. 13. Fig. 16 gives the minimum pressure for NACA 16 profiles as a function of maximum thickness/chordlength ratio. This figure indicates that cavitation will start at 66 knots if t/c = 0.10. To reduce the thickness down to this value is no problem.

For propulsion of hydrofoil crafts the type of z drive shown on fig 38 which was tested at Marintek have been applied with success many times. Powers up to 5000 - 6000 HP have been transferred on 1 or 2 shafts within the strut.

When a propeller operates behind the body and the foil, the downwash from the foil will influence on the propeller characteristica just as for inclined flow (see fig. 43). Model tests with such drives have shown that the propeller will operate in a wake of approx 0.05 - 0.07 and that η_r will vary from 0.97 to l,OO.

3. WATERJET PROPULSION

Many of the problems mentioned in connection with conventionel propellers like:

a. Appendix drag
b. Rudder cavitation
c. Vibration and noise due to cavitation and shaft inclination
d. Restricted maneouverability in calm water

seems to be eliminated with waterjet propulsion, which has been applied with success on many ships not only on high speed craft. The waterjet has been reported to give higher efficiency than conventional propulsion down to speeds as low as 35-38 knots for excample on catamarans. It is evident that the waterjet has many fans not at least because of its low noise level. It is also evident that there are some problems connected to waterjet propulsion.

The efficiency of a waterjet is very dependent on the design of the inlet and the losses connected to the inlet. In order to understand how a waterjet works we refere to fig. 17 where the total energy of a cross section is given as:

$$e_i = \frac{1}{\rho Q} \int (\tfrac{1}{2}\rho V_n^2 + P_n + \rho g h_o) V_n dA_n$$

P_n = pressure in the cross section

V_n = velocity of the flow in the cross section

The volume flow Q is defiend as:

$$Q = \int_{A_n} V_n \, dA \,.$$

We may definedloss factors in accordance with [1] where the inlet loss between station 0 and 1 in fig. 17 is.

$$C_{01} = \frac{\Delta e01}{e_o} = \frac{2g\Delta h_{01}}{V_o^2}$$

The loss in the bend between station 3 and 4 is defined in a similear way:

$$C_{34} = \frac{\Delta e_{34}}{eo} = \frac{2g\Delta h_{34}}{V_0^2}$$

The outlet loss is defined using the outlet velocity as a reference:

$$C_{45} = \frac{\Delta e_{45}}{e_5} = \frac{2g\Delta h_{45}}{V_j^2}$$

Normal loss factors for the type of waterjet shown in fig. 17 are:

C_{01} = 0.20 - 0.30

C_{34} = 0.02 - 0.04

C_{45} = 0.02 - 0.04

Assuming 0 pressure gradient in the inflow we may according to [1] define the following wake factors:

$$wfe = \frac{1}{V_0^2 \cdot Q} \int_A v^* \cdot (V_0^2 - V_*^2)dA$$

$$wfi = \frac{1}{V_0 \cdot Q} \int_A v^* \cdot (V_0 - V_*)dA$$

where the velocity V_* in the boundary layer is defiend as:

$$v^* = V_0 \left(\frac{y}{\delta}\right)^{1/7} \qquad \text{(See fig. 17)}$$

The boundary layer thickness is:

$$\delta = 0.37 \cdot \cdot \cdot (R_n)^{-0.2}$$

The total head may then be expressed as:

$$H = \frac{V_j^2}{2g}(1 + C_{45}) - \frac{V_0^2}{2g}[1 - (w_{fe} + C_{01} + C_{34})] + h_0$$

The thrust is:

$$T = \rho \cdot \cdot \cdot [V_j - V_0(1 - w_{fi})]$$

where

$$V_j = \frac{1}{Q}\int_A V_5^2 \, dA$$

The pump efficiency is defined as:

$$\eta_p = \frac{\rho \cdot g \cdot Q}{P_S} \cdot \rho$$

where Ps is the power supplied to the pump. The total efficiency defined as:

$$\eta_{tot} = \frac{T \cdot V_o}{P_S}$$

which gives:

$$\frac{\eta_T}{\eta_p} = \frac{2 \mu [1 - \mu (1 - W_{fi})]}{1 + C_{45} - \mu^2 [1 - (W_{fe} + C_{01} + C_{34} + 2gh_o/V_o^2)]}$$

$$\mu = \frac{V_o}{V_j}$$

The optimum ratio is obtained for:

$$\mu_{opt} = \frac{(1 - W_f)(1 + C_{45})}{1 - (W_f + C_{01} + C_{34} + \frac{2gho}{V_o^2})} [1 - [1 - \frac{1-(W_{fe}^2 + C_{03} + C_{34} + \frac{2gho}{V_o^2})}{(1-W_{fi})^2(1+C_{45})}]^{0.5}]$$

Figure 18 from [1] gives η_T/η_p for different μ values.

The power supplied to the pump is:

$$P_S = \eta_f \int_{30}^{R_j} R_1 \, 2\pi n \cdot \rho \cdot 2\pi r_3^2 \cdot W_3 (r_3) \cdot V_3 (r_3) \cdot dr_3$$

where η_f is a correction for frictional and other losses. n is the number of revolutions for the propeller.

In order to estimate P_S and T we must find the relationship between V3 (r) and W3 (r).

Bernoul lis equation gives:

$$p_3(r) = p_5 + \rho/2 \, V_j^2 - \rho/2 \, W_3^2 (r) - \rho/2 \, V_3^2 (r)$$

Due to the rotation and curved flow through position 3 we get:

$$\frac{dp_3}{dr} = \rho \left(\frac{w_3^2 (r)}{r} + \frac{V_3^2 (r)}{r^*} \right)$$

where r* is the radius of curvature for the flow in axial direction. Differentiation of $p_3(r)$ and the equation given above is the basis for estimation of $V_3(r) = f(W_3(r))$

For simplification we assume that $p_5 = p_0$ and that V_j is independent of the radius. It is further assumed that the flow is free from rotation at station 5.

We then finally obtain the differential equation:

$$\frac{2V_3^2(r)}{r^*} = \frac{d(W_3^2(r))}{dr} = \frac{2(W_3^2(r))}{r}$$

which yields different V3 - W_3 relations. However the principle of conservation of energy gives:

$$75 \cdot \int_{R_{30}}^{R3} (2\pi)\cdot\rho^2 r^2 \cdot W_3(r_3) \cdot V_3(r)dr = H \cdot g \cdot \rho \cdot Q$$

It is now possible to estimate $V_3(r)$ and $W_3(r)$ for a given thrust. Fig. 21 shows output from calculations according to the principles sketched above.

It is important to secure that the total pressure at station 1 or at the inlet to the pump is high enough to avoid cavitation. An important parameter for estimation of cavitation is the difference between the gas pressure pe and the total pressure:

$$Ah = \frac{p_0 - p_e}{\rho \cdot g} + \frac{V_0^2}{2g}[1 - (W_{fe} + C_{01}) + h_0]$$

If we define the suction number

$$n_{qs} = n \cdot \frac{Q^{0.5}}{(\Delta h)^{0.75}}$$

we finally end up with the empirical relationship:

$$\frac{V_1}{V_0} = 0.015\, n_{qs}^{0.666} [1 + \sigma_0 - (W_{fe} + C_{01})]^{0.5}$$

From $D_1 = \sqrt{4Q/\pi}$ the required diffusion is determined. As a criterium for choise of impeller the following parameter is used:

$$n_q = \frac{Q^{0.5}}{H^{0.75}} \cdot n \quad \text{(Spesific speed)}$$

The typical range for radial pumps is $n_q < 20$. The Francis type is in the range $40 < n_q < 80$. For the mixed flow type typical range is: $80 < n_q < 120$. Above 120 the axial pump type is actual. Very often the problem boils down to the choise of the correct RPM in order to avoid serious cavitation problems, which means that we try to keep $n_{qs} < 120$.

For a waterjet with optimum v_j/v_o ratio we have estimated n_{qs} and n_q for different ship speeds and RPMs. We have assumed that HP = 2025. The result is illustrated in fig. 19 and 20. The figures gives an idea about the possibilities for choise of RPM.

The efficiency of the waterjet is as allready mentioned very dependent on the inflow conditions or the loss factor C_o (see figs 18 and 23). In [4] Haglund et al gave some results for ships equipped with waterjets. We have used the theory given above and assumed a pump efficiency equal to η_p = 0.90 which is close to measurements made at KAMEWA. The calculation indicates that loss factors equal to C_o = 0.20-0.25 were obtained for the flush inlets in these cases as pointed out in [4].

The pump efficiency depends very much on the drag of the profiles on the propeller or the impeller as indicated in fig. 22 where we have calculated η_p for C_D = 0.004 and C_D = 0.008. These values are representative for full scale and for model scale. It is indicated that η_p = 0.90 is a reasonable efficiency. In the following examples we have therefore assumed that η_p = 0.90. We have further assumed that η_r = 1.00 and t = 0 which means that only the viscous part of the wake has been considered. In fig. 25 we have estimated the optimum efficiency for different powers and speeds. Corresponding outlet diameters are given. If we deviate from the optimum diameter we obtain results as indicated in the figs 26, 27 and 28.

The most used inlet is the flush type. Strictly speeking a given inlet is only optimum for one condition. If we deviate to much from the design point separation will block the inlet and give bad working conditions for the impeller leading to cavitation and heavy vibrations. It is therefore recommended to study the inflow to the inlets during propulsion tests, in a cavitation tunnel or in a wind tunnel where the flow can be visualized. There are several excamples where model tests have led to improved design. One problem for hydrofoil boats and SES is the take off condition which require high thrust at low speed. In this case the inlet should therefore be efficient at "off design", as indicated in figs. 29 and 30 from [3]. Model tests and full scale experience have shown that air drawing or ventilation may be a problem with flush inlets specially in a seaway and on SESs where the immersion is restricted. Yentilation leads to thrust and torque reduction and may lead to serious "dropouts" under severe weather conditions. One way out of the problem is to apply a scoop of the type shown on fig. 31. This inlet will work well at the design point but for "offdesign " conditions there will be a blocking of the inlet. The inlet loss due to blocking may be expressed as:

$$\Delta C_0 = k \left(\frac{\pi \cdot R_j^2}{F} \right)^2 \cdot \left[\left(\frac{V_s}{V_j} \right)^{-2} - \left(\frac{V_s}{V_j} \right)^{-2}_{opt} \right]$$

where:

F = inlet area of the scoop.

R_j = outlet area of the waterjet

k is a constant depending on the shape of the inlet. The sharper the inlet the higher k will be. It is clear that ΔC_0 will increase if the ship speed drops as the thrust is maintained. For the scoop shown in fig. 31, we have tried to illustrate this for different values of k. We have assumed that the scoop is optimum at 55 knots and that the speed is reduced while the thrust is kept constant. As shown on fig. 13 and 14 the thickness of the strut should not exceed t/c = 0.10 in order to operate cavitation free above 55 knots. The inlet area follows from the required thrust. For the condition given above we have calculated the drag of the scoop. As shown on fig. 32 the drag is considerable and must be considered before such solutions are proposed.

For the following conditions where the z drive was tested:

V_s (knots.)	35	42	53
HP	3293	2774	3589

we have estimated the efficiency of an optimum waterjet for different values of C_0. It was assumed that C34 = C45 = 0.02 and that the pump efficiency was equal to 0.90.

We got the following results:

V_s (knots)	35	42	53
η_{tot} (C_0 = 0.101	0.617	0.627	0.638
η_{tot} (C_0 = 0.151	0.596	0.606	0.614
η_{tot} (C_0 = 0.20)	0.577	0.587	0.594
η_{tot} (C_0 = 0.301	0.548	0.556	0.562

These results have in fig. 33 been compared to the results for the conventional propeller and the Z drive. The propellers were optimized for 42 knots.

We did not for the water jet consider the thrust deduction factor (the additional drag due to the installation of the waterjet). This factor is normally very small and may even reach negative values. At reduced speed or power, values up to $t = 0.05$ are possible. The wake factors will be of the order of $W = 0.05$ depending on the shape of the afterbody. In our excamples we have included the frictional wake which is relatively small. This means that the values given above for the waterjet should be multiplied by the hull efficiency:

$$\eta_h = \frac{1 - t}{1 - W}$$

ranging from 1.02 to 1.05 depending on the shape of the inlet and the afterbody. We assumed that $\eta_p = 0.90$. Calculations and model tests indicate that the full scale value may be higher. It is not relevant to consider this in a comparison with conventional propellers where scale effects also are present.

4. PARTLY SUBMERGED AND VENTILATED PROPELLERS

If a propeller works near the surface or partly out of the water the efficiency may still be acceptable. The following effects are present when the propeller is ventilated and out of water.

1. The surface is disturbed and an axial velocity is induced which decreases the thrust and the torque.

2. The out of water effect reduces the thrust proportionally with the area of the propeller disk which is out of the water.

3. The suction side of the propeller is vented and the theoretical lift or low pressure can not be maintained.

4. The ventilation will reduce the torque but to a smaller extent that the thrust.

There are several papers on ventilated and partly submerged propellers [2], [7], [8], [9]. The figures 35, 36 and 37 which are taken from [9] show results from open water tests with a 4 bladed propeller tested at different immersions h of the propeller shaft and at different revolutions. The propeller had the follwing main characteristics.

Number of blades z	= 4
EAR	= 0.80
P/D	= 1.5

An important parameter for the ventilated and the partly submerged propeller is the Froude number of the propeller defined as:

$$F_n^2 = \frac{(n\ D)^2}{g\ h}$$

This number and the advance number J_A must be equal for ship and model. Most of the data which has been published are from tests where F_n^2 is far below the full scale value for high speed craft and are therefore not representative. They give too optimistic prognoses. The full scale propeller operate partly cavitating and partly ventilated. Brandt et al [8] studied this combined effect. But it is still not fully understood to what extent the ventilation will dominate over the cavitation or surpress cavitation. Let us apply the propeller mentioned above in an excample:

Assume that the follwing thrust is required:

V_S (knots)	35	41	50
T kg)	3384	2820	2187

and that RPM = 1000 and 1250 HP is available for propulsion. The thrust loading is defined as:

$$C_T = \frac{T}{p/2 \cdot V_A^2 \cdot \pi R^2}$$

For different propeller diameters and h/D = 0.25 we get:

V_S = 35 knots

D (m)	1.0	1.2	1.5
J	1.085'	0.900	0.724
C_T	0.254	0.1-77	0.113
F_n^2	113	135	170

V_S = 42 knots

D	1.0	1.2	1.5
J	1.265	1.055	0.843
C_T	0.1545	0.1074	0.0687
F_n	113	135	170

V_S = 50 knots

D	1.0	1.2	1.5
J	1.544	1.287	1.029
C_T	0.0806	0.0560	0.0358
F_n^2	113	135	170

In fig. 34 we have plotted the efficiency taken from the figures 35, 36 and 37 as a function of F_n^2. Similiar plots based on data in [2], [7], [8] and [9] give the same trend. From this it is temting to draw the conclusion that the efficiency of a partly submerged or ventilated propeller can not compete with the efficiency of a waterjet or a conventional propeller.

However, successfull application of partly submerged propellers combined with a tunnel as shown on fig. 41 have been reported. Full scale tests indicate essential improvements.

5. PROPULSION IN A SEAWAY

In a seaway we may experience air drawing or ventilation due to large relative motions between the propeller and the surface which may lead to "drop out" or may reduce the thrust considerably. The characteristics of a propeller in a seaway is very much dependant on the propeller loading and the initial immersion h_0 to the propeller center, the amplitudes of the relative motions and their periods. This is illustrated in figure 42 where results are shown for a four bladed propeller tested at Marintek in regular waves at two different immersions h/R = 1.0 and 1.5. The pitch of the propeller was:

$$P/D = 0.931$$

For a period equal to $T_0 = \infty$ the propeller characteristics for a similiar propeller is shown in figure 39 as a function of submergence and number of revolutions. The figure is taken from [2].

The important factor when a propeller operates near the surface is as mentioned in section 3 the Froude number. For the propeller shown on the figure the highest value for $T_p/D = 1.0$ is:

$$F_n^2 = \frac{(0.5 \cdot 7.1)}{9.81 \cdot 0.5} = 2.569$$

In full scale we may expect values from 15 to 30, which indicates that the thrustloss in full scale will be much larger than shown on the figures. F_n^2 in this case is $F_n^2 = 4.587$. Let us as an excample assume a SES operating in towing condition or suffering from severe speed loss (fig. 40):

The ship operates at V_s = 5.5 knots, RPM = 625, P/D = 0.90 with a propeller having unsufficient submergence ($T_p/D = 1$). The diameter is equal to 1.25 m. Under these assumptions we get $J = 0.217$ and $F_n^2 = 13.82$. It is evident that the risk for thrust loss is high and that this condition will be critical. It is also evident that the situation is improved considerable if the submergence can be increased from $h_0/R = 1$ to ho/R = 1.5 which is possible with a z drive.

We have data for a waterjet in a similiar situation and have reasons to believe that air drawing should be studied seriously in such cases for excample through model tests in waves.

6. CONCLUSIONS

1. Z drives of the tractor type with a flap on the strut to give side force seems to be an interesting alternative to propulsion with water jet on catamarans and SES for speeds up to 55 knots.

2. The efficiency of the z drive is higher than for conventional propulsion mainly due to elimintation of appendix drag and due to improved propulsive efficiency.

3. The z drive with a flap on the strut is an excellent rudder. For very high speeds cavitation will reduce side force and efficiency if the strut is too thick.

4. Waterjet propulsion is an atractive alternative to conventional propulsion if air drawing can be avoided and the inlet be designed to give minimum inlet losses at the design condition and acceptable losses. at off design. At speeds above 55-60 knots the waterjet is superior.

5. Ventilation can be a problem for ships operating with reduced speed and high propeller loading in a seaway if the immersion of the propeller shaft or inlet is restricted.

6. The z drive may improve the propulsive efficiency of catamarans and SES in a seaway due to sufficient immersion of the propeller.

7. Modeltesting is a vital part of the design procedure for conventional propellers, z drives and waterjets as well. Special attention should be paid to propulsive efficiency in a seaway, design of waterjet inlets and to cavitation and ventilation problems.

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8. LIST OF SYMBOLS

EAR	Expanded blade area ratio
C	Chord length
D	Propeller (impeller) diameter
r	Impeller or propeller radius
$J = V_A/nD$	Advance coefficient
n	Number of revolution
RPM	Revolutions pr. minute
V_j	Jetvelocity
"A	Speed of advance of propeller
V_s	Ship speed
V_o	Ship speed (when equipped with waterjet)
Z	Number of blades
T	Thrust
Q	Propeller torque
$K_T = \frac{T}{\rho n^2 D^4}$	Thrust coefficient
$K_Q = \frac{Q}{\rho n^2 D^4}$	Torque coefficient
P	Pressure
Pe	Vapour pressure
h_o	Static head
h_o	Immersion of propeller center
T_p	Immersion of propeller center
TP	Propeller thrust
HP (or PS)	Power delivered to the propeller
η	Efficiency
ν	Coefficient of kinematic viscosity
ρ	Mass density
g	Mass acceleration
σ	Cavitation number
H	Required head
Q	Volume flow
Fn	Froude number
P/D	Pitch diameter ratio
$C_T = \frac{T}{\rho/2 \cdot V_A^2 \cdot \pi/4 \cdot D^2}$	Thrust coefficient

C_a	Inducd axial velocity by the propeller
ω	Angular velocity
t	Thrust deduction
n_q	Specific speed
n_{qs}	Suction number
t	Max thickness of the profile
D_i	Impeller diameter
D_j	Jet or outlet diameter
ω	Wake fraction

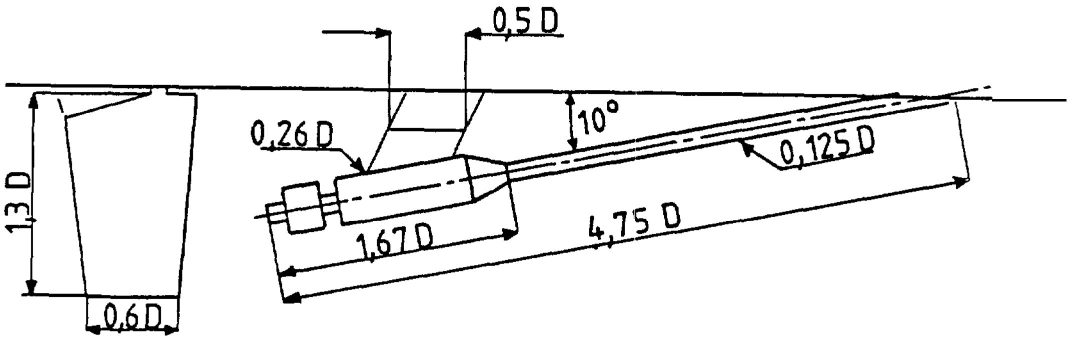


Fig. 1. Dimensions of shaft bracket and rudder used in the example.

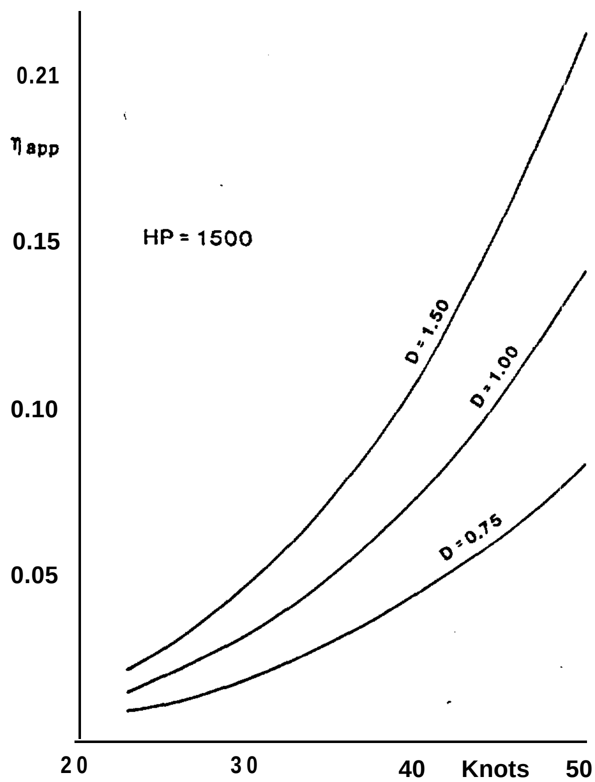


Fig. 2. Appendix drag.

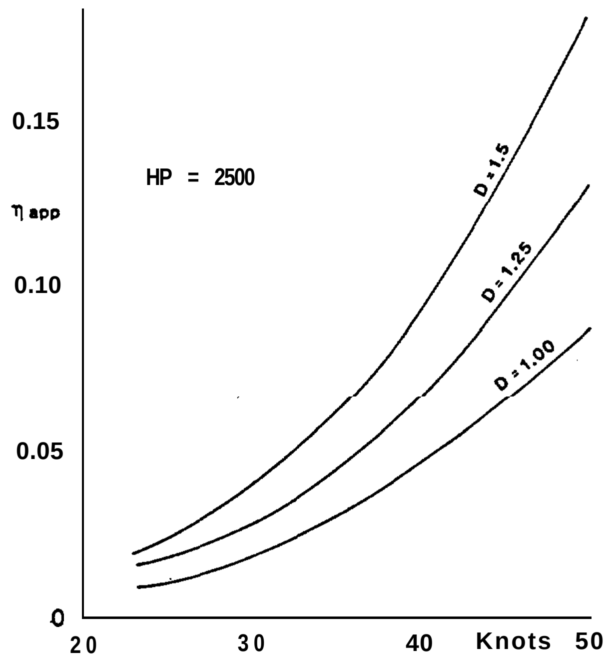


Fig. 3. Appendix drag.

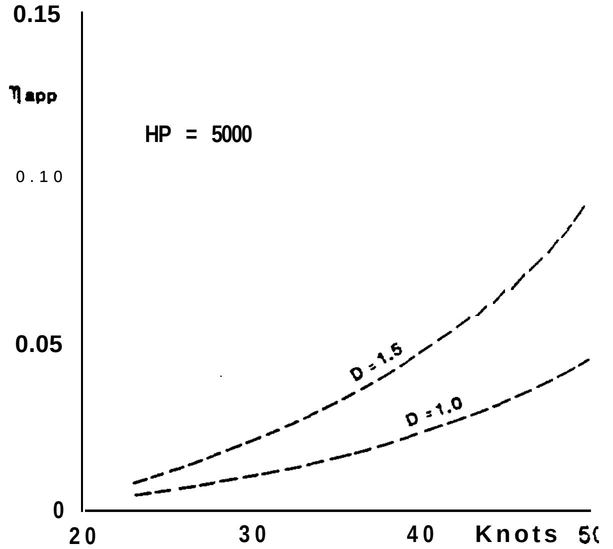


Fig. 4. Appendix drag.

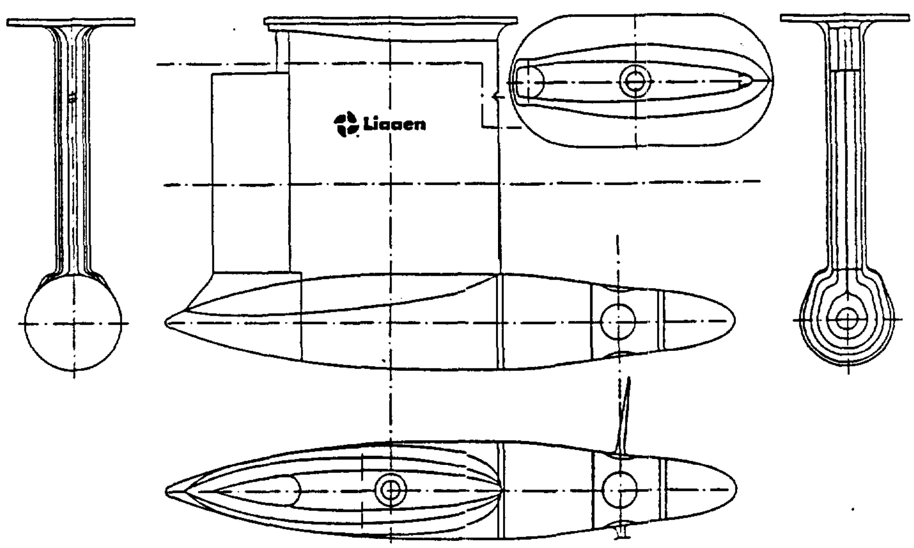


Fig. 5. Z drive for propulsion of SES.

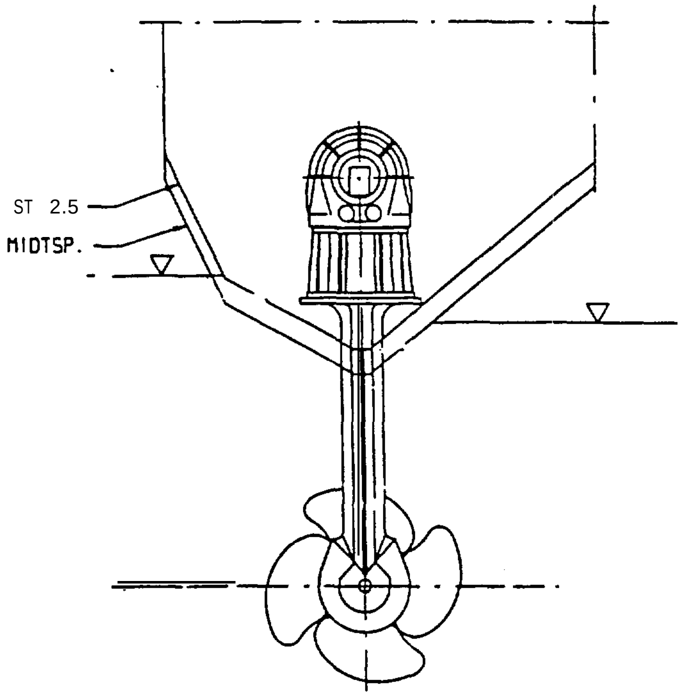


Fig. 6. Arrangement of Z drive.

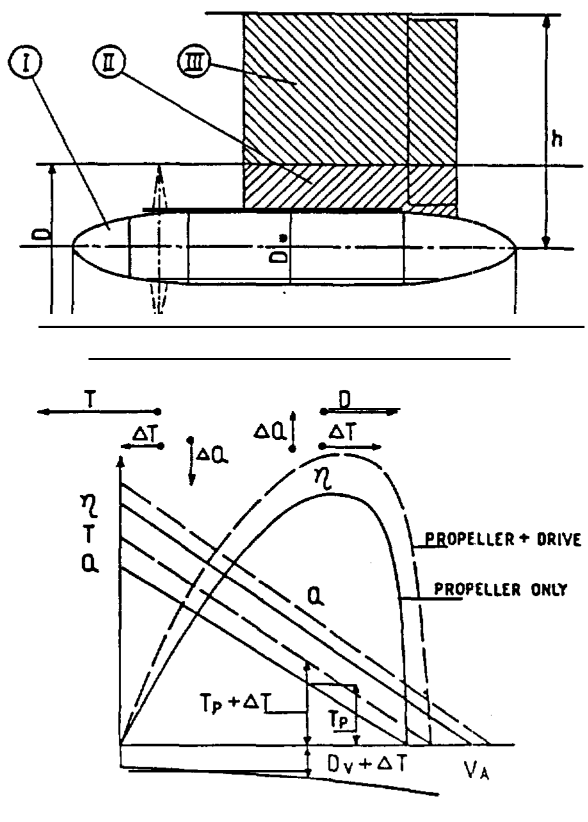


Fig. 7. Interference between propeller strut and body.

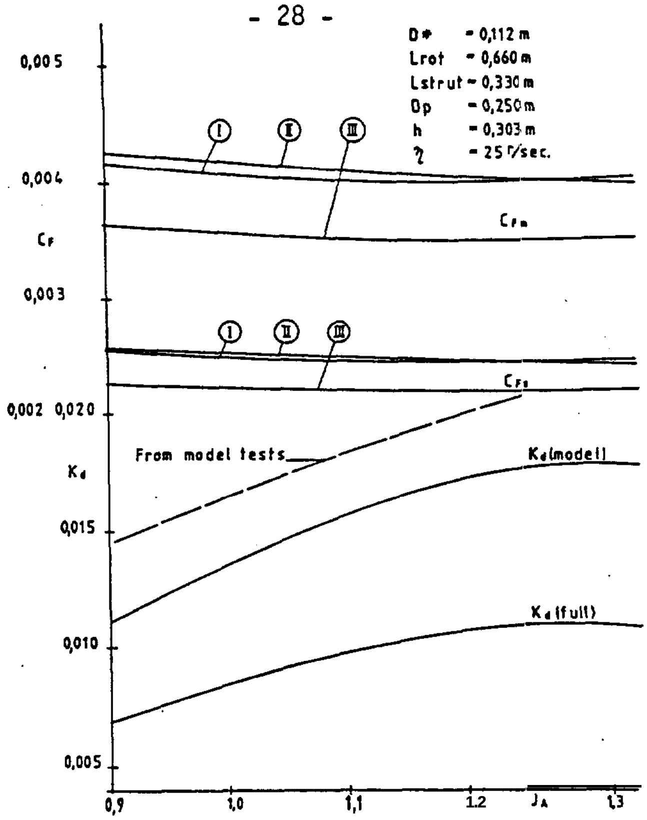


Fig. 8. Viscous drag of strut and body.

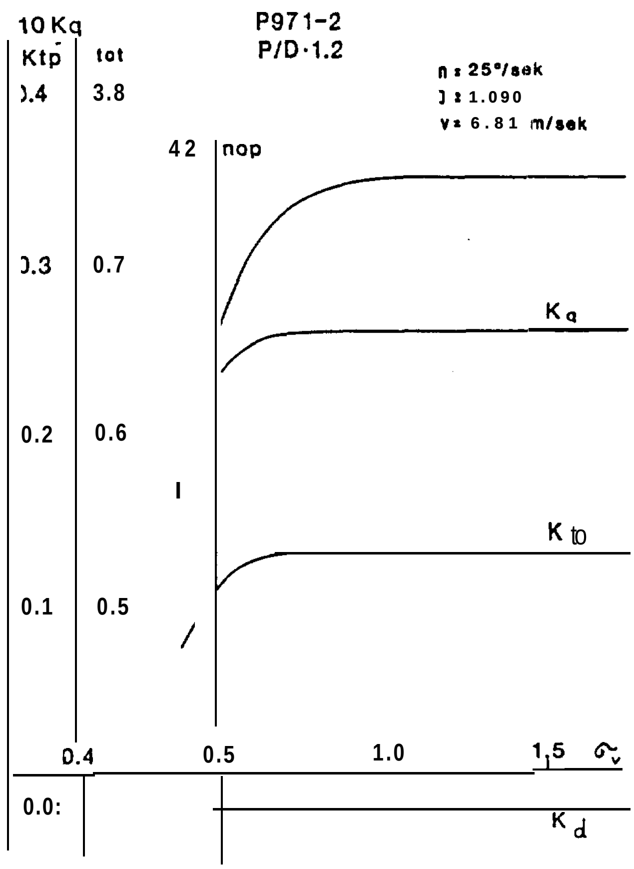


Fig. 9. Thrust, torque and efficiency of the Z drive.

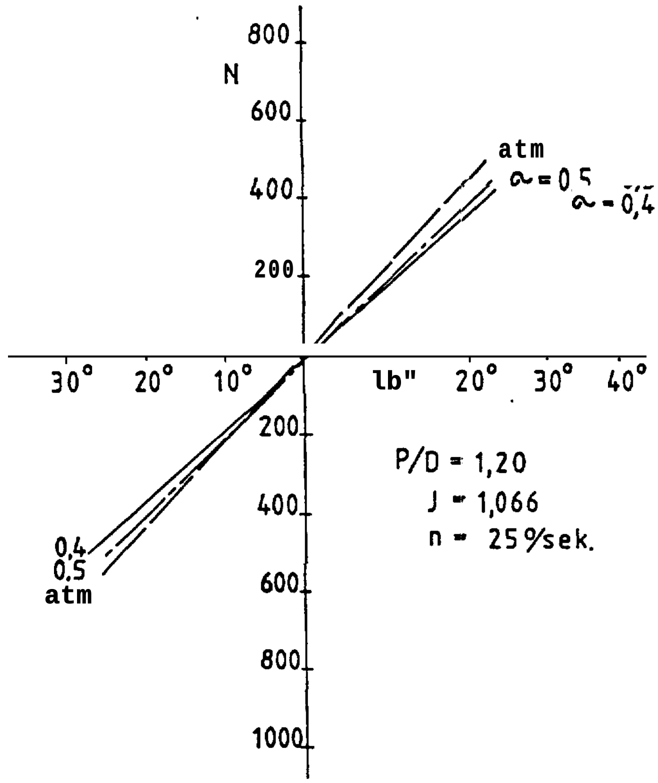


Fig. 10. Side force on the Z drive as function of flap angle.

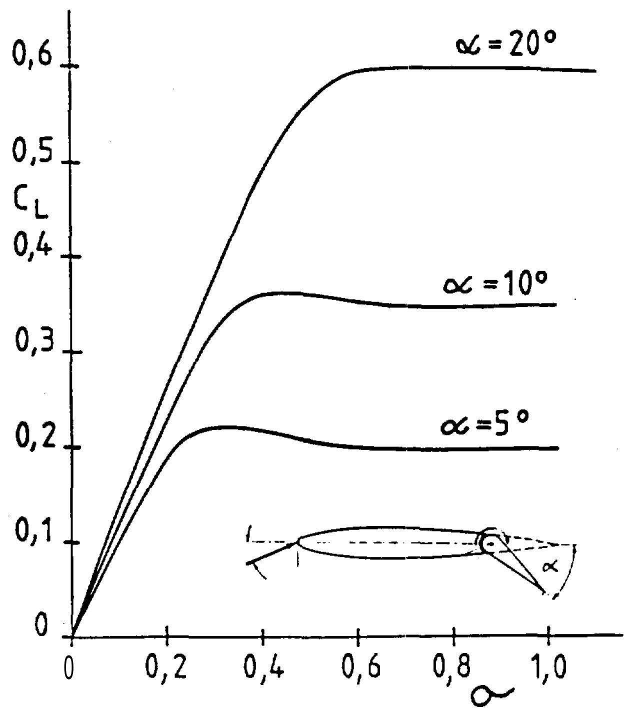


Fig. 11. Reduction of lift due to cavitation.

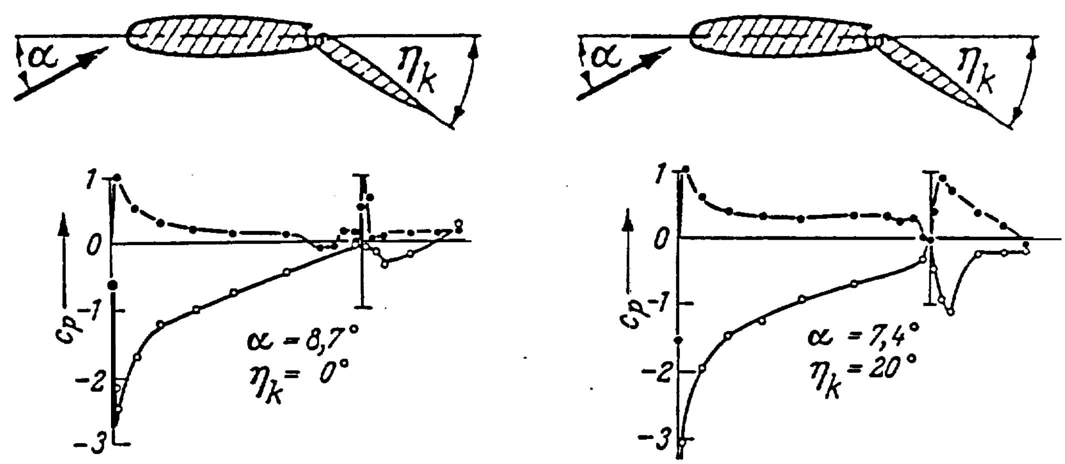


Fig. 12. Influence of flap angle on pressure distributions.

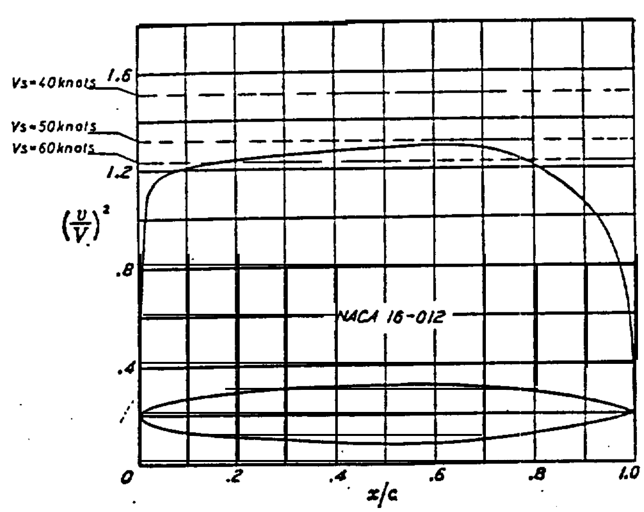


Fig. 13. Pressure distribution along a NACA 16 profile.

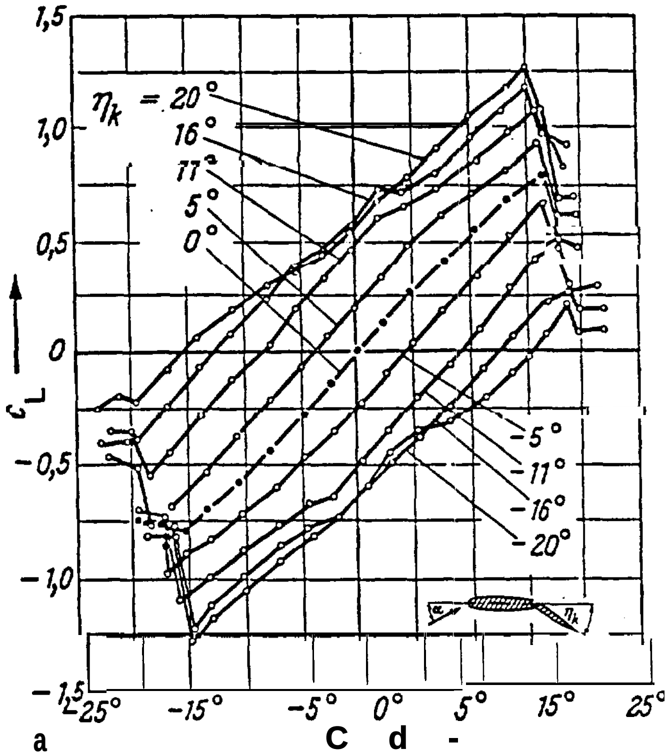


Fig. 14. Lift due to use of flap.

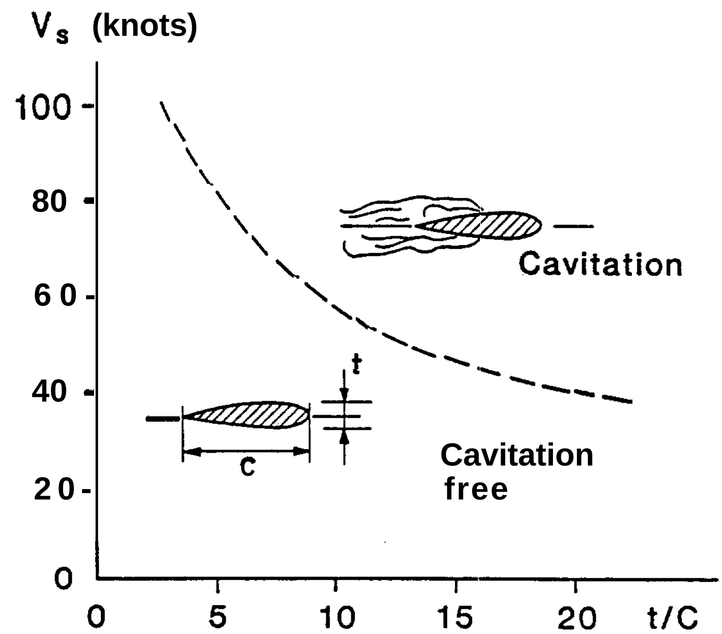


Fig. 16. Incipient cavitation. .

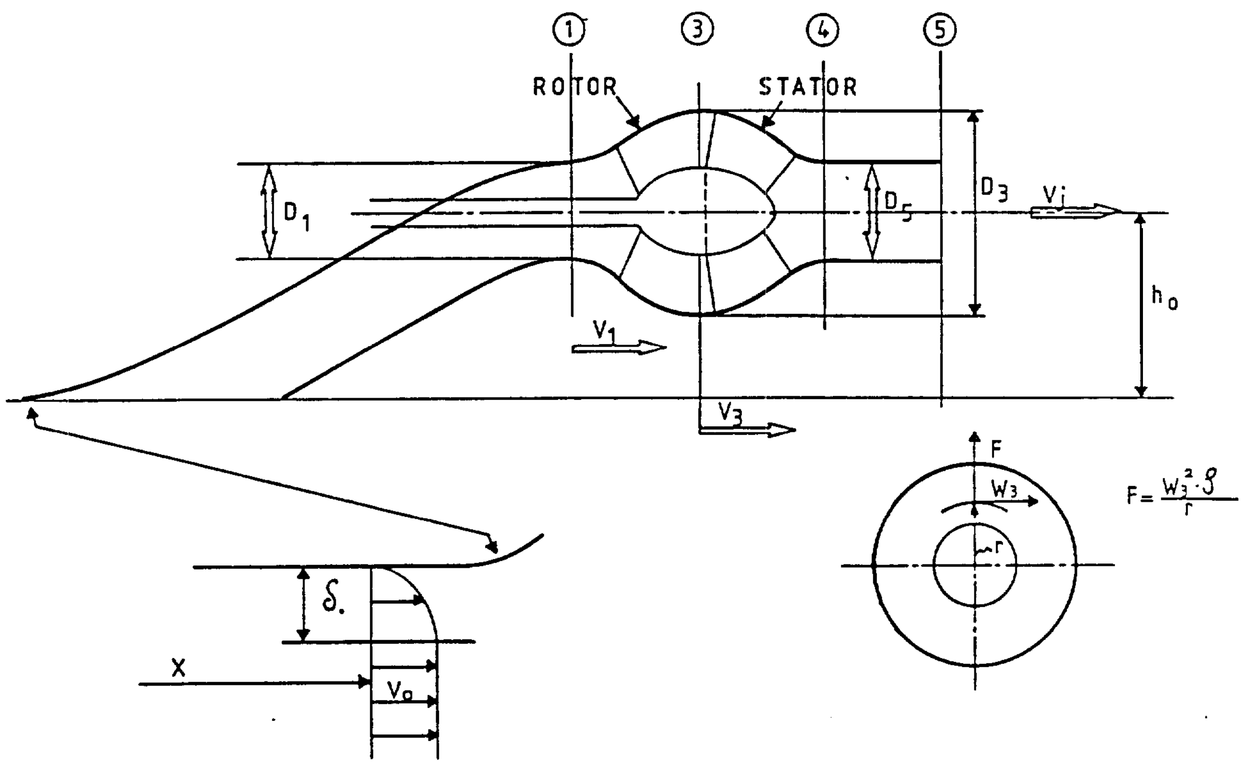


Fig. 17. Water-jet definitions.

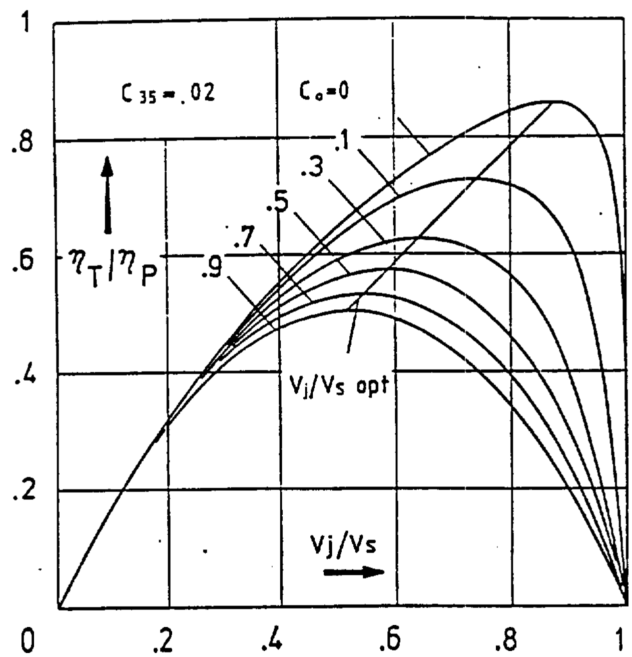


Fig. 18. Optimum jet efficiency.

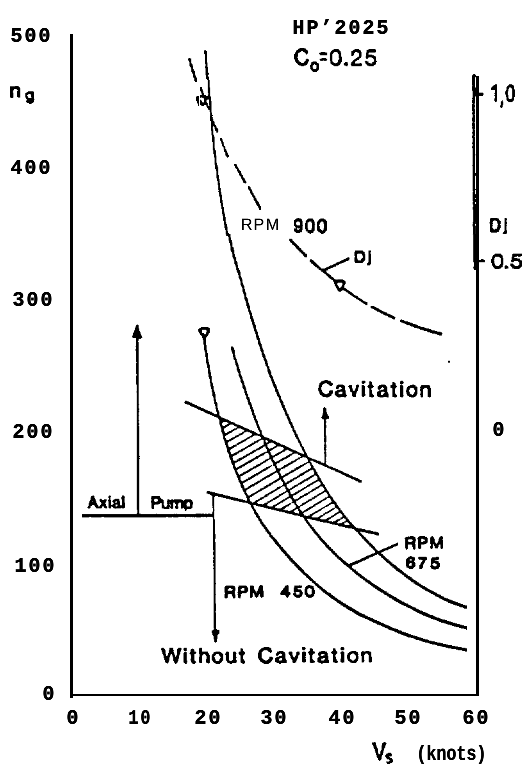


Fig. 19. Specific speed.

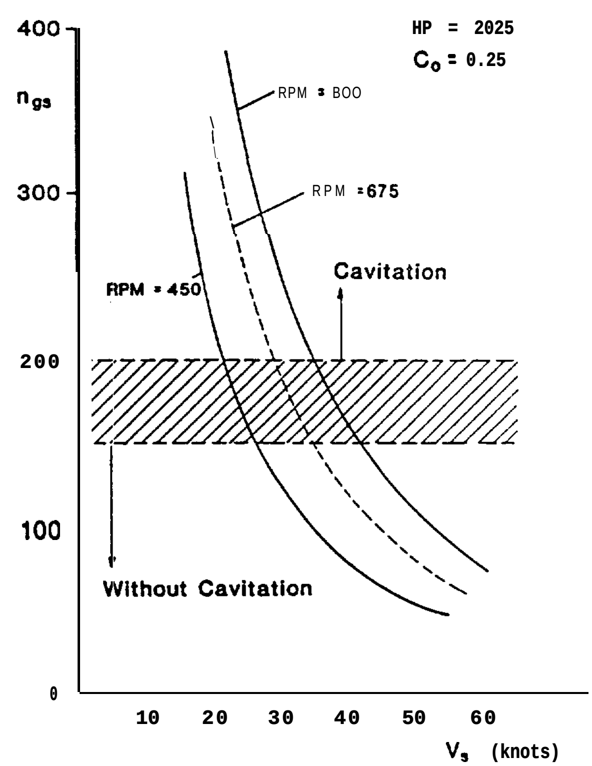


Fig. 20. Suction number.

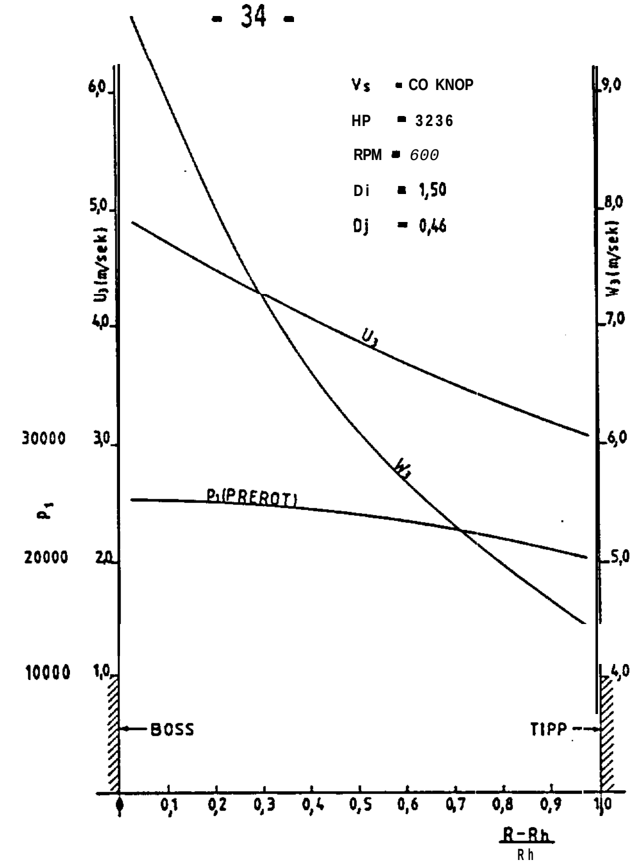
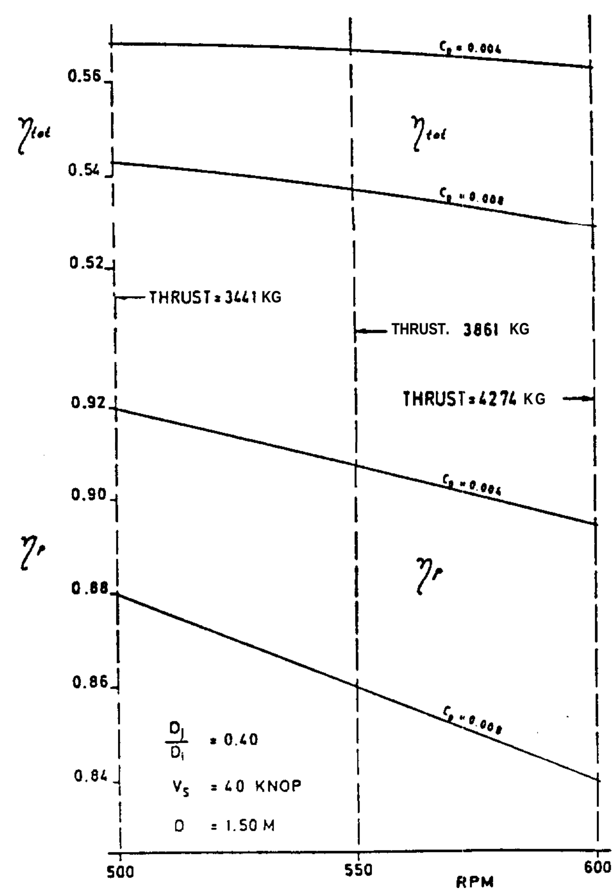


Fig. 21. Static pressure at the pump inlet and velocities at the impeller outlet.



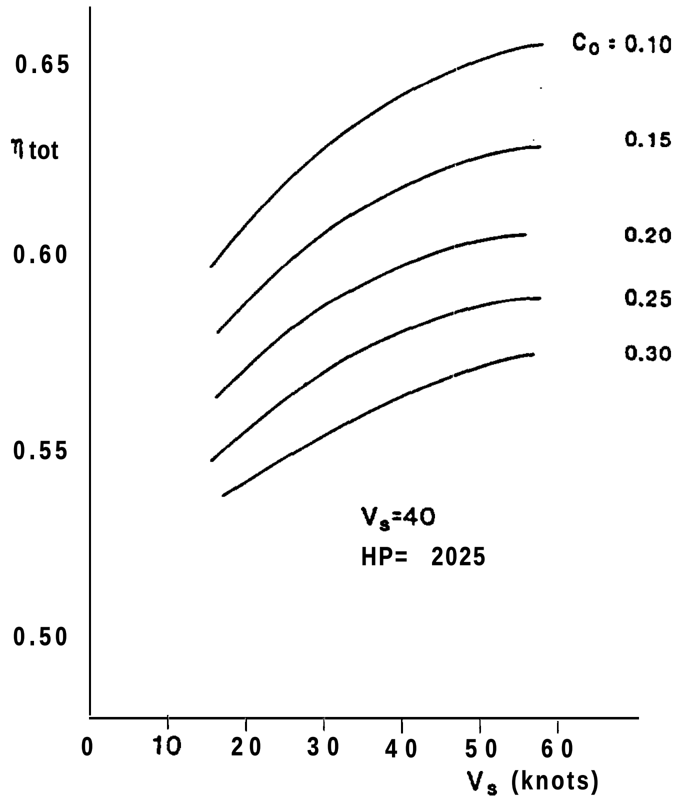


Fig. 23. Influence of inlet loss on total efficiency.

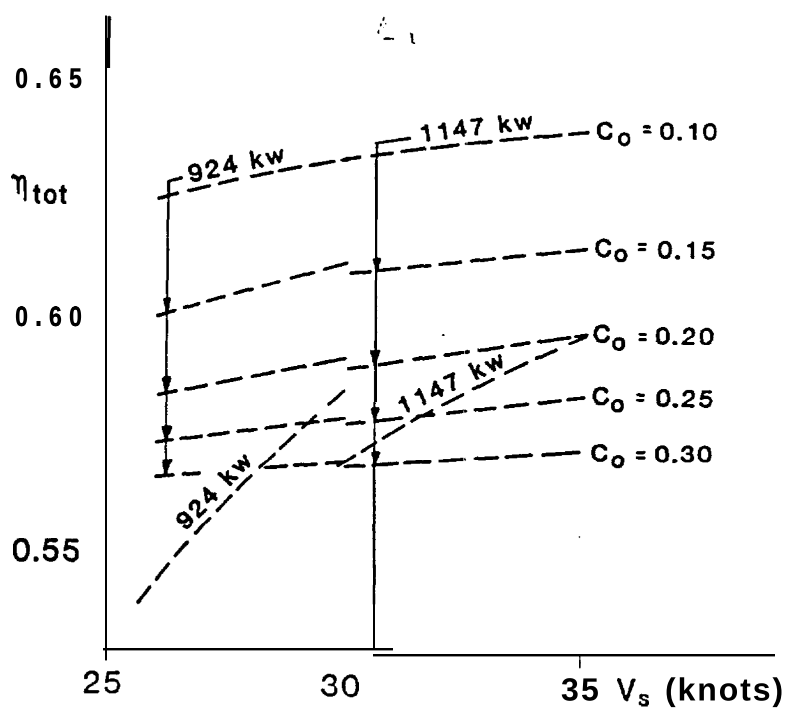


Fig. 24. Full scale efficiency - inlet loss.

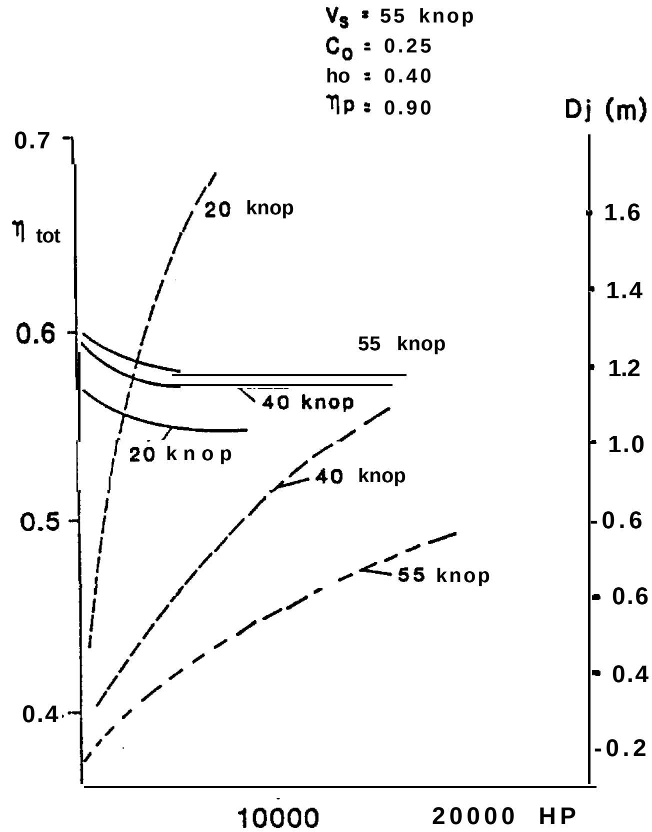


Fig 25. Optimum efficiency,

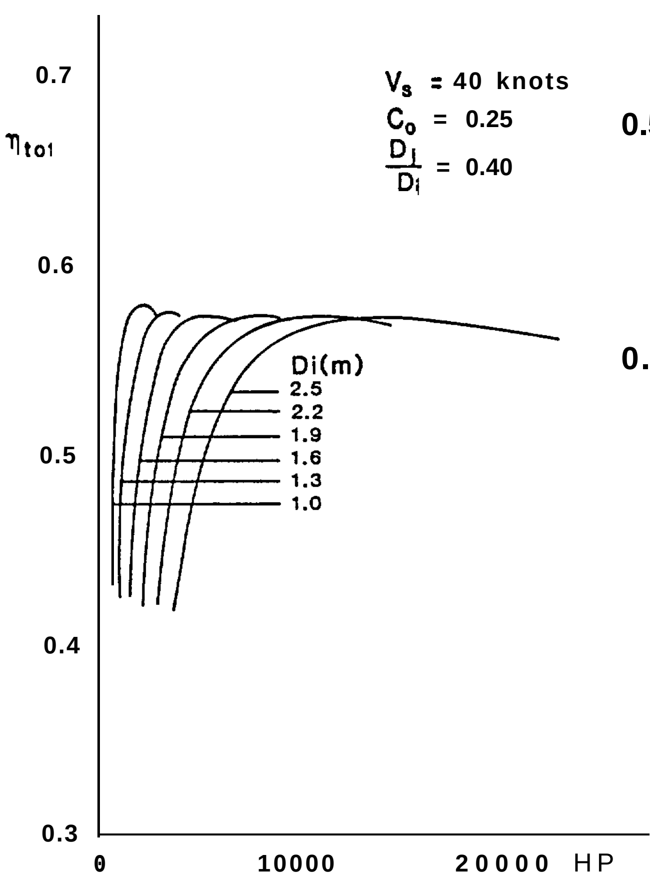


Fig. 27. Total efficiency.

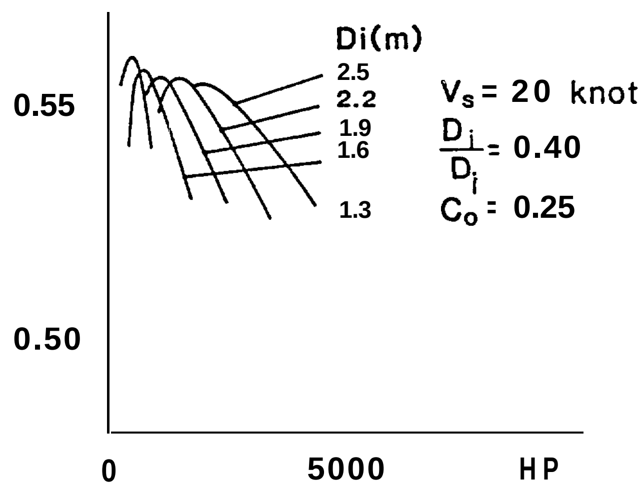


Fig. 26. Total efficiency.

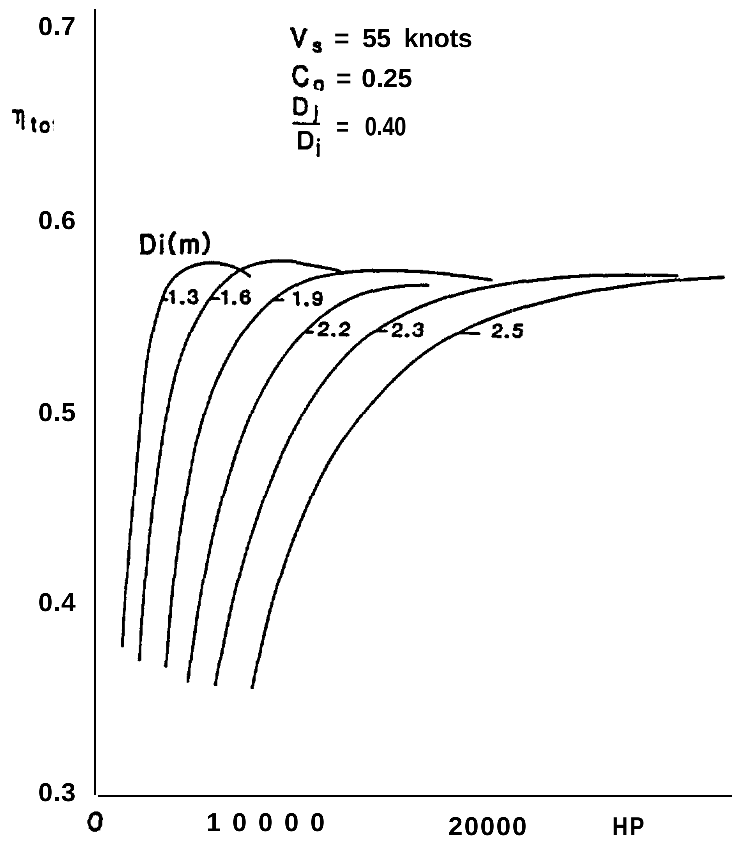


Fig. 28. Total efficiency.

Fig. 29a. Baseline flush inlet model.

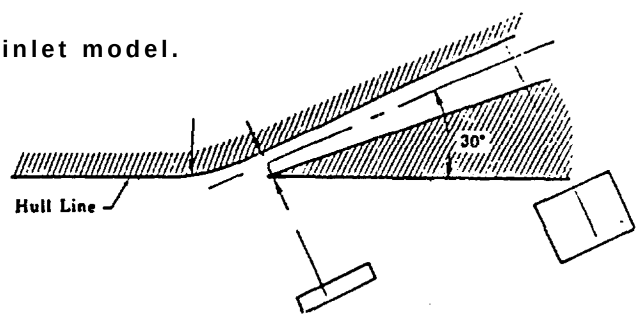
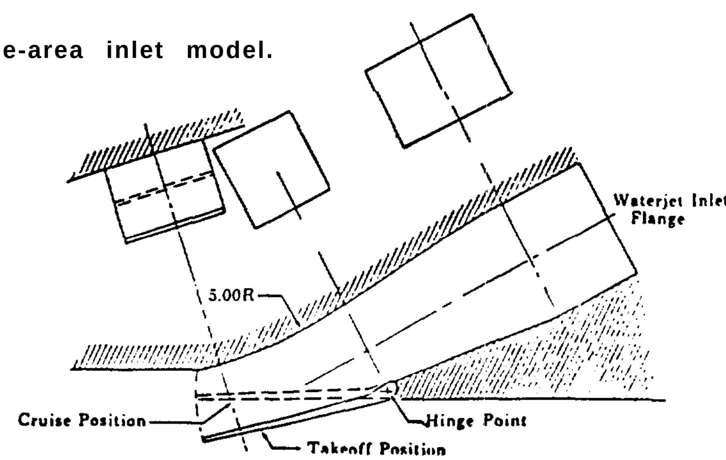


Fig. 29b. Variable-area inlet model.



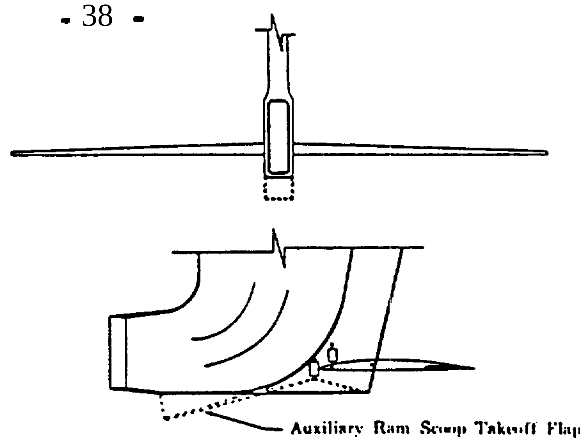


Fig. 30a. Minimum-drag vertical rectangular inlet with takeoff flap.

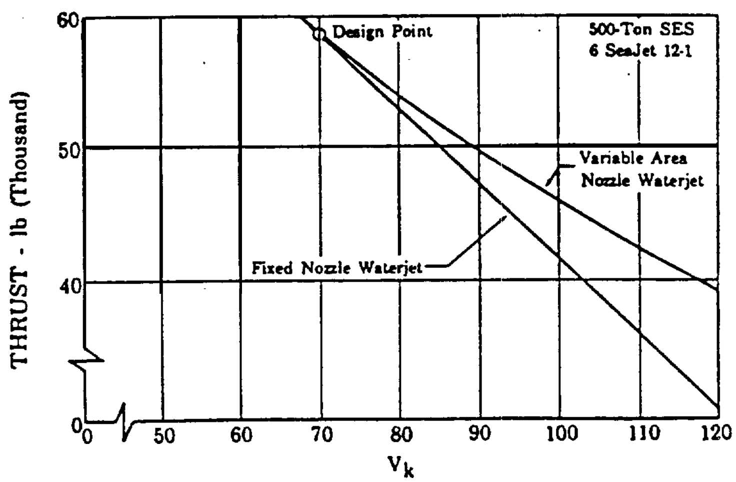


Fig. 30b. Fixed-nozzle versus variable-nozzle performance.

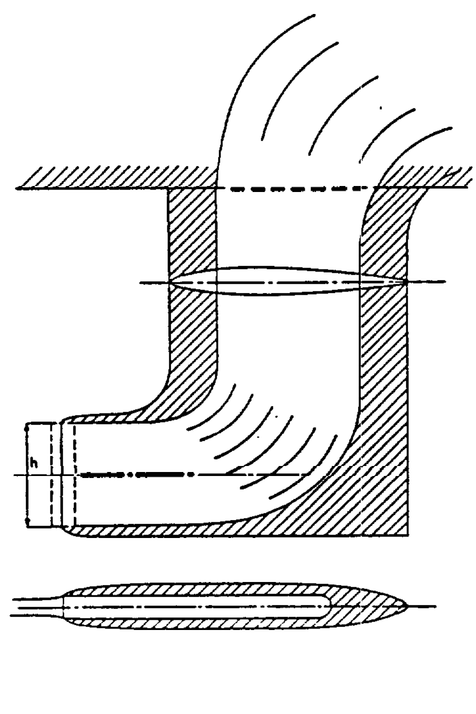


Fig. 31. Scoop inlet.

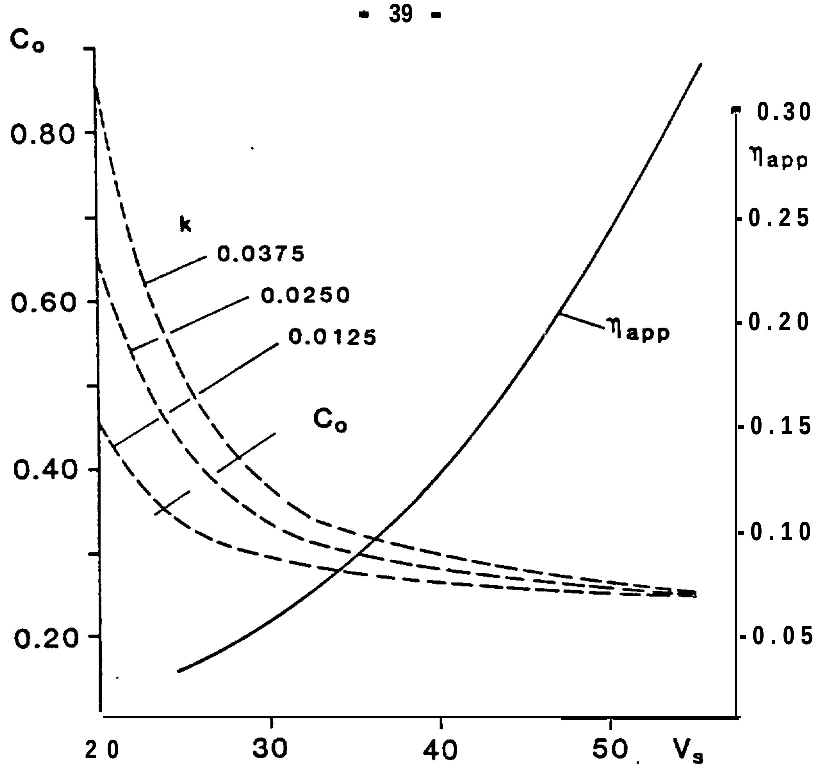


Fig. 32. HP=1000 at V_S=55 knots, constant thrust.

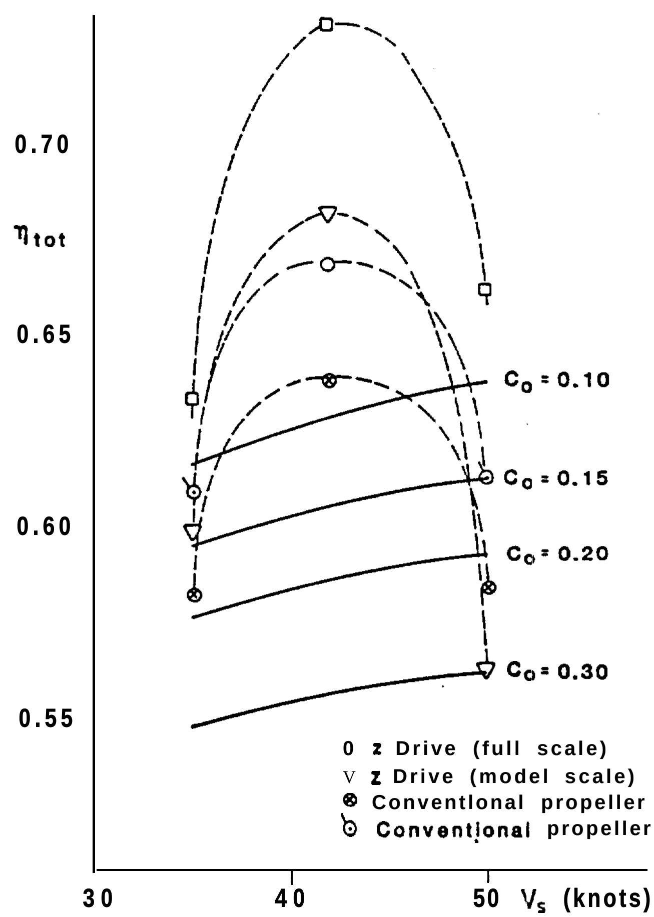


Fig. 33. Efficiencies for Z drive, water-jet and conventional propellers.

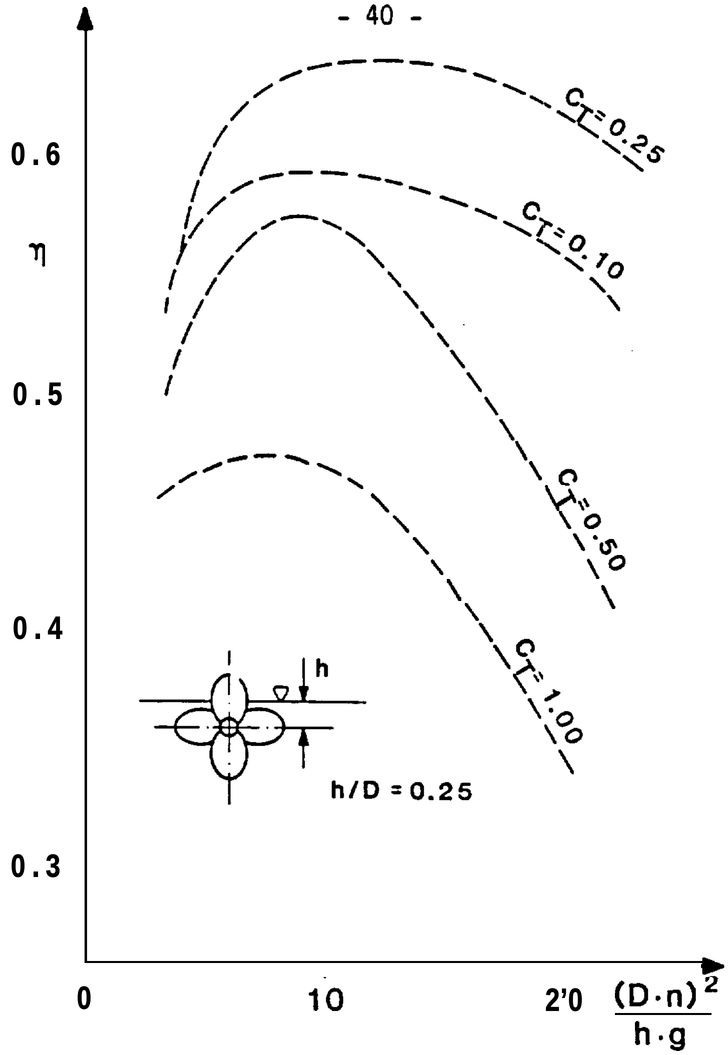


Fig. 34. Efficiency of a partially submerged propeller.

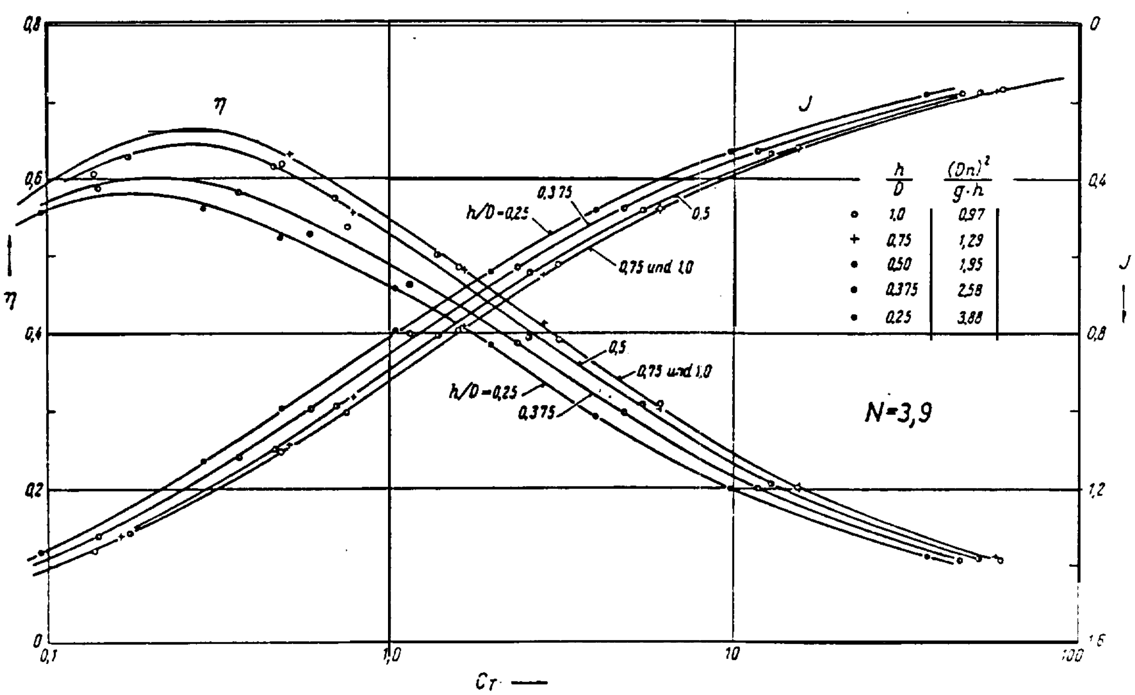


Fig. 35. Characteristics of a partially submerged propeller.

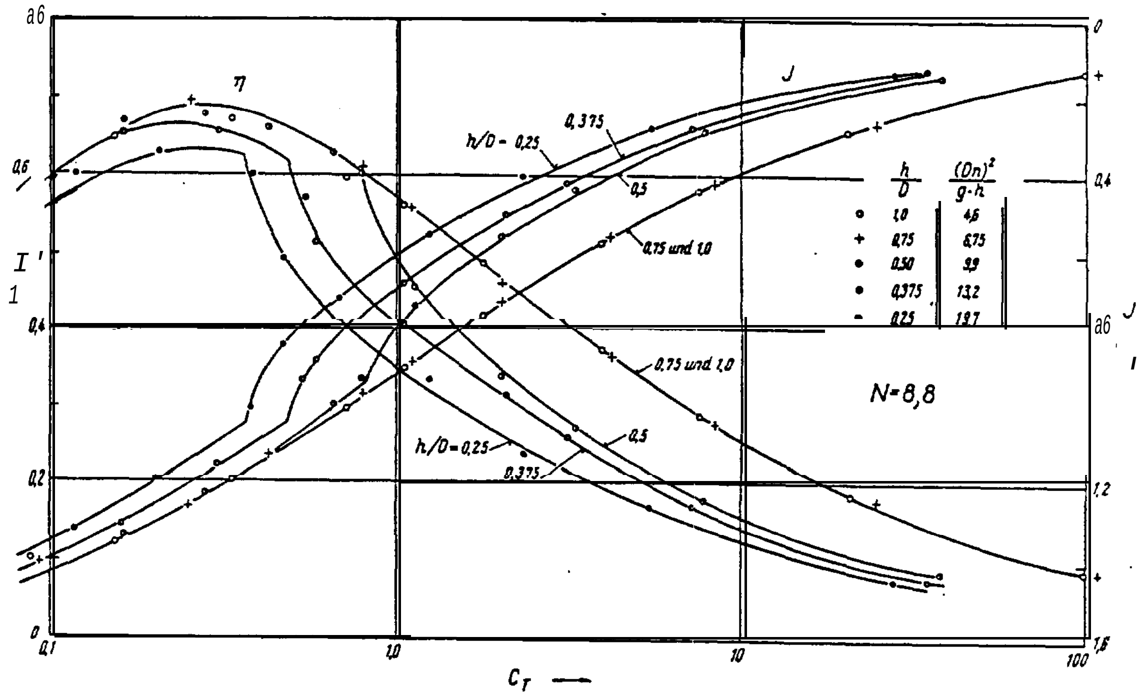


Fig. 36. Characteristics of a partially submerged propeller.

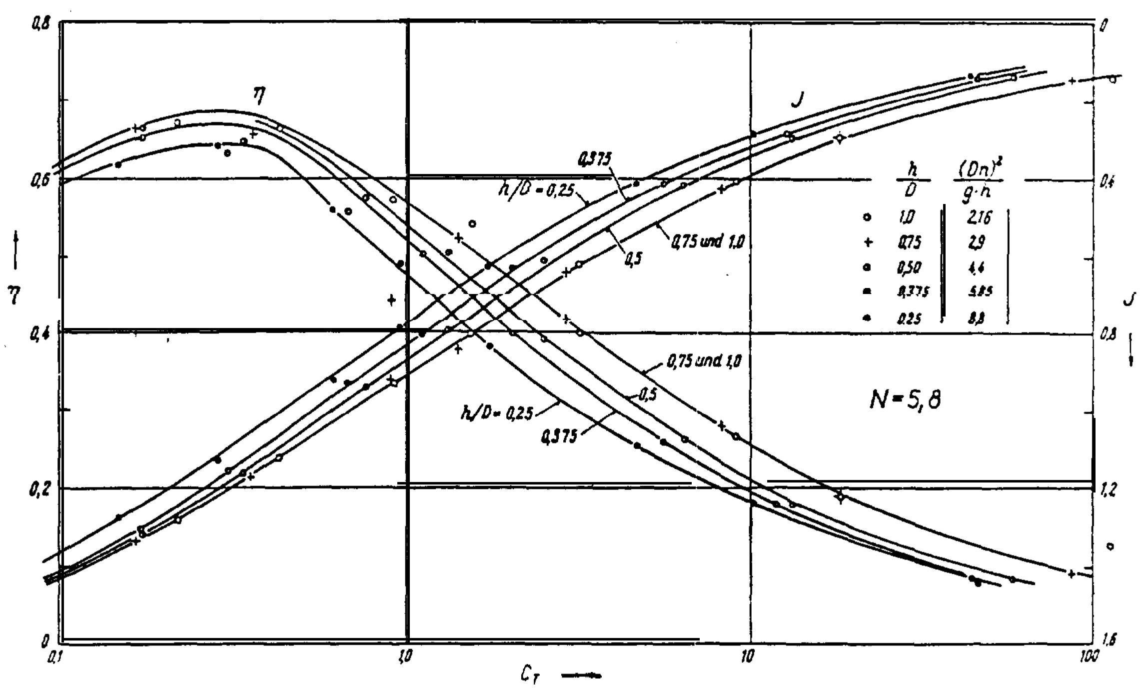


Fig. 37. Characteristics of a partially submerged propeller.

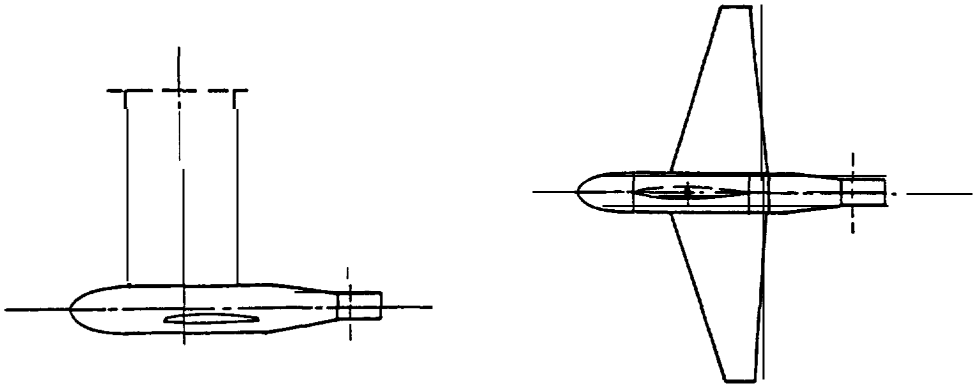


Fig. 38. Z drive for hydrofoil craft. V_S>50 knots.

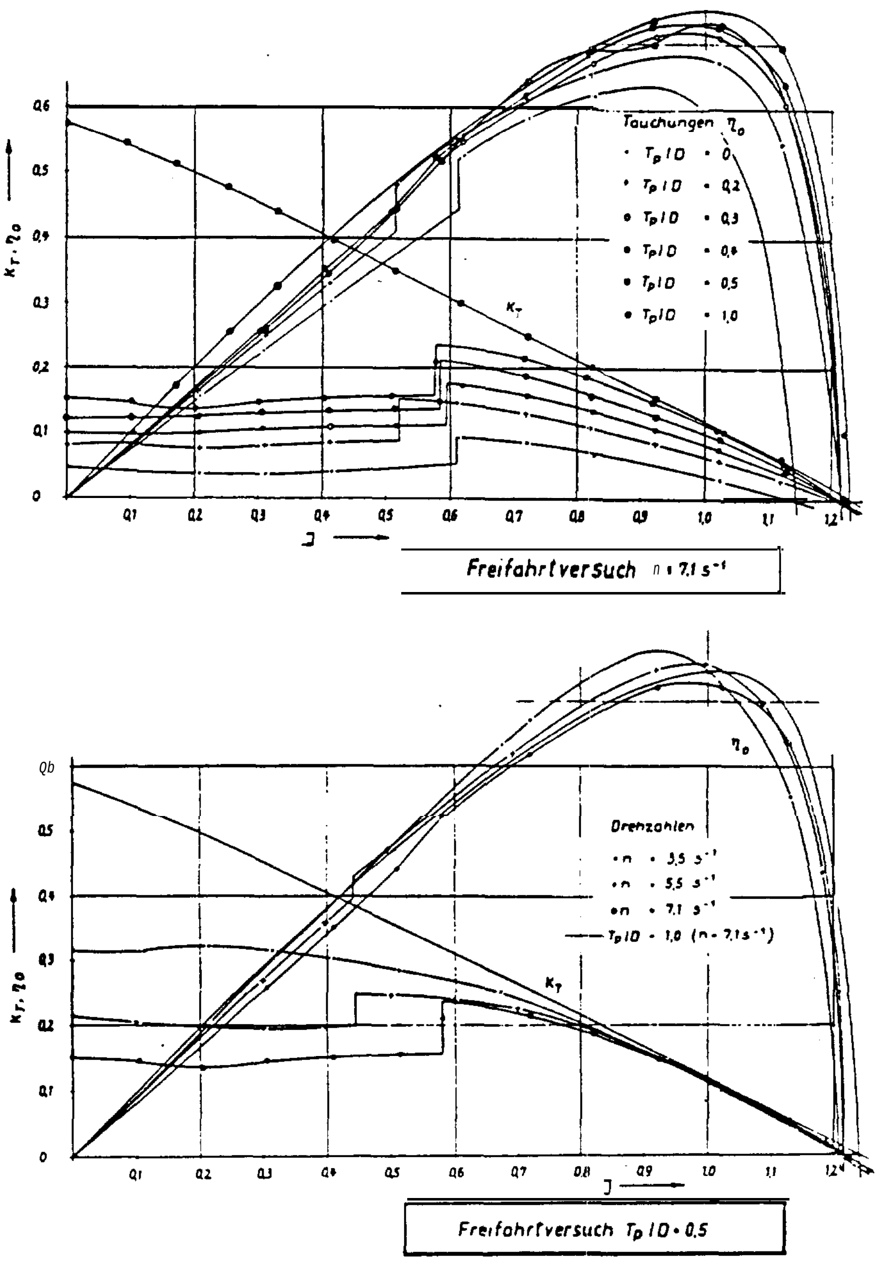


Fig. 39. Characteristics of a partially submerged propeller.

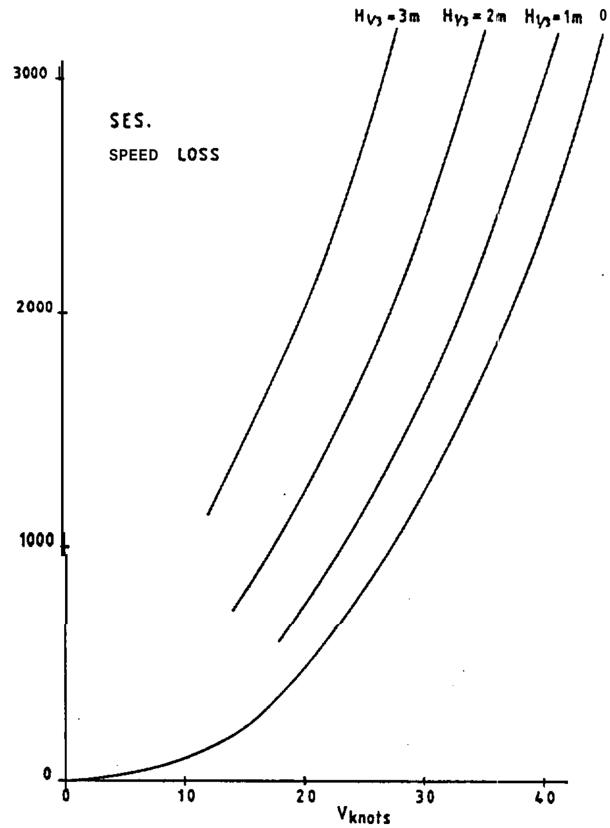


Fig. 40. Added power of a SES.

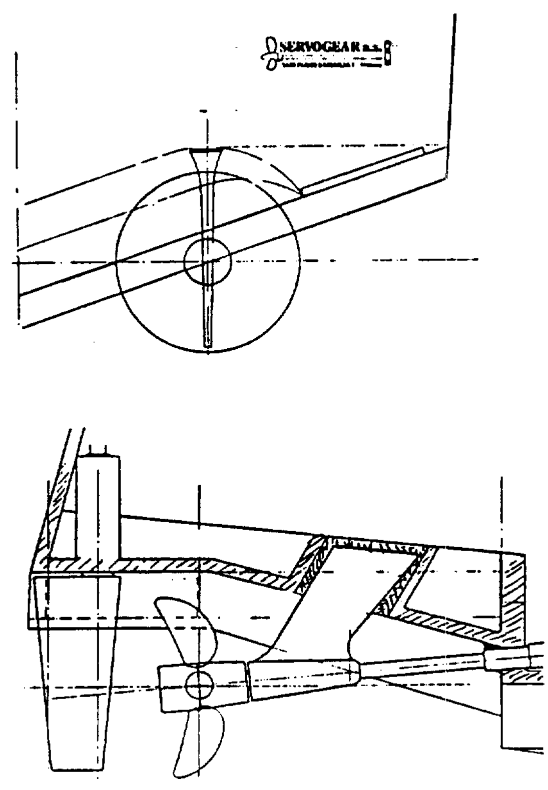


Fig. 41. Partially submerged propeller in a tunnel.

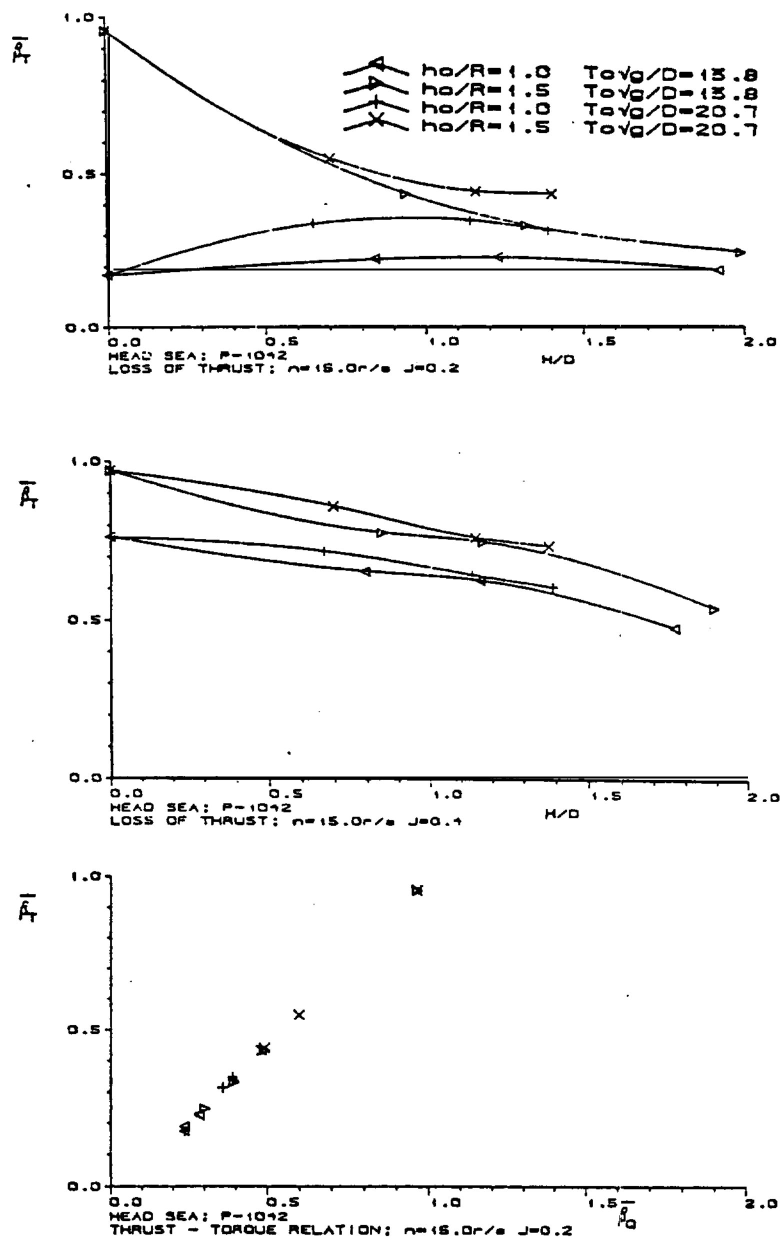


Fig. 42. Influence of relative velocities on propeller characteristics.

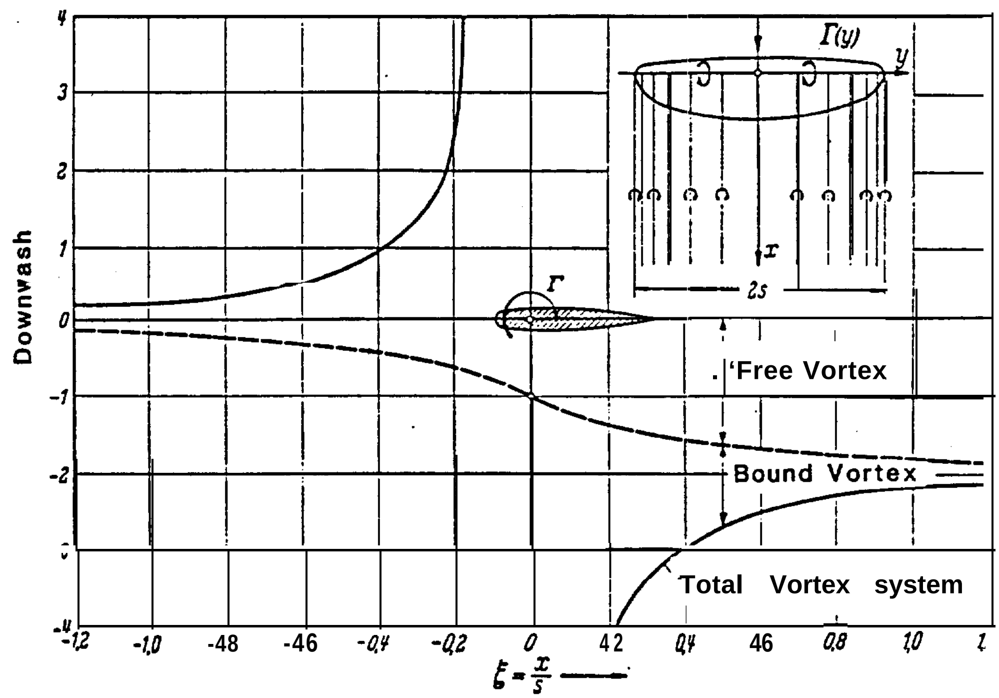


Fig. 43. Downwash from a hydrofoil.

HIGH-SPEED MARINE CRAFT
KRISTIANSAND 4-6 MAY 1988

POWER AND SEAKEEPING PERFORMANCE

OF

HIGH-SPEED MARINE VEHICLES

BY

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CONTENTS

INTRODUCTION

The design of high speed vessels must be among the most interesting areas of naval architecture. Vessels at high speed look impressive and are exciting. However, to select the correct high-speed concept for optimal mission characteristics is an extensive and challenging task.

In meeting the challenge of designing the best vessel the naval architect must realize the different qualities and limitations of the alternative concepts. In particular, one has to be aware of:

* The functional and operational requirements for the specified mission.
* The environmental conditions in the operational area.
* The criteria for evaluation and quantification of seakeeping performance.
* The individual calm water and seakeeping qualities of the alternative high-speed vessel concepts.

It is the object of this paper to evaluate relevant vessel qualities influencing the selection of alternative high-speed naval platforms. The attributes and limitations, calm water and seakeeping performance of different concepts are summarized and design problems and development trends related to these concepts are discussed. In this paper, vessels with speeds of more than 20 knots and lengths up to about 100 metre are considered.

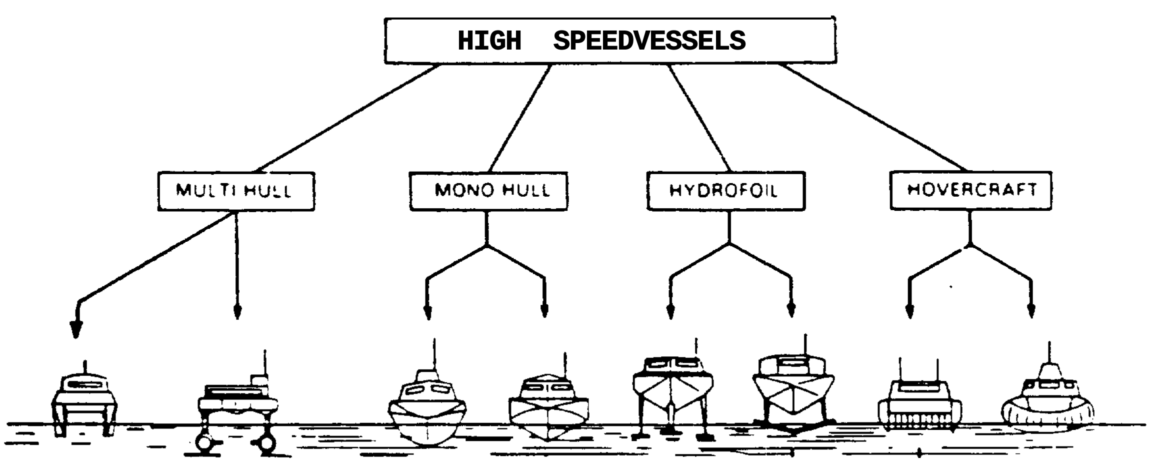


Figure 1 Review of different types of high speed marine vehicles. /1/.

1. HIGH-SPEED VESSEL CONCEPTS.

There has been an international increase of interest in the utilization and development of high-speed marine vessels, both for commercial, patrol and military applications. The number of vessels in service as ferries world-wide has increased by nearly 50% since 1980, the largest part of the increase is in the use of catamaran vessel /1/.

The following section presents the attributes, limitations and development trends for the alternative concepts. The data presented are taken from the MARINTEK High-Speed Database, which continously updates ship parameters, resistance, propulsion, manoeuvring and seakeeping data for existing and proposed vessels. The Database contains data on semi-displacement and planing monohulls, catamarans, hydrofoils, SES' and SWATH's.

1.1 Semi-displacement hulls and planing craft.

Figures 2 - 4 show the Loa-Displacement, Lwl-Beam and payload for high-speed mono-hull vessels. It has been estimated that more than 3000 vessels under 50 metres Loa are in active service world-wide. Vessels of lengths greater than 50 meters are fewer in number.

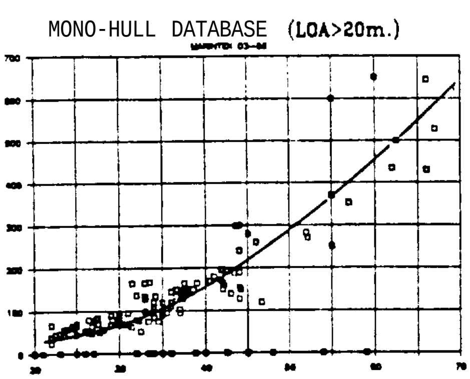


Figure 2

The displacement as a function of L_{OA} (m)

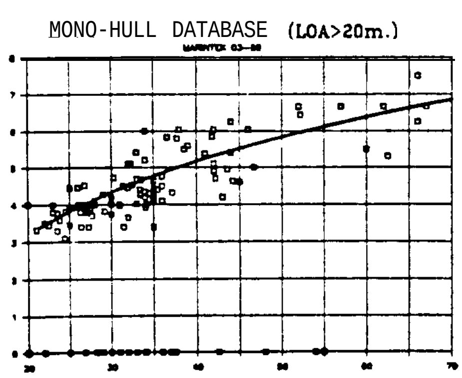


Figure 3

The L_{WL} - B_{OA} ratio

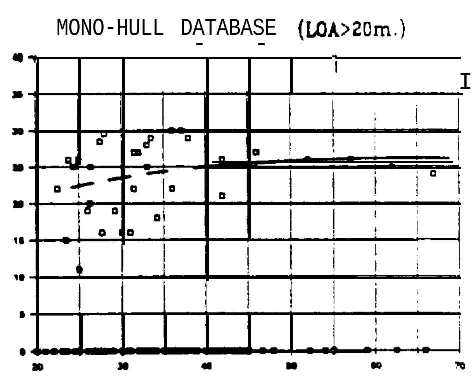


Figure 4

The payload (% of displacement)

Notice that payload is defined as the load of fuel, oil, stores, cargo, crew and passengers, given as percentage full load displacement. The payload percentage and thus the full load displacement is dependent on the vessels mission and on the operational requirements. Variation of payload and the resulting impact on power and seakeping performance will be an important consideration, when studying transport economy of high-speed cargo vessels.

There are two types of hull form employed for high-speed displacement vessels, the round bilge and the hard chine. The former has better resistance characteristics for speeds below Fn = 1.0, and has better seakeeping performance. $Fn = v/\sqrt{g \cdot L}$; v(m/s), g = 9.81 (m/s^2), L (m) .

Chined vessel are actual in sheltered waters, particularly for smaller vessels, with high deadrise and increasing length. At very high speeds the chine form has better roll stability. The chined vessel is assumed to have a weight limitation of approximately 300 tonne (LOA 50 m), above which the structural loads are prohibitive.

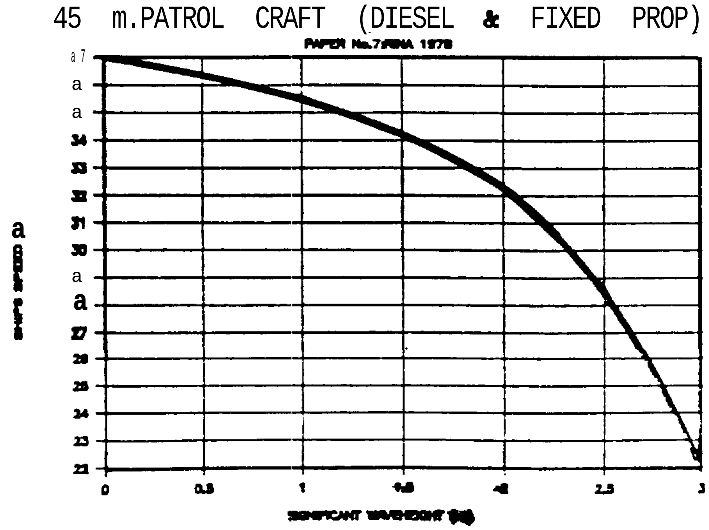
Major attributes of the round bilge monohull are:

* high carrying efficiency
* advantage in reliability due to simplicity and ruggedness of hull structure
* tolerance to increase of weight
* low construction and operational cost.

The capabilitis of the monohull is extremely well studied and documented in the literature.

Figure 5 shows the effect of wave height on speed for a 250 tonnes 40 knots patrol craft /2/.

45 m.PATROL CRAFT (DIESEL & FIXED PROP)



The understanding of the factors which affect round bilge hull performance is still being improved and applied in vessel developments. Some factors are:

* increasing Lwl/Bwl and/or Bwl/T improves both seakeeping and powering properties.
* increasing CB results in minor seakeeping improvements, this also has a favourable influence on powering performance.
* transom wedges have a minor effect on speed in below hump speed range ($Fn < 0.7$). For speeds in the hump region a reduction in resistance of more than 5% is expected. Transom wedges have minor effects on seakeeping performance, except for an increased tendency to broaching. This is due to low trim angles in broaching situations.
* active anti-roll and anti-pitch fins reduces the motions significantly, and improves speed and comfort in a seaway.

1.2 Hydrofoil vessel.

Approximately 250 navy and commercial hydrofoil vessels are operating throughout the world (outside the USSR.)

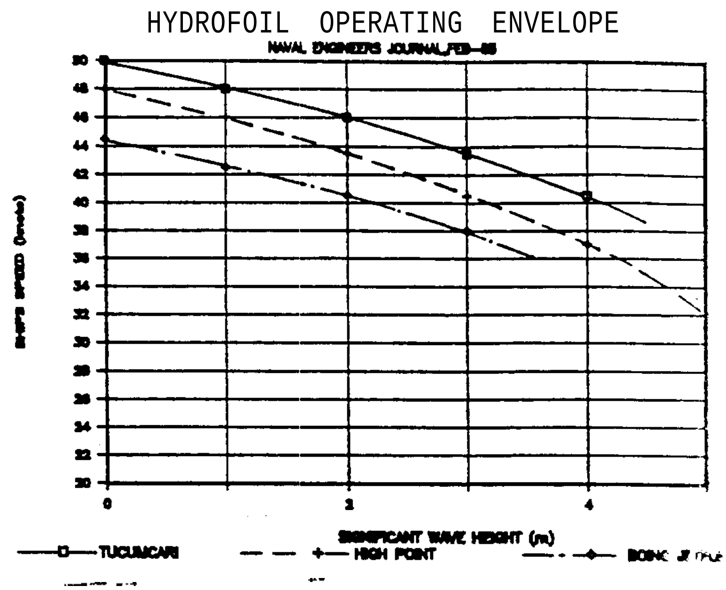
The major attributes and limitations of the hydrofoil vessels are:

* the ability to operate effectively in rough sea environments with a vessel which is small by conventional ship standards.
* an attractive power to displacement ratio in the 30 - 50 knots speed region permitting economical operation at this speeds when good seakeeping capabilities are vital for regular operation.
* vulnerability to damage of both the foil system and the propulsion system through striking floating objects.
* a large increase of drag in the take off mode. For propeller driven vessels an adequate margin in take off thrust is still available. However, larger power installations may pose serious problems.

The successful operation of hydrofoils in rough sea conditions is essentially the result of the interaction of the submerged foils with an automatic control system providing continuous dynamic control of the vessel.

The capabilities of the hydrofoil vessel are well studied in the literature.

Figure 6 shows operating data points for three hydrofoil ships



The largest operational hydrofoil in the west is the US Navy Plainview, a 329-tonnes vessel. Design studies indicate the feasibility of hydrofoil ships with displacements above 2000 tonnes /3/. The current regime of hydrofoil vessels lies in the sizerange up to about 1000 tonnes, and in the maximum rough water design speedrange of 30 - 50 knots.

1.3 Catamaran.

More than 150 high-speed catamaran vessels are in operation as passenger ferries or as cargo liners. The 50 m Westamarin design is the largest high-speed catamaran in operation.

The principle advantages and limitations of high-speed slender catamarans are:

* large deck area which can be designed for trade requirements.
* no design problems and well known construction technology.
* low construction cost, favourable power performance and low operational cost up to fairly high speeds.
* acceptable tolerance to weight increase and weight shift.
* reasonable ride quality and good ability to maintain speed in rough seas.
* large size (Loa > 50 m) experience is limited.

Very little general research has been performed to optimize the high-speed catamaran power and seakeping performance. Reference /4/ gives a bibliography related to model tests of high-speed vessels. It contains some 550 references, but none on the catamaran concept.

The wide design spectrum indicates significant potentials for improvements of the catamaran.

1.4 Surface Effect Ships (SES).

It is more than 30 years since the invention of the Surface Effect Ship (SES) concept. To date over 500 SES's have been developed and are operational throughout the world.

The state of art in SES technology is dealt with in Mr. Crago's paper at this symposium, thus only a short summary of major attributes and limitations of the SES concept follows:

* the SES has a wide design spectrum with a geometry which can be tailored to mission requirements. High L/b ratio for seakeeping and modest speeds and low L/b ratio for speeds as high as 100 knots.
* potentially the SES can be built to any size, as design problems tends to get easier as the vessel gets larger.
* reduced wetted surface of the SES permits higher speed operation with reasonable power.
* when on cushion, the freeboard is significantly larger than that of the similar monohull, thus providing dry decks in rough seas. Slamming does not occur until the significant waveheight exceeds the height of the wetdeck /3/.
* the lift system with active ride control provides a good ride at moderate sea state. However, the vertical accelerations at the higher frequency range can be irritating.
* the speed loss due to wind and waves is considerable.
* the SES is sensitive to weight and to longitudinal center of gravity shift. For the low L/b SES, a power margin is recommended to be included in the design.
* current large size bow and stern seal experiences are limited. Designs and materials for vessels up to 4000 tonnes and cushion pressures approaching 1000 kg/m^2 appear adequate.
* seals require maintenance and periodic replacements, with added operational costs.

Current indications in Surface Effect Ship technology are that new developments will include modest performance goals and safe technology sophistication. Efforts will be concentrated on improving seakeeping qualities and to the improvements of seal designs.

1.5 Small Water-plane Area Twin Hull Ships (SWATH).

About40 SWATH vessels are operating. The largest SWATH today is the 3500-tonnes deep ocean support ship "Kaiyo".

The SWATH vessel differ from the other types of high-speed vessel in that the vessels weight are totally supported by buoyancy, rather by hydrodynamic or aerostatic lift. The designer of SWATHs can select from a wide range of hull form parameters, allowing a hull form with performance characteristics that reflect the operational requirements.

The principle attributes and limitations of the SWATH are:

* The steadiness in a seaway combined with superior station keeping ability makes it very suitable for launching and recovery work.
* The SWATH has superior ride quality and ability to maintain speed in a seaway.
* The steadiness provides passenger comfort not available on other vessels at the same speed.
* The vessel is very sensitive to weight. Adequate weight growth margins must be included since changes in weight result in relatively large changes in draft and cross-structure clearance.
* The SWATH concept is not viable for high-speed applications without stabilizing fins. The reason is the unsymmetrical pressure distribution on the two lower hulls, giving a variable trim moment at increasing speed.
* vulnerability to damage through striking floating objects.
* The vessel may be more costly to operate as a result of its higher calm water resistance. This limitation can be offset by its lower added resistance in waves.

Substantial technology is available in Japan and USA to support the design and construction of SWATH vessels with sizes up to about 4000 tonnes /3/. Design up to 30000 tonnes are possible. However, above this size the beam begins to impose limiting operational factor.

1.6 General.

In general, the qualities of high-speed craft can be improved by:

- * reduction of weight by utilizing lightweight materials, machinery and systems.
- * increasing mobility by improving resistance and seakeeping characteristics either by design changes or by introducing active control systems.
- * increasing transport economy by improving propulsion efficiency or ~~energy~~ engine efficiency.

2. POWER PERFORMANCE

Model tests have played a major role in the development of all the alternative high-speed concepts. However, rather surprisingly full-scale data for resistance power correlation and scaling are not readily available. Numerical programs for resistance predictions are accordingly evaluated or generated by systematic model test series. Power prediction including propulsive performance, where cavitation, air-suction and complex interaction effects between hull and propeller system, is even more dependent on accurate and sophisticated model testing.

2.1 Power Comparison.

General comparison of power performance for the alternative high-speed concepts will, to a large extent, be influenced by the choice of comparison criteria.

To indicate the relative power requirements of the alternative high-speed vessels, it is instructive to compare the installed power of existing vessels. Data from the MARINTEK semi-displacement mono hull Database is given in figure 7. Figure 8 shows the mean transport efficiency at calm water for the alternative high-speed vessels. Note that the transport efficiency include the efficiency of the propulsor.

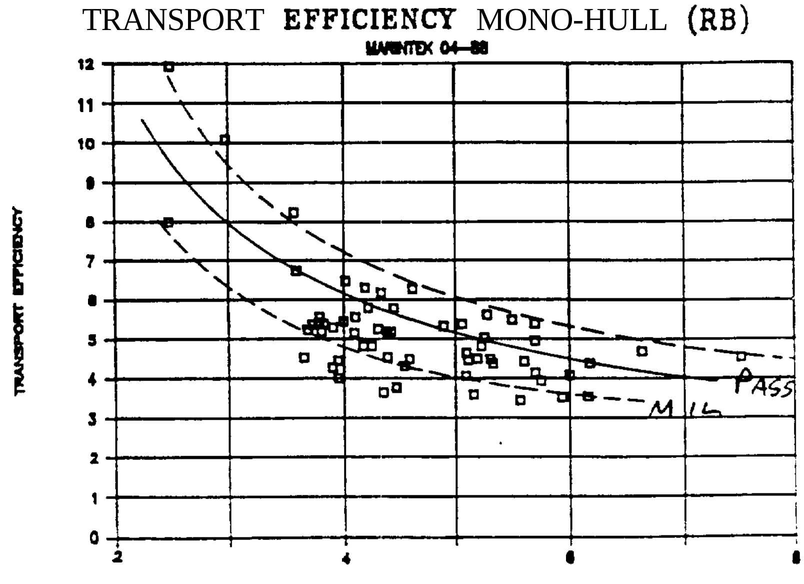


Figure 7

The transport efficiency as a function of $F_{n\nabla}$

$$F_{n\nabla} = \frac{V}{\sqrt{g \cdot \nabla^{1/3}}}$$

v (knots)

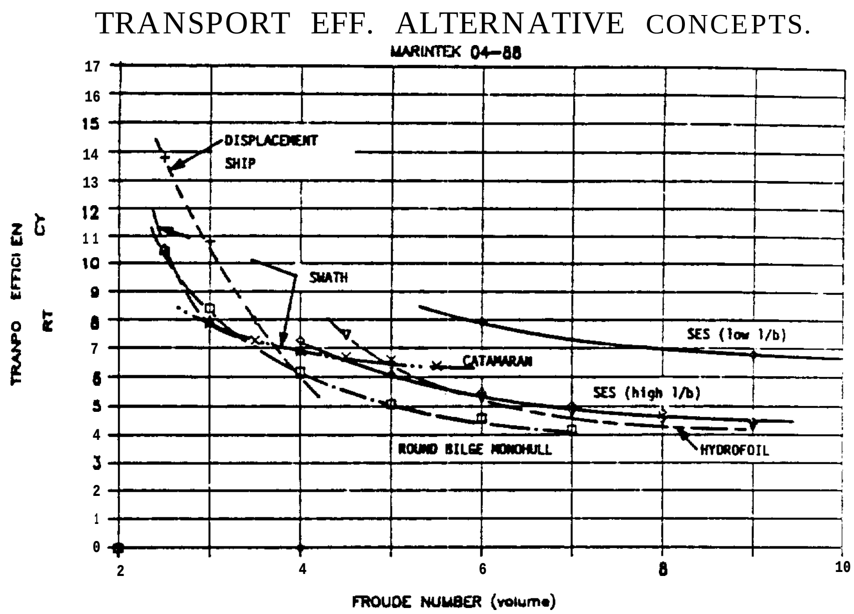
∇ (m^3)

Δ (tonnes)

SHP (hp)

Figure 8

Q alternative concepts as a function of $F_{n\nabla}$ (Froude volume number)



The ratio

$$Q = 6.86 \times \frac{\text{Displacement (tonne)} * \text{Max.speed (knots)}}{\text{Installed power (hp)}}$$

is called transport efficiency. It corresponds to the general "effective lift-to-drag-ratio" used to compare the relative performance of different ship concepts. It is identical to the dimensionless transport efficiency $W \cdot V/P$ with W (Newton), V (m/s) and P (Watt).

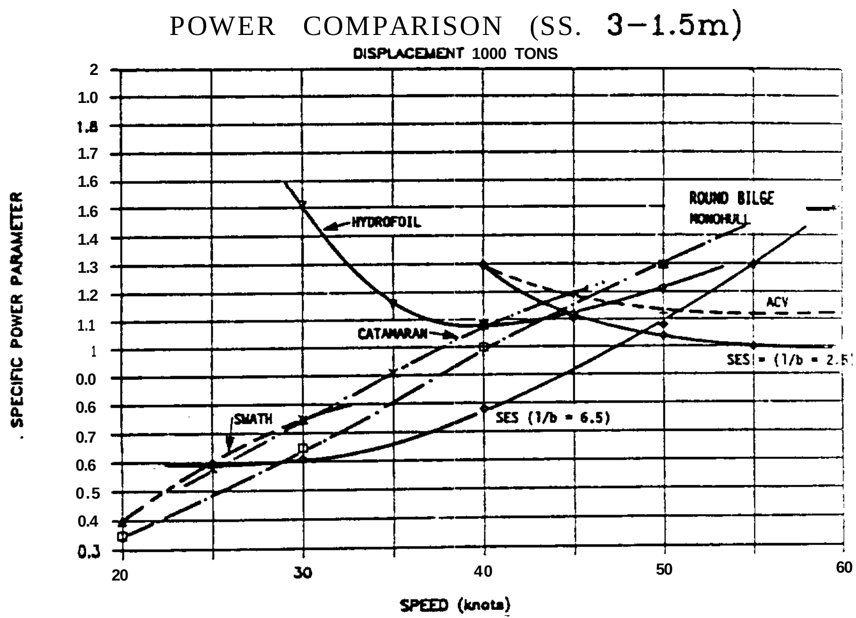
The transport efficiency of conventional displacement ships is also included. Figure 8 shows that:

* Up to about $F_{n\nabla}$ = 3.5, conventional displacement ships have the highest calm water transport efficiency.
* From about $F_{n\nabla}$ = 4.0 to $F_{n\nabla}$ = 5.0, hydrofoil vessels have the highest transport efficiency.
* In the range between $F_{n\nabla}$ = 2.5 - 3.5 SWATH ships have a high efficiency.
* Above about $F_{n\nabla}$ = 5.0 the SES have the highest transport efficiency.
* The high-speed mono-hull in general has a low transport efficiency.
* The catamaran utilized as cargo carrier has a high transport efficiency in the range between $F_{n\nabla}$ = 4.0 and 6.0.

In **figure 9, /3/ 1000 tonnes displacement vessels of SWATH, Hydrofoil and SES (high and low l/b) types operating in sea state 3 (significant wave height approximately 1.5 m) is considered. Power performance data for a 1000 tonnes round bilge mono-hull and a slender catamaran has been added by the author.**

Figure 9
The specific power parameter as a function of speed (knots) on sea state 3.

$$\frac{\textbf{Total Power (SHP)}}{\textbf{Grossweight (tons)} \times \textbf{Speed (knots)}}$$



The figure shows that the best cruise speed for the alternative vessels in sea state 3 are as follows:

*	**Mono-hull**	**20 to 45 knots**
*	**Hydrofoil**	**38 to 50 knots**
*	**Catamaran**	**35 to 50 knots**
*	**SES (low l/b)**	**50 to 80 knots**
*	**SES (high l/b)**	**30 to 50 knots**
*	**SWATH**	**20 to 30 knots**
*	**Air cushion Vessel**	**50 to 80 knots**

The most efficient operation in terms of minimum specific power are achieved by the mono-hull up to 30 knots, the high l/b SES from 30 to 50 knots and the low l/b SES above 50 knots.

It must be noted that the characteristics are dependant on the choice of vessel size, sea state, payload requirements, seakeeping criteria etc. The figure presented is conditional and care must be taken in using it for other purposes.

2.2 Power assessment.

Methods for prediction of hydrodynamic resistance of ships are used in preliminary design studies when general influence of displacement, length, beam, hull form, cushion and foil characteristics etc. on speed and power have to be determined. Analytic prediction methods combined with systematic model series results are available for planing craft, semi-displacement hulls, hydrofoils and partly for SES. No complete prediction method exist for the SWATH or the catamaran. Model tests are usually carried out once these first design considerations have resulted in an interesting or definite design.

Marintek has developed a set of formulae for assessment of the speed and total installed horsepower relationship for alternative high-speed concepts. The method is based on information of installed power and maximum speed given in the literature. The formulae are updated at regular intervals.

As an example the formula for the semi-displacement hull is given below and the accuracy is presented in figure 10.

* Round bilge vessel

SHP = A . (1.03 + 0.085 $F_{n\nabla}$) (Accuracy ± 15%)

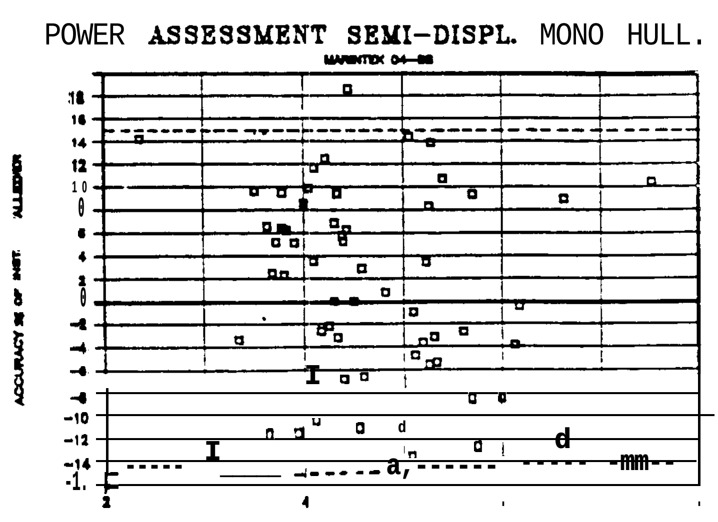


Figure 10
The accuracy (%) as a function of Froude volume number

3. SEAKEEPING PERFORMANCE

The fundamental attributes of any marine surface vehicle are speed and seaworthiness.

It is clearly desirable, at an early stage in the design process, to assess whether the seakeeping ability of a concept will be adequate for a particular mission. To do this it is necessary to quantify:

* seakeeping requirements related to comfort and structural design.
* vessels seakeeping characteristics.
* environmental conditions in the area of interest.

3.1 Seakeeping criteria.

Criteria for acceptable levels of ship motions or ship loads are widely discussed in the literature /6/. The importance of impacts or responses depends on the operation of main subsystems and on the type of vessel. Vital criteria with regard to personnel effectiveness and passenger comfort are:

* vertical acceleration
* lateral acceleration
* roll motion
* pitch motion

Vital criteria with regard to vessel hull and hull safety are:

* slamming
* deck wetness
* vertical acceleration

The designer of marine vehicles has more data to work from when selecting hull proportions to give optimal calm water speed-power performance, than he has when seeking to obtain the best performance in waves.

Model test techniques and motion predictions for specific designs operating in random seas have been available for some time. However, the lack of correlation between model predictions, analytical predictions and full-scale seakeeping data gives some uncertainties. The criterion for vertical acceleration at the bow should be considered as a criterion for comparing seakeeping performance of alternative designs, as it reflects the overall level of vertical motion, which in turn may be critical due to slamming, deck wetness and comfort.

MARINTEK's standard seakeeping criteria is based on performance degradation taking into consideration the mission of the vessel, the vessel size and operational conditions. Thus, the criteria or the seakeeping performance can be related to standard criteria for working effectiveness by

	Acceleration (g)		Roll
	Vertical	Lateral	
* Light manual work	0.40	0.20	12°
* Heavy manual work	0.30	0.14	8°
* Intellectual work	0.20	0.10	6°
* Transit passengers	0.10	0.08	5°
* Cruise liner pass.	0.04	0.06	4°

(Significant double amplitudes).

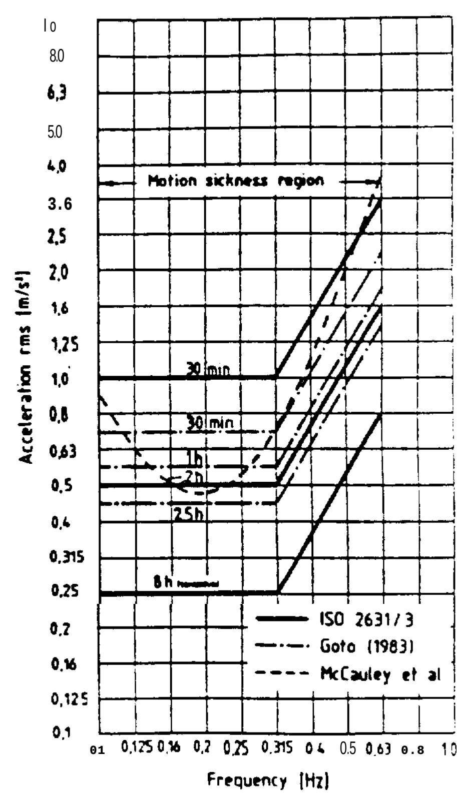


Figure 10

The severe discomfort boundary according to the International Standard ISO 2631/3-1985 with regards to vertical acceleration as a function of frequency for exposure times of 30 minutes, 2 hours and, tentatively, 8 hours. Also shown are Goto's (1983) proposal for a standard and results of laboratory experiments by McCauley et al. (1976) corresponding to a 10% motion sickness incidence ratio (vomiting) amongst unadapted healthy men.

(Significant = 2σ (rms)).

3.2 Prediction Techniques.

The determination of the probability of a vessel achieving a specified seakeeping capability depends largely on the motion-prediction techniques and the comparison with relevant criteria. Information on vessel characteristics can be obtained by theoretical methods, model experiments and full-scale trials.

Numerical prediction models for twin-hull vessels such as SES's, SWATH's and catamarans are discussed in Mr. Faltinsen's paper at this symposium . Extensions of strip theory programs for mono-hulls are applied and limitations of theory are presented.

From extensive model test data it is generally concluded that conventional high-speed displacement and semi-displacement hull-forms are reasonable well represented by using strip theory. This theory provides a good preliminary design tool for motions other than relative motions and added resistance in waves.

At MARINTEK, motions, added resistance and propulsion performance in a seaway is calculated with the program SEAWAY. The combined use of experimental and computational tools is indicated in Figure 11. Specially for small and fast ships where calculations yield inaccurate results, model tests are necessary. In cases where effects of local differences of hull shape are studied model tests are a must.

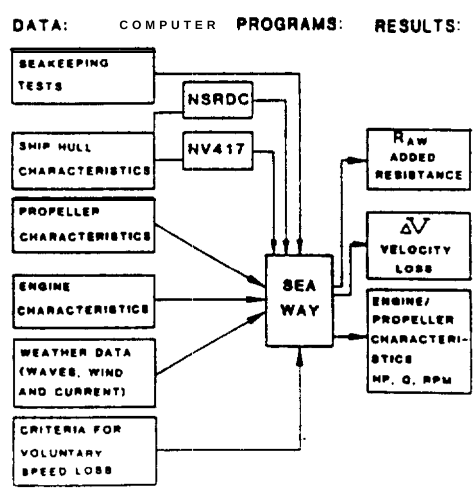


Figure 11

Numerical calculation of performance in a seaway /6/.

3.3 Seakeeping Comparison

The vertical acceleration criterion at the bow reflects the overall level of vertical motion in weather conditions where slamming or deck wetness may be the limiting factors. The vertical acceleration is thus the basic operational requirement both for hull safety and personnel. This vessel motion in addition to vessel speed loss is used to discuss seakeeping performance of different high-speed concepts.

Figures 12, 13, 14 employ MARINTEK's model test data and data from available published literature. The levels of technology of high-speed concepts are roughly shown in these figures. In constructing the figures, Froude scaling and Database information have been used to give the different vessels an approximately 60 tonne payload capacity. The sizes of the compared vessels are:

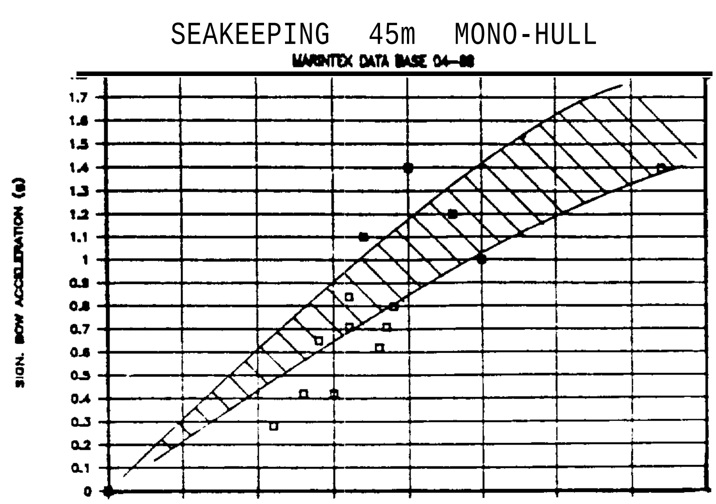
* Mono-hull 45 m - 220 tonne
* Hydrofoil 38 m - 200 tonne
* Catamaran 40 m - 190 tonne
* SES 40 m - 180 tonne
* SWATH 28 m - 200 tonne

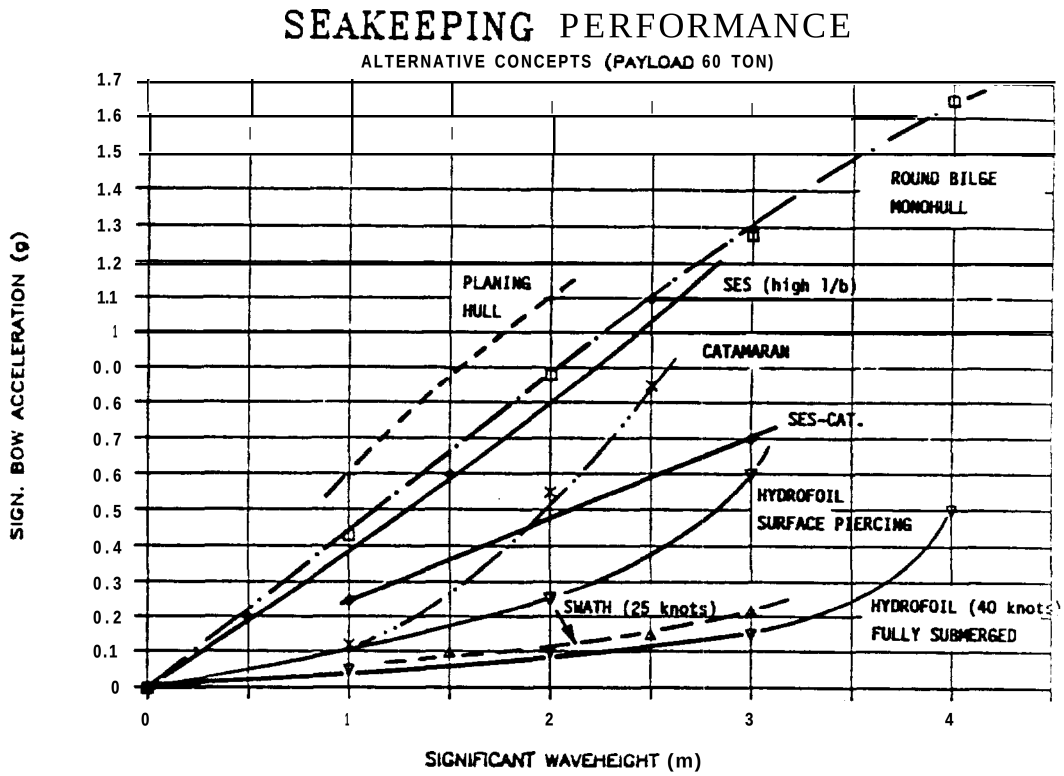
It must be noted that the SES, SWATH and Hydrofoil data include the beneficial effects of ride control systems.

It should further be noted that consistent seakeeping data for high-speed catamarans are completely missing and that the characteristics for this type of vessel, which are shown in the figures, are based on limited model test data and full-scale experience.

Figure 12

Bow acceleration for round-bilge hulls at speed range about Fn = 1.0 (Significant values)





Fiaure 13

Bow acceleration level for alternative high-speed vessel concepts as a function of significant wave height.

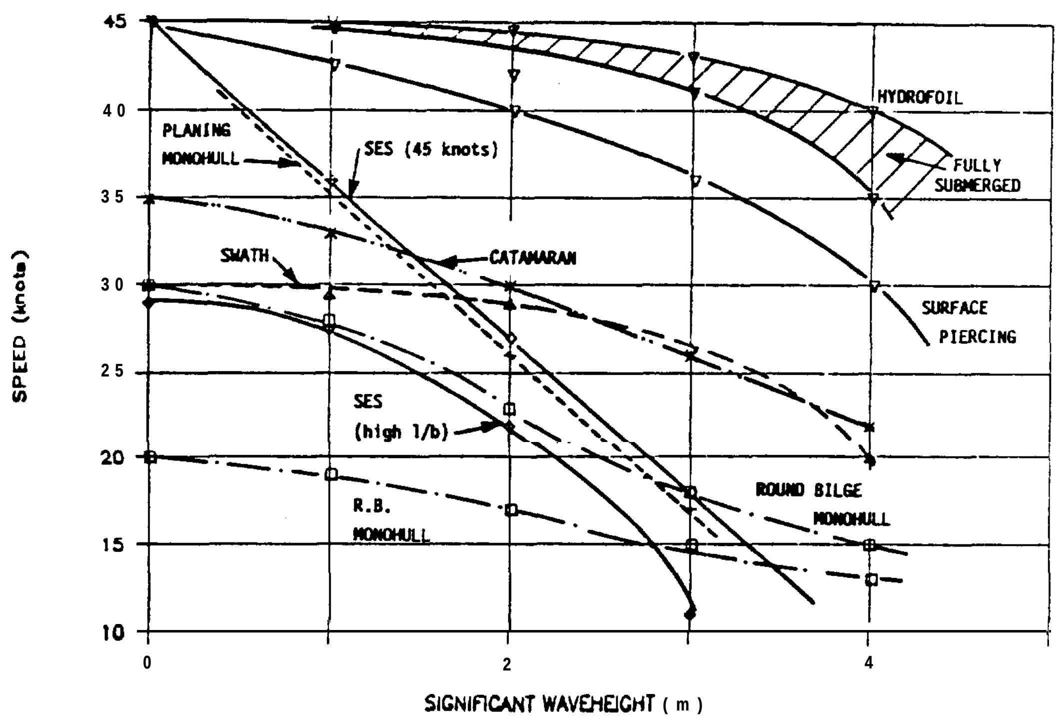


Figure 14.

Involuntary speed loss in waves due to added resistance as a function of significant wave height.

It would be incorrect to assume that the boundaries shown represent the ultimate performance capabilities for the alternative concepts. The capabilities can be extended appreciably by improving design, improving or introducing ride control systems or, of course, by increase of vessel size.

Using Froude scaling, the speed and sea state both increase with increasing vessel size, keeping the given acceleration level or the speed-loss percentage constant.

The seakeeping performance curves demonstrate the following trends:

* Superior speed and ride qualities for fully-submerged Hydrofoil vessel (speed) 30 knots), the qualities degrading rapidly when the design significant wave height exceeds the strut length.
* Some degrading qualities for the surface-piercing Hydrofoil vessel.
* Superior ride and speed qualities for the SWATH vessel at moderate speeds (speed $<$ 30 knots).
* Rapidly degrading speed and ride qualities for planing vessels and SES'. Significant potential for improvements in comfort for 'SES designs (ref. SES-CAT).
* Round bilge mono-hulls have reasonable speed qualities at moderate speeds, but with considerable ride or acceleration limitations.
* Good speed qualities for the slender catamaran vessel, but with rapidly degrading ride qualities at moderate to high sea states.

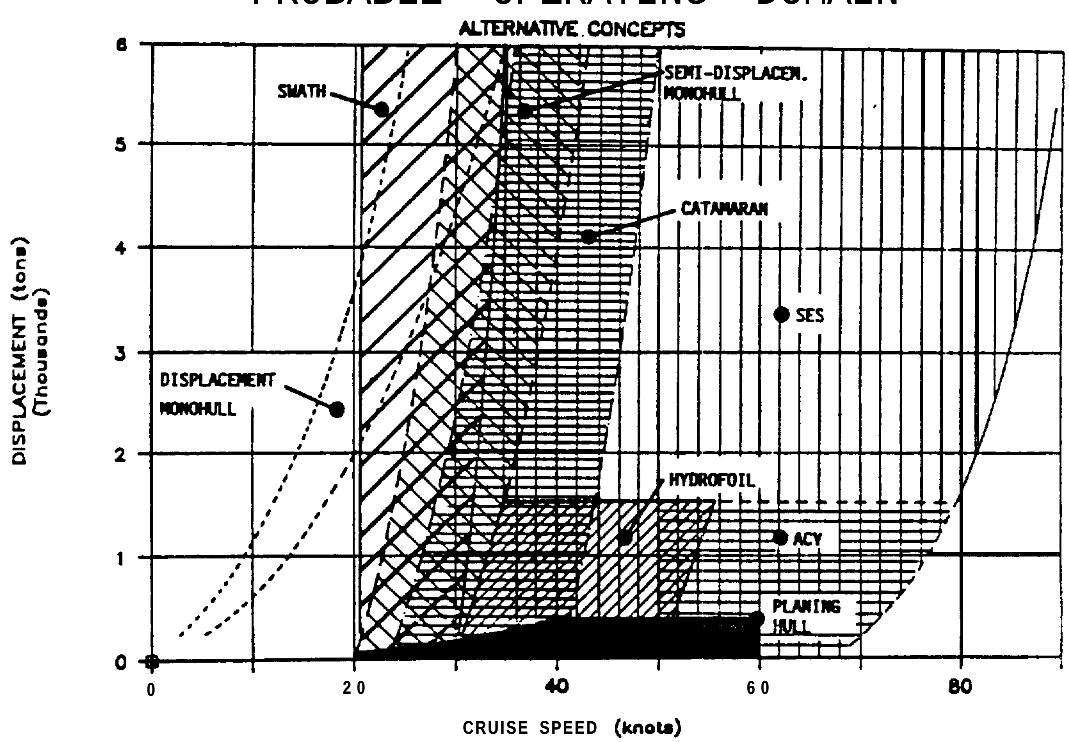
Speed quality is the ability of the vessel to maintain it's speed and heading while operating in rough seas. The complexity of the sea definition and the large differences in vessel designs, combined with operational limitations makes comparison difficult. It is not always clear from published data whether the speed limitation is due to power limitation, or due to comfort limitation of passenger and crew or an imposed limitation due to structural loads.

4. A LOOK TO THE FUTURE

In addition to size, payload, speed, power performance, range, speed loss and ride quality considerations, a comparative evaluation of the operational utility and efficiency of the alternative vessels must include considerations of cargo handling - stowage capability, draft, manoeuvring, building cost, operating cost, reliability etc.

Figure 15 shows the anticipated speed versus total displacement operating domain for each of the alternative vessels discussed. The original figure, /5/, has been adjusted to a limitation of 6000 tonnes displacement and 90 knots speed. In addition the role of the round bilge high-speed displacement vessel and the slender catamaran is evaluated.

PROBABLE OPERATING DOMAIN



The conclusions presented are conditional in nature and care should be taken in their application.

Future technical developments will improve operating characteristics. Optimization of general main parameters, special components and combination of concept characteristics will continue both by utilizing numerical methods, model test technology, operating experience, sophisticated control systems, new materials, construction methods, etc.

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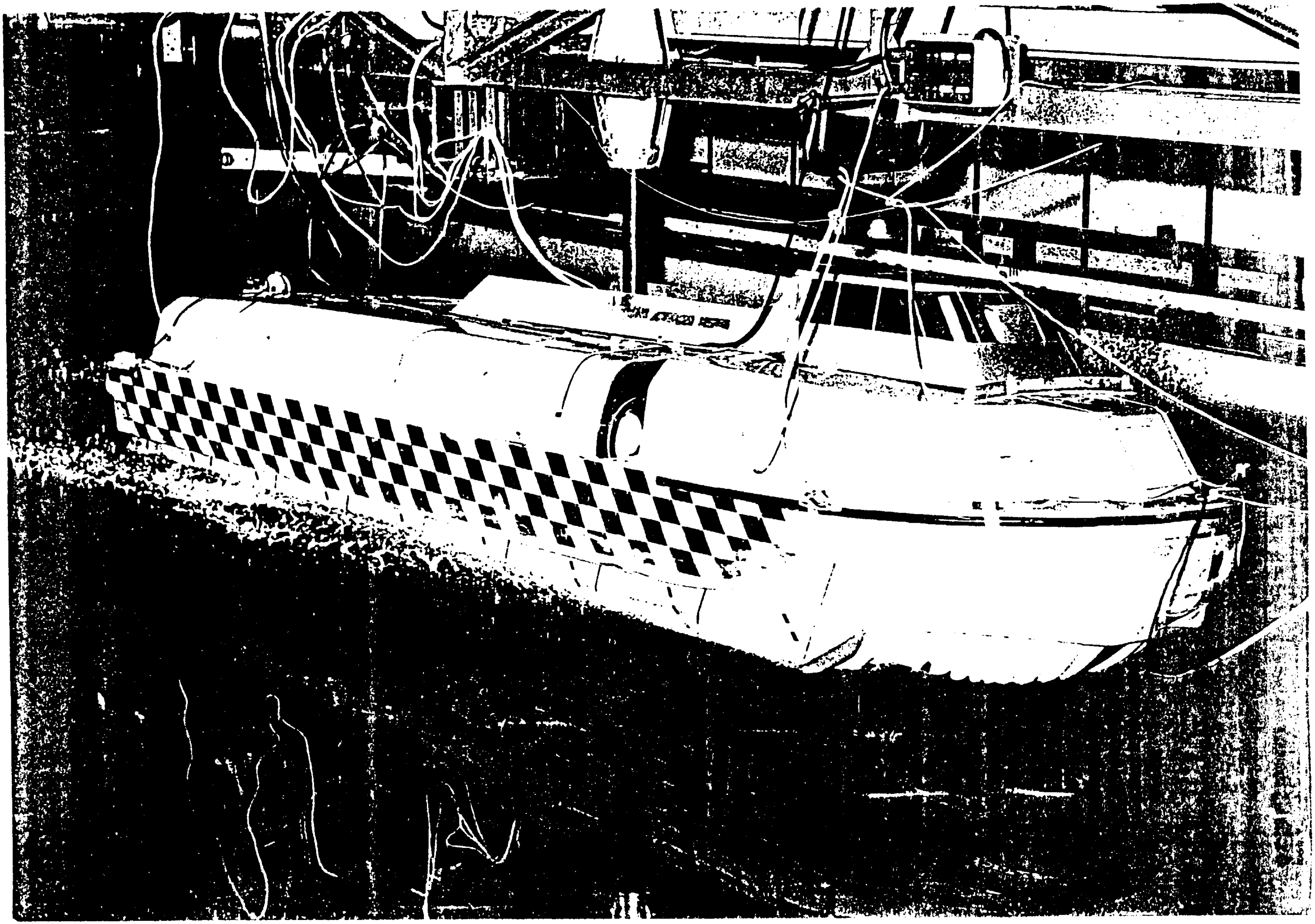
HIGH SPEED MARINE CRAFT CONFERENCE.

KRISTIANSAND.

MAY 1988.

STATUS OF SES TECHNOLOGY.

BY WILLIAM A CRAGO.



FRONTISPIECE

STATUS OF SES TECHNOLOGY.

C O N T E N T S -

Frontispiece,

This shows a model of a Hovermarine SES being tested in a towing tank. on a carriage having a 30 ft. long free to surge rig.

STATUS OF SES TECHNOLOGY.

1. INTRODUCTION.

It will be appreciated that no paper dealing with SES technology can be totally comprehensive or completely up to date. This is simply because the SES principle is currently being subject to research and development in many different centres all over the world. Some of this work is of a military nature and information about it is restricted, while in other areas, it is known that development work has been undertaken but it has not yet been made public for various reasons.

Under these circumstances, the author has not attempted to compile an exhaustive survey, but has concentrated on those aspects, primarily of commercial craft, which he personally feels are of interest. In this connection it is relevant to say that the author has had a long history of intimate involvement with SES and similar craft. In fact, he tested his first SES model in a towing tank in 1959. This was a surprisingly sophisticated craft for those days, and was intended to clarify a number of features of operation which were not understood at that time. It was tested with a form of automatic ride control which was obtained by manipulating the cushion. It also had the facility for air lubrication of the side walls. This model, and others like it, provided valuable insights into SES potentials, some of which, have only recently been implemented in the full scale regime. However, reference to the author's 1960 paper (Ref.1) will show how much progress has been made in many other respects since those early days.

2. SUMMARY.

The paper presents a definition of the SES and then lists the various components of resistance to which such craft are subject. It is shown that the resistance characteristics largely determine the shape of current craft and these shapes and the associated performances are examined. SES performance characteristics are then compared with other high speed craft, in an attempt to highlight those areas where future research and development are most required. Suggestions for such future work are given, and the resources and methods required to carry this out are discussed.

3. S E S SHAPE AND PERFORMANCE.

This section of the paper is intended to provide a proper context for a subsequent consideration of the objectives for future development and testing.

3.1. DESCRIPTION OF THE SES.

The SES can be simply defined as a water going craft which, at speed, is mainly supported by a cushion of air having a pressure greater than that of the atmosphere. The cushion is generated by *means* of fans, and is contained by two water piercing side walls or hulls at its sides, by flexible seals (or skirts) fore and aft, by the water surface below and the SES centrebody above. The seals represent the only really new technology required in the SES.

There are of course different kinds of SES. For example, the centre body may *or* may not be designed to contribute to the buoyancy when the air cushion is not being generated. Nowadays, it is generally recognised that a craft which can operate efficiently as a catamaran when off the cushion (i.e. centre body dry) will have a great advantage when manoeuvering at lower speeds in restricted waters, in docking, and when low speed patrolling is required in a military context. Again, the proportion of the craft weight carried by the cushion when the craft is at speed is an important design variable. Some SES designs may be regarded as "deep cushion" craft. In this case the trade off between transverse stability and the improved sea keeping resulting from the deep cushion must be carefully assessed.

3.2. COMPONENTS OF RESISTANCE.

The SES is different from other high speed craft in that it has some novel components of resistance and it is useful to distinguish the following:-

1. Hull resistance.

Under this heading we may list the viscous friction resistance, the form drag, residual resistance, and particularly with a planing hull, the planing drag; which is the aft facing component of the normal pressure on the bottom and the spray rails. In addition to these, there will be components associated with spray, wetting, the presence of the air cushion and an interference effect from the other hull. The hull will also suffer a resistance specifically associated with the propulsion unit whether it be a water jet or a propeller.

2. Appendage resistance.

In the case of an SES with water jet propulsion, the appendage resistance will consist of that due to rudders (or their equivalents) together with the drag of other excrescences on the hulls. In the case of an SES with propellers, the drag of the shafting and brackets, etc. must also be taken in to account. In most propeller driven SES, the shaft angle has been relatively high, and the associated brackets, etc. have perhaps not been as well designed as they could have been. As a result, the total appendage drag has been very large and in some cases could well exceed 20% of the total drag.

3. Cushion resistance.

The passage of the air cushion over the water surface generates waves, and the associated resistance is felt by the craft. Clearly there is an interaction between the hulls and the cushion and there is another resistance component associated with this. The cushion resistance is also significantly dependent on the depth of water in which the craft is operating.

For most SES, the cushion resistance is the most important component of the total drag. For a low length to beam ratio (L/B) the resistance rises to a peak value and then falls away as the speed increases. For a higher L/B value the peak value of resistance is much lower, but rises to a greater value at high speed. The shapes of the resistance curves and the effects of L/B, etc. are now generally well known, and their theoretical derivation is given in ref.2.

4. Seal resistance.

This is the most difficult of all the resistance components to isolate, understand or predict. There is an element of viscous friction drag as well as pressure, spray and wetting drag. The interaction with the cushion and the hulls still further complicates the situation.

5. Air resistance.

It may be helpful to cite an example in order to demonstrate how important this component can be. Thus, for a craft moving at 50 knots into a head wind of 20 knots, the air resistance associated with a 1200 sqr.ft. frontal area can easily be 60001b. corresponding to say 2000 BHP.

6. Momentum resistance.

A continuous supply of air is required to generate the cushion and this air must be accelerated up to the relative speed of the craft. This gives rise to a resistance which is proportional to the volume of air required. It is obvious that this component will be considerably larger for an air cushion vehicle which exhausts air all round its periphery, but it is not negligible even for an SES. Thus, in a 20 knot head wind and a speed of 50 knots, the momentum resistance of a typical SES could be 1000 lb. corresponding to say 300 BHP.

7. Cushion air resistance,

This component is rather small and may be negative. It is due to the momentum of the air escaping from the cushion. In some applications, air venting from the rear seal has been deliberately encouraged in order to produce a thrust.

8. Wave going resistance.

It is customary to regard the reduction in speed associated with the passage of an SES through waves as being due to a single extra resistance component. This may be a convenient artifice but hardly corresponds with the facts. The trim and heave changes caused by wave encounter have repercussions on each of the resistance components mentioned above.

Propulsive power is required to overcome the sum of the above resistance components. Power is also required to generate the cushion. This is a relatively small proportion of the total power installed in a fast SES, but any improvement in fan efficiency, etc. is worth considering at the design stage as long as it can be shown to be cost effective. Losses occur in the fan intake, the fan itself, in the ducting to the seals and in the feed apertures to the cushion. Any reduction in these losses will obviously reduce the total amount of power required to be installed in the craft.

3.3. CURRENT SES SHAPES AND PERFORMANCES.

A ready appreciation of the various current shapes of SES and the way in which the resistance affects these shapes can be obtained from the data and photographs of ref.3. This data has teen used to produce fig.la which shows the propulsive

BHP per ton for each craft listed, plotted against a simple speed coefficient based on the craft maximum speed in knots and the weight in tons. BHP/ton is, of course, not non-dimensional but is thoroughly practical, and the data fall fairly close to a single line surprisingly well considering the differing natures of the craft, and the different design philosophies involved. Perhaps not too much should be read into this-except that the data is factual and the figure does therefore show what was currently being achieved when ref.3 went to press.
The upper part of the line drawn in fig.1a is poorly supported but it is interesting to note that amphibious hovercraft data (not shown here) fall closely on the line at values of the speed coefficient from 22 to 27.

Again, from the same source, fig.1b shows the length beam ratio plotted against the speed length ratio. The graph shows how the designers have chosen the lower L/B values for the higher speed length ratios largely because this gives a lower cushion drag coefficient. Finally, fig.1c shows the craft weight in tons divided by the length and beam in feet (a rough approximation to the cushion pressure), plotted against a simple function of the weight in tons. It is interesting to see how the pressure increases steadily with weight. At one time it was thought that there would be a limiting value for the cushion pressure above which, flexible seals would not be practical. However, there is no sign of such a limit as yet, and cushion pressures orders higher than those currently in use have been employed in some design studies.

In order to set the curves of fig.1a in context, ref.3 was also employed to obtain comparable data for other craft which make contact with the water. This is shown in figs.2a and 2b with mean lines drawn through the points. Fig.2c compares the envelopes of the data for catamarans and nonohulls, hydrofoils and SES respectively. Bearing in nind the limitations of this kind of approach and the nature of the original data, it would none the less appear that current SES usually require a lower brake horse power per ton of weight than either the hydrofoil, catamaran or monohull when operating at the same value of $Vk/\sqrt{\Delta_T^{1/3}}$ or Fn_∇ .

Another way of setting calm water SES performance in context is to make use of a "Transport Efficiency". For present purposes, this has been defined as:-

$$T.E. = \frac{\text{Speed in knots} \times \text{Total weight in tons}}{\text{BHP}}$$

All the available data from ref.3 were analysed in terms of

T.E. and the envelopes for the three categories of craft are shown in fig.3. Again it must be emphasised that this presentation does not set out to show the potential of the three forms of hull but rather the existing state of affairs. From this simple presentation it would appear that monohulls and catamarans are currently limited to speeds below 33 knots. The hydrofoil principle has been employed up to speeds in excess of 50 knots with a high T.E. while the SES has also been built to operate at speeds up to 50 knots with a considerably better T.E.

It will be appreciated that the definition of T.E. employed above is rather crude. A better definition would employ the payload instead of the total weight. This was not possible using the available data, so it is interesting to look at the average values of number of passengers per ton for the three categories of craft as follows:-

Craft	Av. value of passengers per ton.
Cat.	3.1
Monohull.	2.7
Hydrofoil.	2.2
SES.	2.7

Of course, these figures depend, in part, on the standard of comfort provided. However, they suggest that a more correct definition of T.E. would not change the conclusions drawn from fig.3. It is also tempting to conclude that the high figure for catamarans is because of their simplicity and large deck area while the hydrofoil is penalized through having to carry the weight of the foil system. The SES has to carry fans and ducts which use up space.

3.4. PERFORMANCE IN A SEA WAY.

The way in which an SES traverses a rough sea also requires comment in this section. It would be very helpful if curves simiiar to those given in fig.2 could be prepared for all the craft of ref.3. These might show, for example, speed plotted against significant wave height. Unfortunately this data is not available in a consistent and comparable form. Full scale trials in rough weather are notoriously more difficult to carry out than appears at first sight, and very little data has been published. However, ref. 13 provides information from which it is possible to create one plot of the desired form. The craft concerned was the SES 200 and the speed was limited to that which the commanding officer thought prudent. Rather surprisingly, in head seas, the speed achieved varied only ±1.5 knots over a range of significant heights from 1.9 to 8.2 ft. Unfortunately it is not possible to be sure of the criteria used by the commanding officer when he deliberately reduced the craft speed, so that this data, while of great practical value, is too readily capable of misinterpretation.

Ideally, the waves should lift the seals and pass through the cushion with very little disturbance to the craft itself. Unfortunately, this is not what happens, and the mechanism of wave pumping when the cushion pressure varies with the changing volume of the cushion as it passes over the wave, has long been recognised. Furthermore, there appears to be a threshold value of wave height, above which, water impacts the bottom of the SES centre body at the bows and the acceleration response to the waves markedly deteriorates.

Research into a ride control system was commenced in the USA over 15 years ago and, apparently, was successfully demonstrated on the US Navy's SES 100B craft using a cushion air dump valve. Recently, a "pneumatic" ride control system was retrofitted to the SES Norcat and improvements in seakeeping were reported.

In the case of the SES 200 as reported in ref. 13, the ride control had little effect on the speed achieved in head seas. However, it did appear to make the ride less harsh, although this effect was not confirmed by the measurements taken.

The majority of SES currently operating, do not have a ride control system.

4. OBJECTIVES FOR RESEARCH AND DEVELOPMENT.

In the light of the overall survey given in the preceding section of this paper it is possible to draw only a limited number of conclusions as to the directions which future research should take. However the evidence can be supplemented by reports from individual craft together with a background knowledge of the subject so that a series of objectives for future SES research and development can be proposed.

4.1. PRELIMINARY.

Two preliminary general comments are worth making concerning the aims for SES research and development.

Fundamentally, the purpose of any air cushion vehicle research programme should be to generate understanding and improvement - the one generally leading to the other. Unfortunately, commercial pressures often dictate that research is carried out to devise "fixes" for immediate problems without obtaining an adequate understanding of the phenomena involved. However, a good theoretical understanding is essential in order to make real progress. Sadly, this has all too often been left to be provided by academics in some University, working several years downstream of the action. This is not only inefficient, but not good for the researcher's morale. Furthermore, without proper theoretical backing, development can easily progress down blind alleys with disastrous results.

The other general comment , which cannot be made too strongly is that, leaving aside military usage, the SES operates in a very commercially competitive field. One might have concluded from figs.1,2 and 3 that, at least in calm water or in moderate seas, the SES is obviously better than it's competitors. In fact there are strongly held views that this is not so. The author found it salutary to discuss the issue with those involved in Sales and in the financial aspects of high speed water borne transportation. Their view was that the catamaran posed a serious threat because it was simple, relatively cheap, higher speeds than those currently being obtained were possible, and that a ride control system could just as easily be applied to the catamaran as to the SES. On the other hand, the SES was too complicated, the first cost was too high and the maintenance costs were also too high. It was felt that the SES was basically a rectangular box supported between thin hulls and it was surprising that first costs were not lower.

It would appear from this, that it is essential to carry out research and development in order to reduce costs. However, the author feels that this may not be wholly a technical problem, and it is suggested that SES manufacturers will have to take care with their pricing structures if the commercial SES is to have the kind of future that it deserves.

Turning now to the details of future research and development, it is generally agreed that the only really new technology required for the SES is the seal or skirt. Therefore it will be convenient to divide the recommended items for R & D into two groups, the first dealing with the seals.

4.2. OBJECTIVE> FOR SEAL RESEARCH AND DEVELOPMENT.

1. Development of new geometries for passive seals in order to obtain better wave contouring and "tuck under" performance in various sea states.

2. Reduction of seal drag possibly through air lubrication and a better detailed understanding of how the seal drag arises.

3. Development of active seals. Ideally, the seals should respond to the proximity of the water surface to give a roughly uniform rate of air leakage. This may prove to be impossible to achieve without some form of active control, perhaps using transducers to monitor the waves just ahead of the seal. Again, ideally, the movement of the seal should not adversely affect the pitch or roll stiffnesses significantly.

4. Improvement of seal life by reduction of wear, better material strength, and resistance to material delamination, cracking and fatigue of the substratum material

5. Improvement in the ease of maintenance and replacement of seals with the SES in the water. Easier and cheaper repair. Development of better and easier bonding techniques. There is no doubt that a great deal has already been accomplished in this area., but further improvements would help to reduce operating costs.

6. Further development of non flexible or semiflexible seals for larger craft.

7. Reduction of manufacturing costs.

4.3. AIMS FOR OTHER RESEARCH AND DEVELOPMENT.

1. Propulsion/appendage studies. The appendage drag associated with the use of propellers on SES, has undoubtedly been very high. This has been largely due to the high shaft angles that have been employed. These high angles also cause other undesirable features such as loss of thrust, vibration, etc. While the appendage drag can be greatly reduced by using a water jet, the author believes that a propeller with a low shaft angle (perhaps associated with a tunnel), together with carefully designed brackets, will still provide a better overall efficiency, at least at speeds presently envisaged for commercial craft. Research will obviously be required to

see if a propeller tunnel can be made compatible with the rquirement to avoid venting. The application of some form of Z drive may well be the best solution.

2. Reduction of total resistance to obtain better speed/range. A requirement for research aimed at reducing resistance does not arise strongly from the preceding sections of this paper, but it may be that significant reductions can be obtained without too great a cost, and the possibilities should be explored. As an example, the SES has plentiful supplies of air at moderate pressure and this air could conceivably be employed to air lubricate the seals or the side hulls so that a significant reduction of drag could be obtained.

3. A low length/beam ratio is desirable for smaller craft designed for high speeds. This poses a problem in that the associated hump drag is large and sensitive to craft weight overload and may lead to a more complicated and expensive propulsion system. However, traversing an otherwise insuperable hump may be found to be possible by means of a judicious variation of the cushion pressure with speed. Tests are required to provide data for establishing optimum compromise with varying cushion pressure and trim control,

4. The performance of some SES are reported as being much more sensitive to L.C.G. position than others. This situation needs to be clarified and the necessity for trim control by means of hydrofoils or planing flaps may arise out of this.

5. Improvement of ride quality to permit higher speeds in waves with adequate passenger comfort is believed to be a very important requirement. Severe slamming can occur when the sea severity exceeds a certain threshold value, particularly if the forward seal or the shaping of the bows fails to alleviate the effect of the wave impact. Such slamming generally occurs at synchronism in pitch. Development of devices to increase the pitch damping is desirable, This may involve further development of active cushion venting systems and/or active fans, or hybrid concepts with active hydrofoils, or planing surfaces added to the basic SES.

6. Further work is required to investigate roll stability standards over the complete speed range in waves in deep, and shallow water.

7. Computer studies, with model test validification, are required to investigate the performance and stability of multi cushion craft (Catamaran, trimaran, four cushions, etc.) with a view to obtaining deep cushion craft with good sea keeping characteristics with adequate transverse stability.

8. Better off-cushion capability.

9. Safety.

10. Simplification of construction to permit reduced first cost.

5. RESEARCH AND DEVELOPMENT RESOURCES AND METHODS.

The resources and methods necessary to achieve the research objectives described in the preceding section of this paper will now be discussed.

The main resources are as follows:-

1. Towing tank.

A towing tank is required with a high speed capability and sufficient length to permit a reasonable time at the desired speed so that accurate measurements can be obtained.
The speed requirement is illustrated in the following table:-

Full scale SES length in ft..........	50	70	100	250	700
Assumed max. full scale speed in kts...	50	60	70	60	60
Model length in ft...	7	7	7	10	10
Max. tank speed in ft/sec............	32	32	32	21	12

To give some physical meaning to the table it may be noted that a tank having a 32ft/sec. capability with a 10 second test time at speed, an effective carriage acceleration and deceleration of say 0.2g, and an emergency braking length of 150 ft would have a total length of 650 ft.

The towing tank should have a wave maker for testing in regular and irregular waves and it should be possible to drive the carriage at various controlled speeds in both directions, so that tests in following seas can be carried out.

Ideally the carriage itself should have no aerodynamic effects on the model.

2. Manoeuvring basin.

The basin must be large enough to accommodate the desired manoeuvres. It should have wave makers which can produce controlled and repeatable long and short crested waves. It should also be large enough such that a sufficient number of wave encounters occur during a test run so as to permit an adequate assessment of the model's response characteristics.

3. Wind tunnel.

It will prove convenient if a wind tunnel is available which is large enough to permit the towing tank model to be mounted in it on a ground board to represent the sea. Special wind tunnel models will have to be made if the available facilities are too small to allow this, Other test facilities will be touched on later in this paper.

The methods of testing will now be discussed under the headings of modelling and full scale testing.

5.1. MODELLING.

5.1.1. PROBLEMS OF SCALING.

The method of testing an orthodox ship model in a towing or manoeuvering tank has a long and fruitful history, and it has been successfully employed with air cushion vehicles of all types. As in the case of the ship model, Froude scaling is mandatory because the SES makes waves. Suitable corrections are made to account for the fact that viscous components of resistance are dependent on the Reynold's Number.

In the case of air cushion vehicles, it is also necessary to consider the fact that the atmospheric pressure in the model regime is not Froude scaled. There are very few towing tank facilities in the world where the atmospheric pressure can be controlled (e.g. NSMB, Krylov.) but these are expensive and difficult to use. Thus, in the majority of test facilities it must be accepted that the absolute pressure and compressibility of the cushion is not correctly scaled, although the difference in pressure between the cushion and the atmosphere in the model regime produces the correct scale lift force.

The High Speed Marine Vehicle Committee of the ITTC (HSMVC) has addressed this problem, and has published a description of a method developed by the present author whereby a sealed volume within the skirt system may be made to obey Froude scaling requirements. The method employs a spring loaded concertina type pressure compensator. (Page 180 of ref.4) Such sealed volumes have been employed on full scale craft to reduce impacts in heavy seas, and where they are employed in a full scale prototype, the use of pressure compensators on the model is absolutely necessary.

More recently, the HSMVC has concluded that the use of model data to predict the full scale heave accelerations at the heave natural frequency is not appropriate (Ref.5)

The heave natural frequency in Hz is given (as in ref.6) by:-

$$f = (1/2\pi). \left[\frac{sh.g(1 + At/pc)}{h}\right]^{1/2}$$

where:-

sh is the specific heat ratio of air.
At is the atmospheric pressure.
pc is the cushion pressure.
h is the cushion depth.

This expression is clearly not consistent with the Froude scaling law. Thus the heave natural frequency measured on a model, when Froude scaled to the full scale regime will be different from the actual full scale value.

The reply by the HSMVC to comments made at the 17th. ITTC (Ref.7) is relevant and is reproduced below:-

> "Data contained in the HSMVC report to the 16th. ITTC as well as other unpublished data show that model data adequately predict the motions of SES in the 100 to 200 ton size. However, full scale data verifying the existence of the heave pressure mode natural frequency were obtained on two ships of this size by driving the

ride control system at the specified frequencies while in relatively calm water and recording the resulting pressure and acceleration oscillations. The heave-pressure mode natural frequency for similar craft are generally higher than the encounter frequency of most seas experienced. This however will not be tne case for multi 1000 ton SES."

Thus, while the predicted heave natural frequency undoubtedly exists, it will only be of importance for large craft. This may be illustrated by considering a 100 ton SES with a cushion depth of 6 ft. and a cushion pressure of 75 lb/sqr.ft. This craft can be Froude scaled up to larger sizes and the heave pressure mode natural frequency calculated in each case. The results in terms of natural period are shown below:-

Craft weight in tons.	100	1000	10000
Natural heave period in secs.	0.4	0.9	1.9

Tuning with large amplitude waves will not occur for the shorter periods although the effect of small waves may be discerned as a rough ride called "cobblestoning" by both SES and hovercraft operators. The problem is considered at length by Afremov (Ref.8). Lest it be felt that the problems of scaling are overwhelming, it should be said that the pitch motions and the low frequency heave motions of the model scale up very well.

A ride control system, in effect, improves the heave damping and will partially smooth the craft response, particularly at it's natural heave period. Various methods have been employed. Some have used dump valves in which case cushion pressure has been thrown away and extra fan power is then required to compensate. In other cases, the fan itself has been made to respond dynamically to the cushion requirement, This latter method is obviously better from an overall efficiency point of view but it has been found difficult to ensure that; the system responds quickly enough, and it may well be expensive from a first cost and maintenance point of view. Another proposal is to employ both these methods together. All these systems can be easily controlled by using a dedicated microprocessor provided with suitable inputs from the craft's behaviour.

There is no real problem in simulating the mechanical part of the ride control on the model but its effect will be modified by the considerations of natural heave period as discussed above.

5.1.2. MODEL DESIGN.

E.G. Stout, then of Consolidated Aircraft Corporation, gave a fine definition of one of the model types which will be required for SES research as follows:-

> The completely dynamic model is defined as a complex integrating mechanism that automatically picks up every known or unsuspected force, in the proper magnitude, point of application, direction and sequence, integrates all these reactions instantaneously, and provides the observer with the resultant motion and rate.

While such a perfect model may be not possible, the important point is that practical dynamic models will respond to unsuspected forces which would never have been incorporated into a theoretical analysis or a computer model, simply because they were unsuspected.

Amongst other things, the dynamic model must not only have the correct scale dimensions and weights, but it must also have the correct scale moments of inertia. The seals must also have the correct weights and stiffnesses

This latter requirement may well prove difficult to meet. If we consider a 1/12th. scale model of a 100 ton SES the scale material weight would be about 12 oz./sqr.yard. An off-the-shelf proofed fabric of this weight would have a bending stiffness far greater than the scale value. One crude solution which has been employed is to use a much lighter material having the required stiffness, and to increase it's weight by means of lead pellets distributed over it's surface. A much better solution has been to commission the special manufacture of a very light weight nylon fabric, coated with a soft synthetic rubber in order to obtain both the desired weight and stiffness. Methods of determining the stiffness experimentally are given in section 3.6.8. of Ref.4.

As a point of model design philosophy, the writer is a firm believer in the principle of constructing models primarily intended for tank testing in such a way that they can also be taken out to sea and run in really rough conditions. If such tests might be contemplated at a later date, then it is imperative that the model be designed with this in mind from the very beginning. In this case it will have provision for different motors and services built in, it will be of a rather more rugged construction and be designed to survive a corrosive salt water environment. It will also have provision for radio control and on-board recording facilities for the instrumentation. None of these features militates against the model's successful employment simply as a towing tank model, so the expenditure of the extra design time required to provide these features may be considered prudent.

As has been inferred already, the capabilities of the towing tank in which the model is to be tested, largely determine the scale to be employed. It is perhaps worth mentioning again, that the maximum reverse carriage speed is an important parameter since high speed tests will be required in following seas. Also, as a rough rule, the model length should be less than twice the tank depth.

As far as the model construction is concerned, it will be desirable to employ materials having a high strength/weight ratio. The author's experience is that closed cell foam covered with GRP provides an excellent solution. Chines will have to be made sharp, and reinforced so as to be tough enough to stay sharp. The superstructure will also have to be represented fairly faithfully in order to obtain the correct aerodynamic lift and drag forces together with the correct aerodynamic moments. This approach will also help to ensure that the flow into the air intakes is also reasonably representative.

The fans, intakes and internal ducting should be built to scale although it may be found necessary to run the fans a little faster than the scale speed in order to obtain the required output. It will also be desirable to get. the pressure/airflow rate curve as near correct as possible. The model designer wiii also bear in mind that the gyroscopic effects of the rotating masses in the model should be reasonably similar to those occuring full scale so that manoeuvering characteristics will not be unduly influenced.

As far as propulsion is concerned, there is a lot to be said for the school of thought that feels that in the case of the SES, the model propulsion system is only there to push the model along at the required speed. This is probably fair enough for initial studies, but eventually a representative system will have to be installed, particularly if semi submerged propellers or water jets are to be used in the full scale regime. Very advanced and esoteric propulsors are more likely to be proposed for use on SES than for most other kinds of craft, and the water tunnel is an essential tool to [illegible] development, especially since ventilation or cavitation is likely to be present.

In the case of the water jet, variable geometry inlets may be considered for advanced designs and both tank and tunnel work will be required.

5.1.3. MODEL TESTING.

The author is strongly of the opinion that the SES model should be tested in the towing tank with a towing arrangement which **allows** it to surge freely. This is particularly important in the case of head sea tests in large waves, where any tendency to dive will result in sudden decelerations - possibly followed by a recovery. For following sea tests, the surge movement may be quite large and must be allowed to develop freely. Also, in the **case** of safety tests, it may be necessary to simulate engine failure or bow skirt failure, in which case the free-to-surge rig will be vital.

The frontispiece of this paper shows a dynamic model SES being tested in a towing tank on a 30 ft. free to surge rig.

For manoeuvering tests, and tests in waves in other than head or following seas, the availability of a warm, dry, brightly lit manoeuvering basin with wave makers and all the necessary instrumentation is a tremendous **asset. However** the SES is fundamentally **a fast** ship, and even quite large facilities may not be large enough to accommodate the required acceleration distance together with an adequate number of wave encounters. The only alternative then is to conduct the tests out at sea.

5.1.4. DATA.

Much of the analysis of SES model data is sufficiently similar to that employed with other high speed craft not to warrant comment here. In the case of the seals, however, there are considerable problems and differences between the methods employed by various practitioners. Thus, Wilson **(Ref.9)** suggests that the theoretically **calculated** cushion resistance using Doctor's method **(Ref.2)**, together with the frictional resistance of the hulls, should be added to the air resistance (determined by means of special runs in the tank or by wind tunnel tests). This total should then be subtracted from the total measured resistance to yield a nominal seal drag. Wilson goes on to present a method of correcting this derived seal drag to give a full scale value. Ref.9 shows that this method gives good correlation with full **scale** results for the cases investigated.

Other workers have assumed that there is no scale effect on the seals and have taken refuge in an overall correlation coefficient (the easy way out!) while others have sought to estimate the seal wetted area and make a friction correction based on that. The writer confesses a predilection for this last method, even though it is very difficult to measure the wetted area in practice.

Fig.4a shows the results of **scaling** the results of towing tank model data to full scale. The data is for four 100 ton craft and the cushion pressure was maintained nominally constant throughout. It will be seen that the lower length to beam ratio was associated with a **pronounced** hump in the resistance curve. This hump decreased in magnitude as the L/B increased. At the higher speeds the lower L/B showed a slight advantage. The curves of **fig.4a** are not representative of some modern practice because the proportion of the total weight carried by the hulls was unusually high. Other test results show that the low L/B can have a **larger** advantage at high speeds. However, the hump has to be traversed before the high speeds are reached and so a suitable design compromise is required as to the choice of L/B.
The model data of **fig.4a** can, of course, be scaled to give the **characteristics** for a craft of any other weight. For a 1000 ton craft the resistance hump speed for L/B = 2 occurs at about 27 knots and the craft with L/B = 5 gives a much lower resistance up to 50-60 knots.

Thus, for a 1000 ton craft and a design speed of say 45 knots the adoption of the larger L/B would at first sight appear to be most desirable. However, if a reasonable cushion depth is also required so that the craft can operate in large **waves**, the associated C.G. height/ Beam ratio becomes so large that the the transverse stability is called into question. Ref.10 proposed a catamaran SES to overcome this problem and **concluded** that such a craft, with a 790 ton displacement could operate in **a** Sea State 5 at a speed of 20 knots with motions and accelerations which were sufficiently low such that a helicopter could be operated from it.

There is *no* reason why the SES catamaran concept cannot be extrapolated to a trimaran, or even a **four** cushion craft, with the cushions being adjusted to suit the current conditions or performance requirements.

Fig.4b shows another example of the towing tank's output, namely the so called added resistance due to **waves** for an SES with an L/B of 3. A complete set of results clearly would permit the maximum speed in waves to be estimated for a given engine installation, propulsion system, etc.

5.1.5. MODEL TESTING AT SEA.

The concept of testing models at sea has already been touched on. The technique allows the models to be tested in a realistic environment and also in very rough conditions with short crested seas which could not be created in a towing tank or manoeuvering basin.

The technique is difficult and rather expensive, but it can be extremely rewarding. The author has been personally associated with the testing of twelve different models of air cushion vehicles at sea, beside some 40 fast ship models, flying boats and record breaking hydroplanes. The sizes ranged from about 4 feet in the case of the hydroplane to some 36 feet with a displacement of over 2.5 tons in the case of warship models.

Models to be run at sea must be designed so that they can either carry a man or be operated entirely by radio control.

In the case of the manned model, the weight of the man must be taken into account, and it is also important that he be strapped in, because the human body is surprisingly efficient at using small involuntary weight shifts to damp out unpleasant motions. As can be imagined, test personnel do not like-this very much! The manned model must also be supplied with all the appropriate safety gear such as radar' reflectors, bilge sniffers, etc. A disadvantage of the manned model is that it cannot be employed in dangerous experiments where, for example, the craft might turn over. However, this can be done with a radio controlled model with, a great deal more equanimity.

Fig.5a shows a manned model underway.

In the case of the radio controlled model, access to the interior of the model itself during the trials is undesirable except to rectify mechanical or electrical failures. To illustrate the practical nature of the problems to be faced, the following list shows some of the functions which will have to be remotely controlled through the radio link:-

to

1. Rudder.
2. Autopilot on/off.
3. Throttle.
4. Data Recorder on/off.
5. Bilge pump on/off.
6. Cooling water on/off.
7. Electric starter.
8. Choke.

In addition to this, waterproof manual switches will be mounted on the superstructure in order that the main batteries and the radio control batteries may be isolated when the model is being transported to the test site, and so that the gyro can be erected prior to a sortie.

The general philosophy adopted for manned or radio controlled model tests is similar to that for a full scale ship, in that it must be accepted that it will not be possible to specify beforehand the precise conditions under which the tests will be conducted. However, reference to local weather forecasts, and a suitable choice of location will generally result in the tests being conducted under conditions which fall between specified limits.

The sea state actually encountered during the tests can be measured by using small small wave buoys.

The data obtained during the tests can either be telemetered to the accompanying parent vessel or recorded on board the model. Experience of both methods leads the author to suggest that the second method is more reliable, and reliability is particularly important in this kind of work.

Subsequent to the trials, a spectral analysis can be carried out and the root mean square of each of the measured parameters can be plotted against the ship's heading relative to the dominant wave direction, for each sea state investigated.

Fig.4c illustrates the kind of data which can be obtained. In this case it is the RMS vertical bow acceleration, measured at 20 knots, plotted against the craft's heading relative to the dominant wave direction, for two significant wave heights. A typical sortie would provide similar plots showing the RMS variation of the following:-

Pitch angle.
Yaw angle.
Roll angle.
Forward vertical acceleration.
Forward lateral acceleration.
Aft vertical acceleration.
Aft lateral acceleration.
Rudder angle.

Another kind of output from such trials which is very useful to a design department is shown in fig.4d. This gives the distribution of maximum vertical acceleration over the hull length.

In the very realistic conditions that pertain at sea, extremely interesting and very valuable film records can be obtained. A television camera has also been located at bridge height on the model to obtain records of wetting, etc.

One of the most exciting possibilities associated with testing at sea, is that of running two competitive designs side by side. With experienced seamen on the parent vessel who can also take the helm of the radio controlled models, very useful and practical lessons can be learned, particularly in the area of handling in very rough weather.

5.1.6. OTHER MODEL TECHNIQUES.

Seal, or skirt systems have been the subject of intensive research in the past, and more needs to be done. For this purpose it is clearly not efficient to use a model of the complete craft which could otherwise be employed say, in towing tank tests. In any case a rather larger scale is called for. Thus, a special seal rig can be constructed as shown in fig.5b. The rig consists of an open fronted box with perspex sides and viewing panels in the floor and back. The rig incorporates a movable ground board to represent the sea and has a suitable air supply system. The rig can be used to investigate the inflated geometry of the seal, the air flow required, and the dynamic stability and response characteristics.

Seal dynamic stability can be an important issue. The author recalls riding on an early version of a large SES in 1980 when it was noted that there was an unpleasant vibration of about 25 Hz. which was particularly noticeable near the transom. Behind the boat there was superimposed on the wake a series of lateral waves with a wave length corresponding to the vibration frequency. It was clear that the rear seal was unstable and was flailing the water surface (and probably the bottom of the centre body **as** well!). "Fixes" for this kind of phenomenon are best derived from investigations in a seal rig **as** described above.

Fig.6 shows the results of development work in such a rig. **Fig.6a** shows a single finger pattern, several of which can be folded to produce the simple bow finger system shown in **Fig.6b**. Various seal designs have attempted to increase the vertical "spring rate" as the seal is immersed. **Fig.6c** shows a finger design intended to do this. Air is fed into the central smaller finger and this provides an increased stiffness when it becomes immersed. **Fig.6d** illustrates the results of an attempt to design a simpler molded finger. Air is fed down the tube and escapes from a nozzle at it's bottom. The single web provides stability.

Fig.6e shows a typical bag-finger combination. The bag runs at a higher pressure than that of the cushion and so provides a measure of increased stiffness when it contacts the water surface. **Fig.6e** also shows a membrane AB. This membrane may be **provided** to prevent the bag from developing a low **frequency** oscillation which could result in structural damage. Ref.11 shows that this oscillation did not occur in the case of **a bag** without **a** membrane **at** higher values of the bag prsssure /cushion pressure **ratio,** but it started up if the **ratio** were decreased to lower and more desirable values.

The membrane may be fitted with **a** non return **valve as shown** in **fig.6e** In this case **a severe** water **impact** which might sweep the bag and finger system back under the craft, would also attempt to force the air out of the front part of the bag and would thereby close the **valve.** This, in turn, would generate a sealed volume which would provide a shock absorbing cushion for the structure even if all the rest of the air in the bag were forced out.

Fig.6f shows **a** typical rear seal for an SES which **was** developed in **a seal rig.**

The rig can also be invaluable **in** the development of active flexible seal systems which respond to the proximity of the **sea.**

Another interesting model technique is that which employs a moving ground with the model being held stationary above it. This approach **has** obvious limitations but it permits the study of response characteristics with a relatively small capital outlay. Also in this category, comes the use of rotating arms which have also been used to study response characteristics.

Fans have been subject to a great deal of development testing in the past and suitable model test rigs to investigate fan characteristics can be devised. The author feels that such rigs **are** invaluable for the calibration of fans for tank models and the detailed development of active fans. However, most of the basic work on orthodox fans has now been done.

Lastly, under the heading of modelling, it must be **recognised** that the computer model provides an invaluable tool for the researcher and the designer. However, it should **be said, that** particularly in the early stages of the development of **a new** form, the computer model can be misleading. By the way of illustration, in the early days of the air cushion vehicle, each new dynamic model of each new project seemed to uncover a new and unexpected phenomenon. The mechanism of the phenomenon and the forces involved were unknown until the model tests uncovered them, and the computer model would not **have** revealed their existence at that time. Of course, as experience is gained the computer model becomes more comprehensive and reliable. However it will by no means supplant the testing of models of new high speed craft, at least in the foreseeable future.

5.2. FULL SCALE TESTING.

The full **scale** trials of an SES are sufficiently similar to those of other high speed craft not to warrant much comment here except to **say** that they should be done in conformity with **an** accepted standard procedure, and in 'particular, should be carried out in deep water.

Reliable information on full scale performance in waves, obtained from properly instrumentated trials conducted by impartial observers is very much needed.

Another **area where** full **scale** tests have been, and continue to be of considerable importance, is that of the **seal. Seals may** be either flexible, semi flexible or "rigid". Only the first two are currently being employed **as** far **as** the author knows, and these will be briefly discussed below.

5.2.1. FLEXIBLE SEALS.

Nearly all the flexible seals which have been employed in the various kinds of air cushion vehicle have consisted of **a** basic substratum fabric which provides the strength, and a rubber or **plastic coating** to protect it. One of the most important problems has been to obtain a satisfactory bond between the fabric and the coating since loss of the coating invariably resulted in the subsequent destruction of the fabric.

Initially, orthodox methods of materials testing were employed to assist in the choice of materials having good tensile strength, tear strength, peel strength, etc. but it soon became clear that materials which appeared to have quite good characteristics did not give a good life in service. Fig. 25 of Ref.11 shows the frightful state of some early fingers on a hovercraft after only three hours operation. Some of the fingers were completely destroyed by a process of delamination which appeared to be caused by a high frequency oscillation. This became known as flagellation. To obtain a solution to this problem it was clear that it was necessary to recreate in a laboratory environment, the finger wear which had been observed on the craft, so that those features which caused the wear could be studied and a cure developed.

This proved to be surprisingly difficult to do, since all the orthodox approaches failed to produce the kind of delamination which had been found on the craft, Ultimately it was discovered that a specimen of seal material mounted in the nozzle of a blower could be made to flap like a flag and under certain conditions would produce delamination very similar to that which occurred on the craft'. When the air velocity was increased, the rate of wear was increased, so that months of wear could be simulated in a few hours. Thus an effective laboratory method was found which could be employed to examine and develop new materials having much longer life. One set of fingers were found to suffer very high accelerations of the order of 4000g at frequencies which exceeded 200Hz. and this certainly accelerated the wear!

Various substratum woven fabric materials were tried in the flagellation rig and on full scale craft such as Nylon, Kevlar, carbon fibre and various species of wire netting (some of which looked like chain mail), Coating materials such as PVC, nitrile, neoprene and various kinds of natural rubber were also investigated. As a result of all this, finger life improved dramatically and as the flagellation resistance of the materials improved, it became necessary to look at other sources and types of wear. Thus, erosion, abrasion, cracking, fatigue and failure of finger attachments all came in for investigation. Developed fingers were found to have lives which were a function of speed and cushion pressure, but none the less, had an acceptable life. In this connection, it has recently been reported that the SES Fjordkcngen requires no seal maintenance between its annual refurbishing.(Ref.12) However, it is uncertain whether this refers to the bow seals as well as to the stern.

As larger and larger SES are built with higher cushion pressures and higher speeds, it seems inevitable that flexible skirts will require more development work, and some spectacular advances may be hoped for. In this connection, it is interesting to speculate as to the weight; of flexible seal material that might be required for larger craft. A crude extrapolation on a basis of plotting present values of weight per square foot of material against the cube root of the craft weight suggests 2 lb. per square foot for a 1000 ton craft and 4 lb. per square foot for a 10000 ton craft. This implies that the bow seal on the larger vessel would consist of 25 to 30 tons of rubber or more, which is currently a bit daunting! Whatever the real values turn out to be, future development work will be based on an accumulation of considerable experience, and it may be that combinations of, say, Nylon and steel will find acceptance with other highly developed coating materials.

5.2.2. SEMIFLEXIBLE SEALS.

Semiflexible seals have components which are rigid and able to sustain bending and compressive loads. Flexibility is brought about by means of hinges and still further flexibility may be conferred by adding areas of flexible material. Semiflexible seals may generate planing forces and may also be subject to rather large impact loads in heavy seas. They do, however, avoid flagellation problems. Most semiflexible seals constructed so far, have been fabricated from GRP and the potential for employment of this kind of structure with much larger SES looks promising. However, full scale experience is limited so far.

6. CONCLUSIONS.

The SES has progressed rapidly from its inception just over two decades ago to the point where it appears to be very competitive with other forms of high speed transportation on water. Most of the basic technology required to achieve this remarkable advance, together with most of the necessary research and development resources, already existed within the aeronautical and naval architecture worlds.

The new technology required has been concerned primarily with the seals or skirts, and here again, rapid progress has been made.

Further work is now required to assure the success of the commercial SES in the future, and this paper has high lighted the need for reduced costs and improved seakeeping with adequate roll stability. A careful assessment is also required of the relative merits of the best form of propeller drive and water jet

It is believed that further research and development can maintain the momentum already given to the SES concept by past work, snd that further significant advances can be made.

The paper has concentrated on commercial operations, but the application of the SES principle to a number of military tasks appears to be attractive.

7. ACKNOWLEDGEMENTS.

The author wishes to express his gratitude to Hovermarine International Ltd. for permission to publish the frontispiece and fig. 5a, and to British Hovercraft Corporation for permission to publish Fig. 5b.

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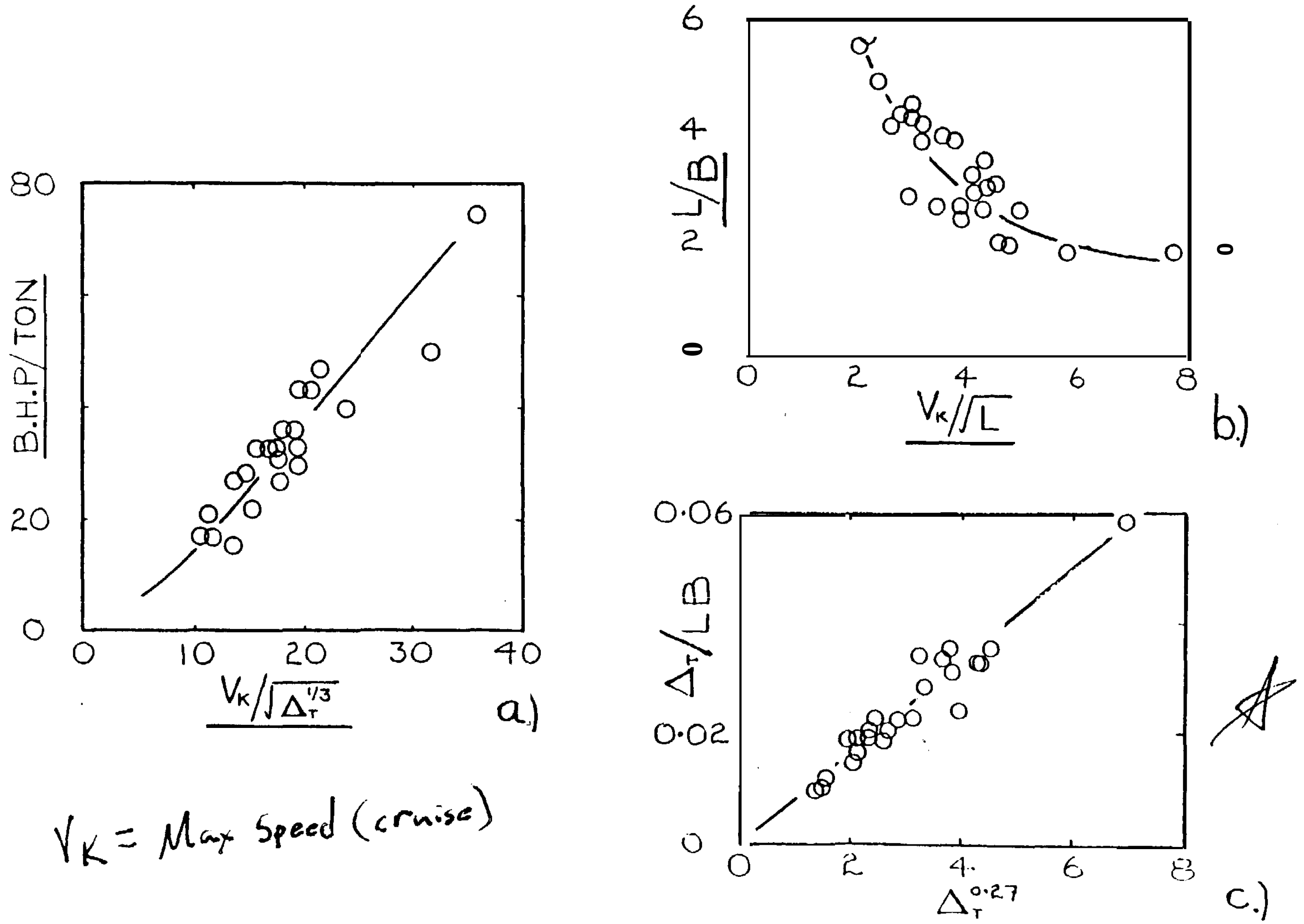
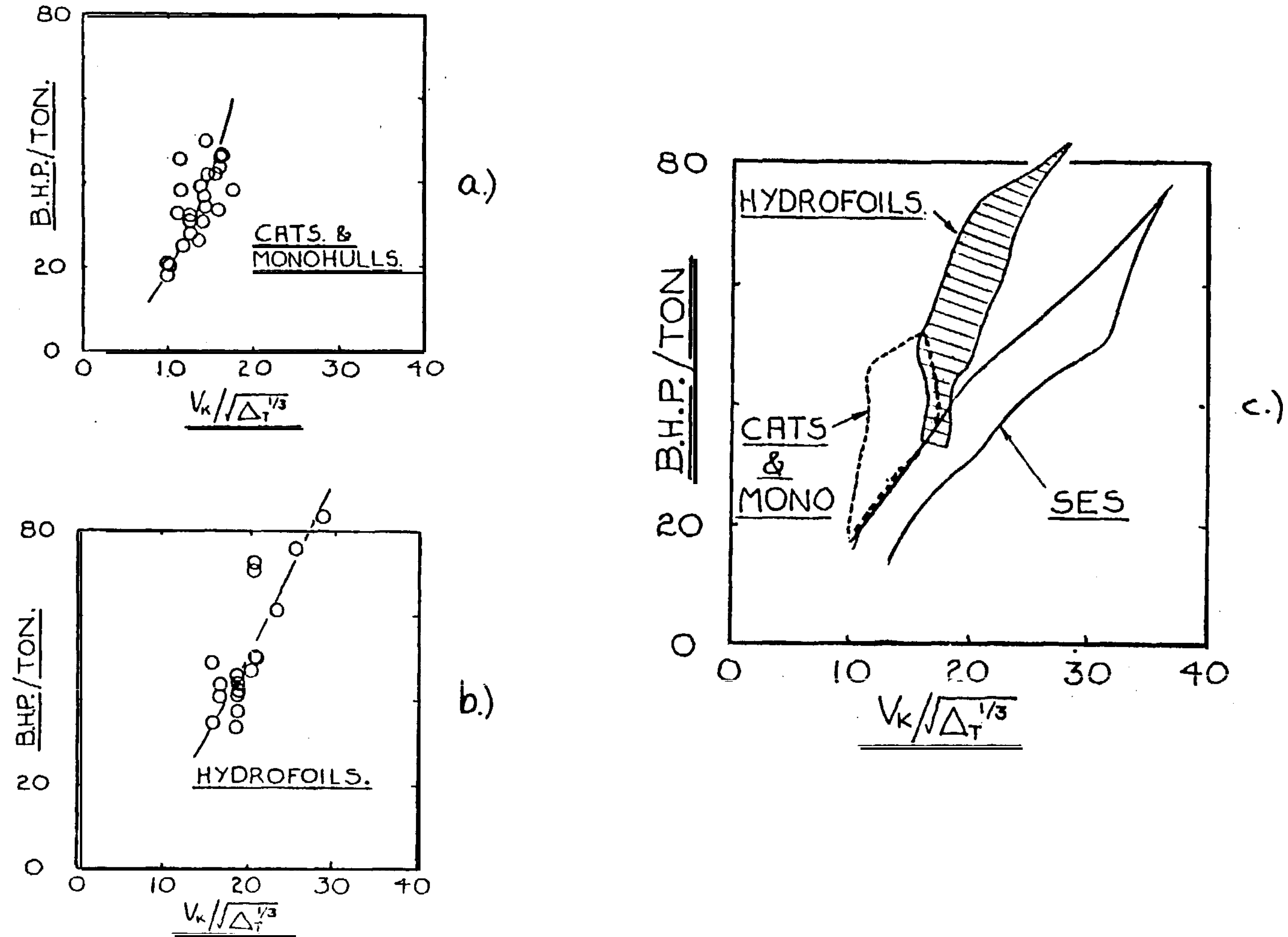


V_K = Max Speed (cruise)

SES CHARACTERISTICS. FIG.1.

From Amyin 87 Surface ...

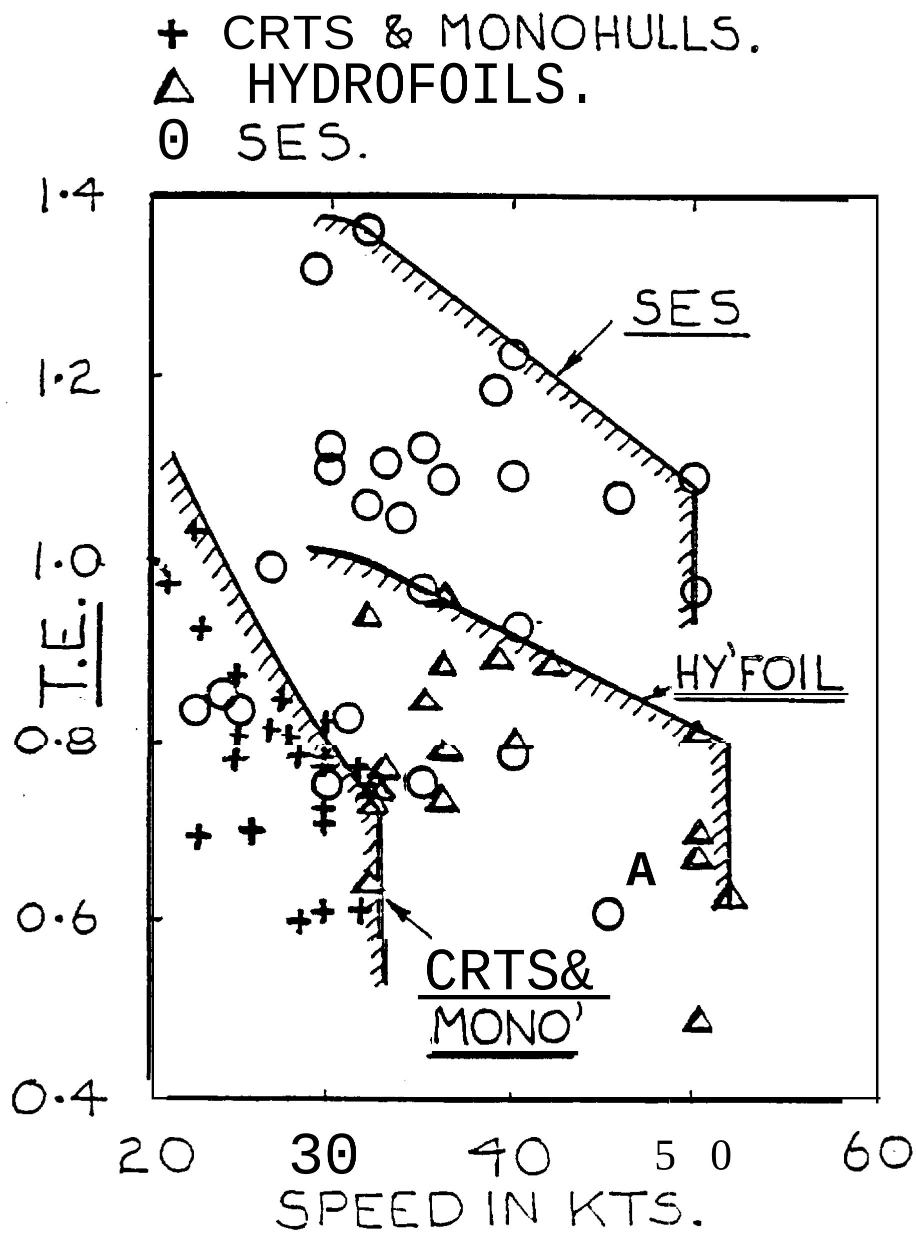


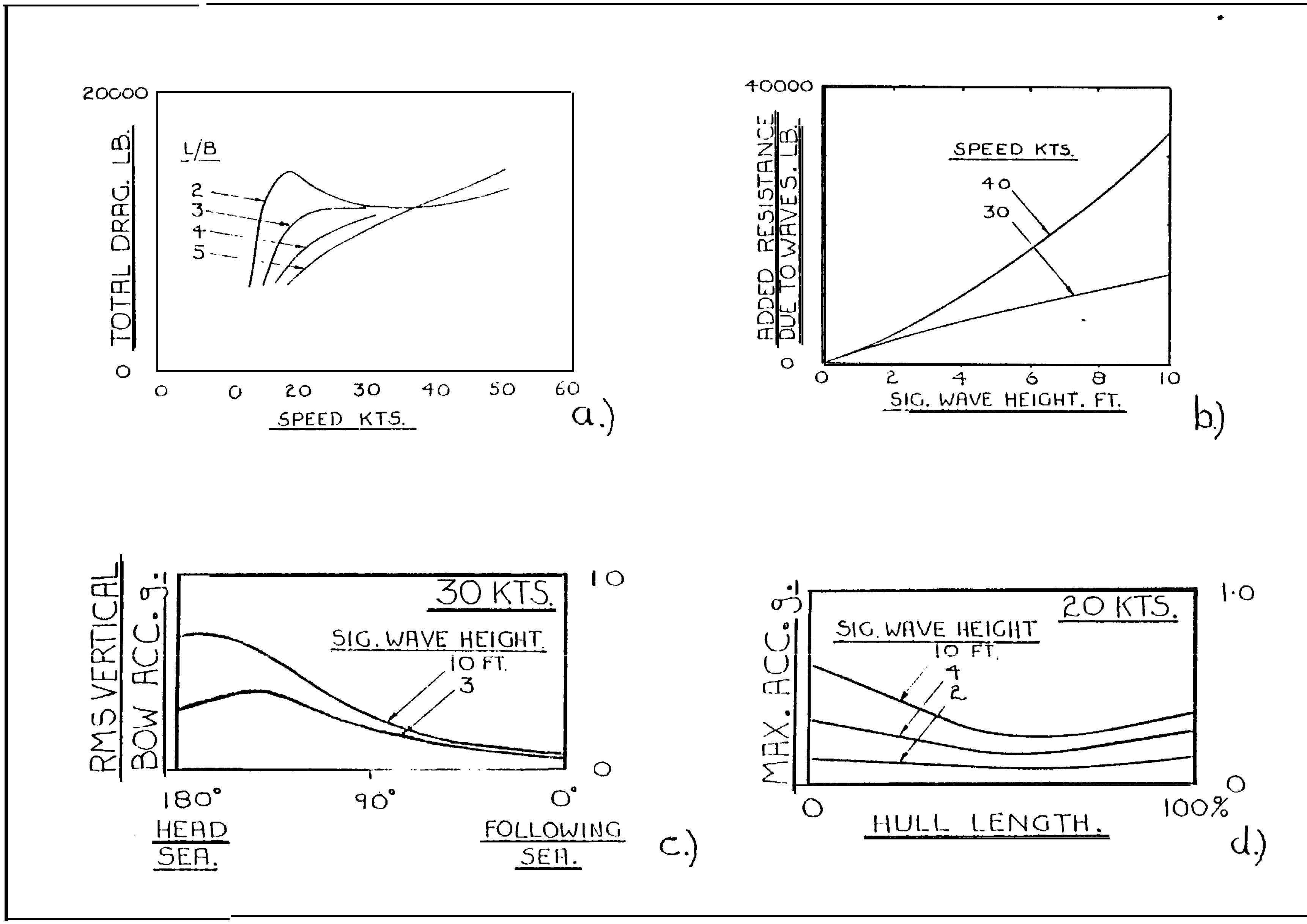
CRAFT COMPARISONS.

FIG. 2.

Deff on Pg. 5

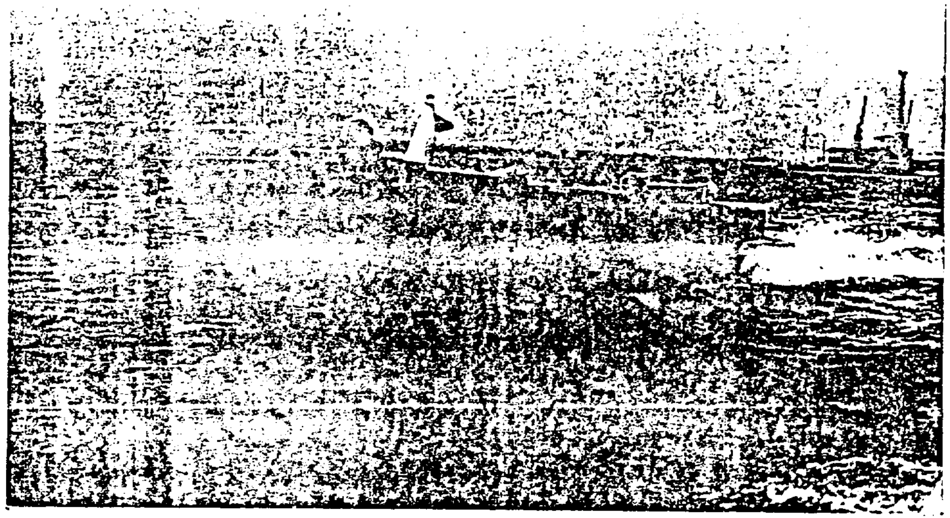
TRANSPORT EFFICIENCY. FIG. 3.



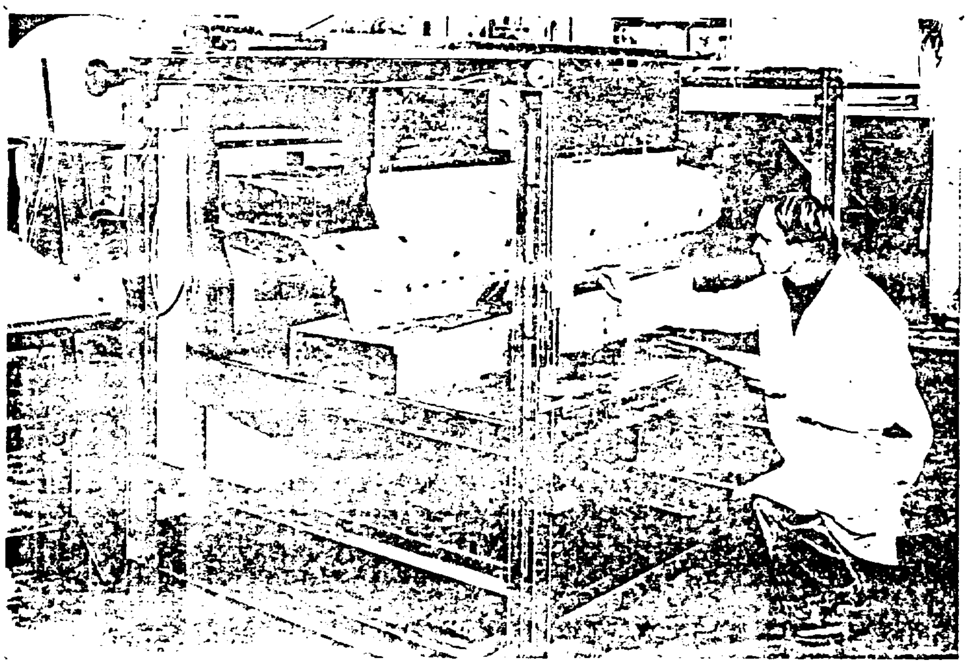


RESULTS OF MODEL TESTS. FIG. 4.

MODELS FIG.5.

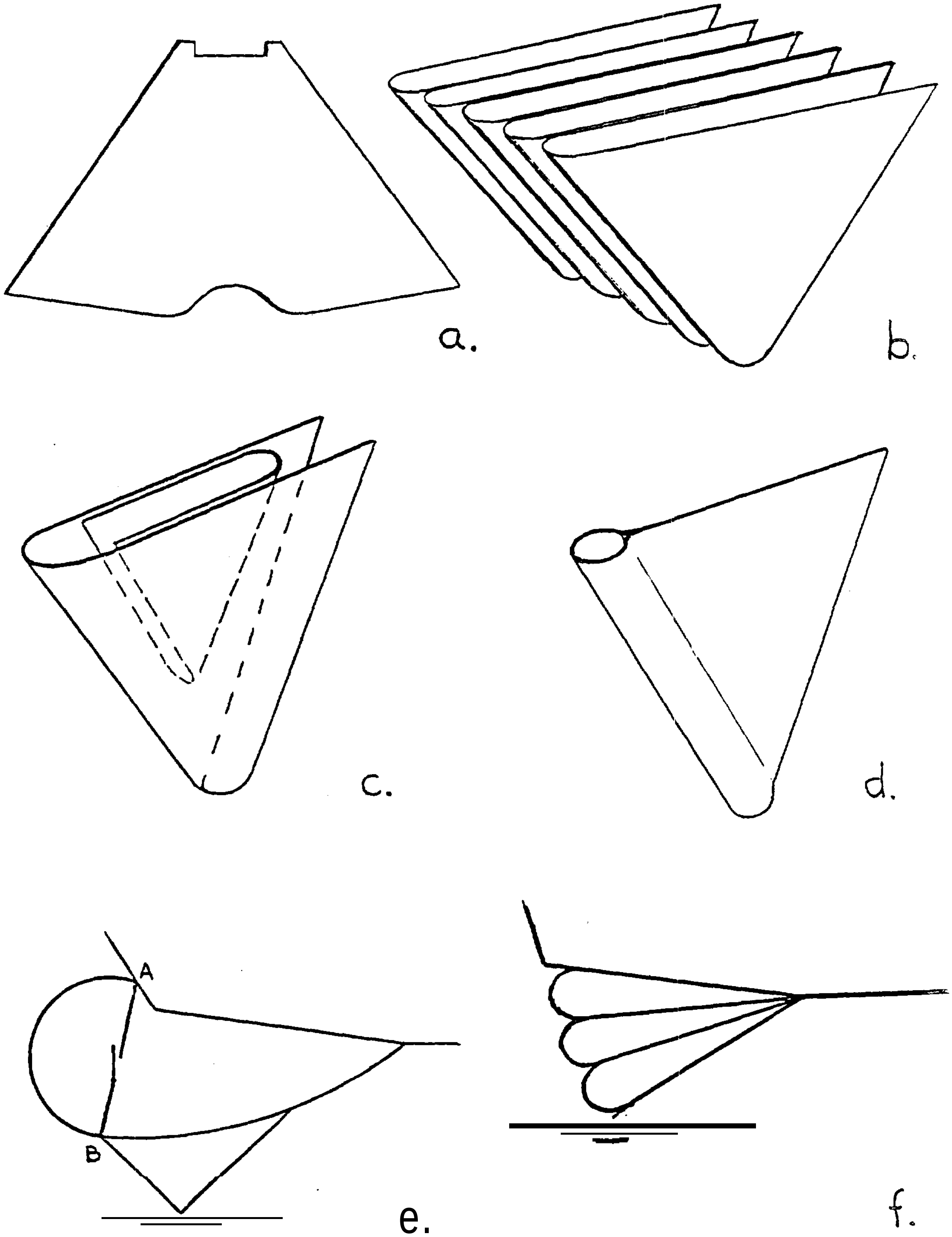


1/10TH. SCALE MANNED MODEL OF DEEP CUSHION SES. FIG.5a



[illegible] FIG.5b

FLEXIBLE SERL DETAILS. FIG. 6.



HIGH-SPEED MARINE CRAFT
KRISTIANSAND 4-6 MAY 1988

MARINTEK HIGH SPEED CRAFT RESEARCH PROGRAMME

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MARINTEK -

C O N T E N T

1. INTRODUCTION.

MARINTEK initiated and proposed to NTNF - the Royal Norwegian Council for Industrial and Scientific Research - in 1986/87 to establish a 3-5 year national research program on high speed crafts.

To be well-prepared for a development towards larger high speed unconventional crafts and to bring further the traditions of Norwegian shipping industry more extensive systematic research and development work is necessary. More fundamental and practical know-how within the various type of technology have to be provided. A 4-year research program will now be
3 established and started from 01.01.1989.

MARINTEK, however, started early in 1987 with several research projects financed by NTNF basic-fundings to meet the more urgent needs requested through actual commercial projects, both for national and international customers.

In this presentation the following items will be discussed:

- Background and objectives of MARINTEKs research activities responsibility and relations to the industry.

- MARINTEK activities and facilities, highlighting some actual projects.

- The necessity and importance of international cooperation with emphasis on MARINTEK participance in such programs and committees.

2. BACKGROUND AND OBJECTIVES.

The high speed craft industry is currently at a level where the manufacturers/yards are very careful with respect to design details and "new" ideas, which may be an indication that this is a rather "young" industry. However, many of the so-called "new" ideas have been introduced previously either within the shipping or aircraft industries. Until now a number of ideas has not been possible to realize because of shortcomings within certain technologies. To-day, however, the most important technologies seem to have simultaneously reached the same high levels of development, giving a unique possibility to realization of new concepts.

The basic technologies considered to be the most important for development of high speed crafts are:

hydrodynamics with special emphasis on ship motions, slamning, foils (wings) and air cushion/regulation technique.

development and application of new materials including structural design methods, testing and production technology.

machinery and propulsion systems.

operational procedures, training and navigation systems.

In general, the main responsibility of MARINTEK, and research institutes in general, is to be at the forefront of the development with respect to know-how, facilities and services so to provide knowledge assistance to the Norwegian industry in developing and improving their services and products. This also includes general activities with respect to public safety in co-operation with governmental authorities and classification society - DnV - to establish rules and regulations.

The principle objectives of MARINTEK research program/projects are to:

increase knowledge and physical/technological understanding appropriate to all fields.

develop design methods, guidelines, and testing facilities to be used by industry directly or through associated institutes.

To prepare the Norwegian industry, the number of qualified engineers should be increased at technical university levels of M.S. and Ph.D. and courses should be modified or new ones started within the various fields. Project and diploma work have already been initiated and Ph.D. studies will be connected to the research programs. The number of engineers at the various levels to be educated in the next years should be based on prognosis from the industry-builders, operators, governmental authorities and the Navy. At the operational level, special courses and training facilities should be established to satisfy the safety rules and regulations.

The close cooperation between MARINTEK and NTH makes it possible to combine research, education and industrial projects.

High priority should be given to international cooperation through projects and exchange of people to stimulate international experts working and teaching in Norway.

3. RESEARCH ACTIVITIES AND TEST FACILITIES.

3.1 Objectives of Tests and Procedures.

Through the years MARINTEK has tested quite a number of high-speed vessels, including resistance, propulsion, cavitation and seakeeping tests in head or following seas. Vehicles tested include: planing hulls, round bilge hulls, catamarans, hydrofoils, SWATH's and SES.

Compared to conventional ships the detailed design and optimization of vessel performance, propulsion systems as well as safety, seakeeping and handling characteristics of large high speed vessels require more extensive experimental investigations.

It is clear that developmental testing on a model will cost a great deal less, and also be less time consuming, than comparable tests on a full scale ship. This is particularly true in the case of a relatively complex craft such as an SES where the parameters that can be changed (e.g. skirt geometry, side wall shape, etc.) are not only large in number, but each can have a profound effect on performance, manoeuvrability, handling, etc. Thus, the first objective of a model test program is to achieve, in an economic manner, a craft design which will satisfy the basic performance requirements for successful operation.

With any high speed craft, the requirements of safety can affect the design much more than in the case of a conventional ship. Establishing the envelope of safe operation might well be dangerous and destructive if carried out on the full scale craft without the guidance provided by model test results. Model tests should be run with various simulated failures to ensure that the eventual full scale trials and practical operations should have an adequate degree of safety.

Experience with high speed crafts has shown that progress in design is invariably associated with problems or phenomena which have been either unexpected or even entirely new. These aspects of performance are impossible to foresee in the design procedure, simply because of their unexpected or unknown nature. However, model tests have been successfully employed in the past to bring these problems to light well before the design has been finalized and the occurrence of these problems on the ship has been avoided. Thus, a further objective of model testing is to attempt to reveal unexpected problems in the design and to find cost effective solutions well before the design is finalized.

The main objectives of the MARINTEK research projects in 1987-88 are therefore to:

- establish and improve testing techniques, equipment and procedures.

- develop design methods for both the early and detailed design stages including programs for operational studies with respect to transport economy and safety.

3.2 Performance and Cavitation Tests.

Reliable performance data are obtained on the basis of resistance and propulsion tests. Self propelled models are used, with stock propellers of diameter 0.10 - 0.25 m. Water jet units are also designed, manufactured and tested.

The first propulsion test with a high-speed model is performed with suitable stock propellers selected from an inventory of over 1100 propeller models or waterjet models. The final propeller design either provided by the customer or by MARINTEK can be tested in the cavitation tunnel in order to minimise cavitation, loss of efficiency, vibration and noise. The cavitation tunnel is also used to test propellers in inclined flow and to study cavitation on appendages, rudders, hydrofoils etc.

The following facilities are used for high speed vessel testing:

Towing Tank (280 m long, 10 m wide, maximum towing speed 10-12 m/s and maximum significant wave height H_S = 0.50 m).

Typcial scaling for towing and seakeeping tests are:

Max. speed	20 knots	30 knots	40 knots	50 knots
Scale	5	10	10	15
Max. H_S	2.5 m	5.0 m	5.0 m	7.5 m

Typically, models of length greater than 3.0 metres are used.

Cavitation tunnel. Diameter of test section 1.2 m

Max. water velocity	18 m/s
Max. propeller speed	3000 RPM

3.3 Seakeeping and Manoeuvring Tests.

For operational and seakeeping tests in random seas MARINTEK has developed advanced testing techniques for High-Speed Marine Vehicles (HMSV's) utilizing the large Ocean Basin (80 x 50 m) and free running models. In the Ocean Basin both unidirectional and multidirectional waves can be generated with maximum significant wave-heights of 0.50 m and 0.35 m respectively. The adjustable bottom allows both deep water and shallow water testing.

The capability to conduct tests in multidirectional waves is very important since experience has shown that more severe slamming occurs in multidirectional waves as compared to uni-directional waves.

Typical scaling of high speed vessels and the corresponding wave heights in the Ocean Basin related to maximum significant waveheights and maximum achievable speed are given in the table below:

Ship displacement	100 tons	200 tons	1000 tons
Model Scale	10	12.5	20
Max. speed seakeeping oblique waves	22 knots	25 knots	30 knots
Max. speed seakeeping head/following waves	43 knots	50 knots	60 knots
Max. sign. waveheight	5.0 m	7.5 m	10.0 m

The high speed seakeeping tests in the Ocean Basin with computer controlle free running models include measurement of speed, heading, pitch, roll, vertical and lateral accelerations, water on deck, waves, propeller RPM, shaft torque, water-jet momentum, rudder force, rudder angle, etc. The computer controlled test technique allow testing and tuning of active antiroll fins, auto-pilot systems and dynamic positioning systems.

MARINTEK's Ocean Laboratory is currently implementing a two year development programme on high-speed vessel testing techniques. This includes the development of:

- extremely stiff and light hulls manufactured with glass and carbon fibre reinforce epoxy.
- light-weight/high-power electrical and combustion motor systems.
- light-weight instrumentation and data transport (telemetry) systems for free-running models.
- methods for adding and analyzing short run statistics.
- test equipment for measuring slaning pressure., forces and moments.

The combination of Ocean Basin long-crested and short-crested wavemaking features and the light-weight model and instrumentation techniques enchances the opportunity for a "quantum leap" in the testing technology of high speed crafts. The development of testing techniques will be combined with further studies of ship/model correlation and development of computer programs for supporting the concept and detailed design of HSMV's.

3.4 Design Methods.

In addition and as complementary tools to the testing facilities, MARINTEK has also developed a set of preliminary design computer methods that include:

- Calm water resistance of SES-vessels.
- Resistance of semi-displacements monohulls.
- Optimum resistance of hydrofoil crafts incl. effects of cavitation and flaps.
- Estimation of hydrofoil lift, for both monohulls and twinhulls,
- Efficiency and main dimensions of waterjets and pumps.
- Dynamic stability of hydrofoil crafts.

These programs/methods are used at the preliminary design stage and to "design" an experimental set-up and procedure through systematic parameter variations before the detailed tests start.

3.5 Actual Project.

One of the most challanging and interesting of MARINTEK's present projects is the development and testing of a 1200 tdw. SES/multihull ocean-going vessel to be operated at a crusing speed of 35-40 kts. Several configurations like standard SES, catamaran, trimaran, combinations of air-cushion and foils will be tested within the next months. The project includes development of test rigs for fans, a rig for testing in waves allowing large yaw motions as well as small light-weight engine systems.

4. INTERNATIONAL COOPERATION.

Within the high speed craft research and development programnes international cooperation and relations are of great importance. In addition, in order to take part in establishing international standards, criteria and regulations. As the future navy vessels within the various countries may represent a kind of technology breakthrough for unconventional high speed crafts cooperation within the NATO-group is of special importance.

MARINTEK takes part in 3 different multinational cooperation projects or committees.

4.1 Nordic Research Project. Seakeeping Performance of Ships.

When planning the design of new vessels one always needs to have a rational basis for a techno-economic evaluation of alternative designs. This evaluation should include the vessel's operational performance where the seakeeping capability is one of the most important factors. To assess seakeeping capability of a vessel, however, one should have precise criteria and accurate methods for verification of seakeeping performance.

"Seakeeping Performance of Ships" is a joint Nordic research project coordinated by NORDFORSK, the Nordic Co-operative Organization for Applied Research. The aim of the project was to improve the knowledge of seakeeping capability of a vessel by developing criteria and methods for the verification of the seakeeping performance.

The project was carried out by the four Nordic ship research institutions:

Danish Maritime Institute
Technical Research Centre of Finland
Norwegian Marine Technology Research Institute AI'S
SSPA Maritime Consulting AB

In the Nordic countries fishing, shipping and offshore oil industries are important parts of the total industry. So far, there has been a lack of precisely defined criteria to judge seakeeping performance of ships. There has also been a lack of standard methods, testing techniques, measuring techniques and theoretical methods, which should be used to establish the motion characteristics of vessels. These facts make it difficult to decide on the best design in many cases. It is often difficult to compare results from different institutions, yards and consulting firms that design vessels.

A similar project and studies should be carried out for high speed vessels,

The final report - "Assessment of Ship Performance in a Seaway" - is now available and may be ordered from NOROFORSK or MARINTEK.

4.2 ITTC - High Speed Marine Vehicle Committee - HSMVC.

Members of the 18th ITTC HSMVC are:

Chairman:	Mr. Klaus R. Suhrbier, Vosper Thronycraft (UK) Ltd.
Secretary:	Mr. Kjell Holden, MARINTEK
	Dr. Jean Paul Bertrand, Bassin d'Essais des Carenes
	Dr. Dan Cieslowsky, David Taylor Research Center
	Mr. Bert Koops, Maritime Research Inst., Netherland
	Prof. Kirill Rozhdestvensky, Leningrad Shipbuilding Institute
	Dr. Olle Rutgersson, SSPA
	Dr. Hiraku Tanaka, Ship Research Institute

In addition MARINTEK /NTH participate in the Cavitation Committee and Seakeeping Committee of ITTC.

The hydrodynamic technology and understanding required for proper development of model test procedures for HSMV's are organized by the High-Speed Marine Vehicle Committee of the ITTC in order to be consistent with the increasing activities in high-speed vehicles and to properly serve the towing tank community. This will ensure the continued development of the specialized technology required for these craft, which, in turn, will be transferred from the towing tanks to the designers of HSMVs.

The committee period is 3 years and meets l-2 times per year.

The 18th HSMV-committee - 1987-89 - will concentrate on the following topics in addition to continous world-wide survey of crafts:

1. Appendage effects (drag, scaling).

2. Extrapolation methods/procedures
   - SES.
   - Aerodynamic effects (model, full scale)

3. Model/full scale powering correlation.

4. Analytical/experimental prediction procedures for semi-displacement and planing craft.

5. Hull/prop interaction problems.

6. Ship motion effects on propulsive performance.

7. Cavitation scaling - comments. Ventilation effects on propulsor performance.

8. Seakeeping
   - Linearity problems.

9. Manoeuvring
   - Model/full scale
   - Correlation of manoeuvring characteristics

10. Dynamic stability in waves with special emphasis on effects of different propulsors (i.e. broaching).

11. Dynamic problems - seakeeping/manoeuvring.

12. Structural loads.

13. Cooperative experiments with high speed craft model.

14. Trial Procedures.

Sea trials on high speed crafts should include the following observations/measures:

- Deviations from standard procedures
- Trim, observations of conditions, ventilation
- Transom separation. Wetted area
- Observation of spray
- Cushion venting
  No. of runs
  Sea - wind
- Water depth
- Displacement change (fuel, LCG)
- Ship motions
- Waterjets, unconventional propulsion

Standard test-programs and procedures should be developed.

15. Numerical Methods.

Survey of numerical methods approaches, procedures
Samples, comparisons
Listing of existing computer codes

<u>4.3 NATO - SGE (HYDRO)</u>

In 1982, the formation of the Special Group of Experts on Naval Hydromechanics and Related Problems (SGE(HYDRO)) was authorized by the Defence Research Group (DRG) with the participation of Canada, Denmark, France, the Netherlands, Norway and the United Kingdom. The Group first met in June

1983, by which time the membership had grown to include Germany, Spain and the United States of America. There has since been the occasional involvement of Greece and Italy.

MARINTEK represents Norway or the Royal Norwegian Navy in this group.

The Specical Group of Experts is concerned with a variety of aspects of naval hydromechanics testing and research.

The emphasis of the Group is based on naval issues such as the reduction of drag and noise of submarines; the behaviour of surface naval vessels in high sea states; the dynamic behaviour of towed systems, bodies and arrays; and the performance of underwater weapons and advanced naval vehicles. Furthermore, the Group is not limited to hydrodynamics in the narrow sense, but includes the inter-relation between the medium and moving bodies and the resulting elastic effects, stresses and vibration covered by the term "hydromechanics".

Four Research Study Groups (RSG) have been formed which are reporting to the SGE (Hydro):

- Full Scale Wave measurements
- Sea Loads, Slamming and Green Seas Impact
- Wake Measurements
- Cavitation Noise Scaling.

These studies will be finished within 1988. MARINTEK and Oceanor have in 1987 been involved in a large cooperative research program LEWEX - Labrador Extreme Waves Experiment.

A cooperative multinational trial was carried out in March 1987 off the coast of Newfoundland. The object of the trial was to evaluate available directional wave buoys and RADAR-systems provided by aircraft. Directional wave buoys were deployed from the Canadian research ship CFAV QUEST and the Netherlands research ship R.NL.S. Tydeman. Additionally, operational wave forecasts were produced by the United States Fleet Numerical Oceanography Center, Monterey. The Center provided directional wave spectral forecasts from the Global Spectral Ocean Wave Model.

There were six nations participating in the trial: Canada, France, the Netherlands, Norway, Spain and the United States. Each country participated in the trial by contributing scientists and instrumentation to teams aboard CFAV QUEST and R.NL.S. TYDEMAN.

The waters of the North Atlantic and North Pacific oceans provide some of the most severe wave conditions of the oceans of the world. As these areas are of considerable strategic importance to the navies of the NATO nations, there is considerable interest in the degree to which the wave environment hinders the operability of naval vessels, and in the aspects of ship response which are primarily responsible for reduced seakeeping performance. Slamming and green seas impact play a major role in limiting the speed of naval vessels of the frigate and destroyer size. Typically in seas approaching sea state 5 (6 m significant wave-height), the most modern ships of this size are limited to speeds of about 20 kts.

The importance of slamming and green seas loads on naval ship operations placed this topic high on the list of research projects that could be undertaken cooperatively. The aim of the research was to investigate the mechanisms responsible for slamming and green seas impact with a view to identifying means of improving ship performance in high sea states.

Both full scale trials and related model tests will provide a data base for bow flare slamming and green seas loading with which results from theoretical methods will be compared. Drop tests of two-dimensional sections were also be conducted. The will provde insight into the development of the girthwise slarrrning pressure distribution and the influence of scale effects.

New research study groups covering other areas will be started in 1988, where problems related to SES-technology may be given high priority and interest.

5. CONCLUDING REMARKS.

The following items have been adressed:

Within high speed craft research an open-minded and co-operative industrial attitude is of great importance to proceed the development towards successful design, production and operation of larger high speed crafts. Development and production of experimental prototype crafts would be of great advantage.

An important objective is to stimulate to education of engineers for the industry at all levels including operational and maintanance aspects. Builders, operators and authorities are at present in lack of skilled, professional engineers, within this field.

Development of rules and regulations should be emphasized to avoid "built-in" conservative attitudes which represents limitation and hinders successful results. The shipping industry is traditionally rather conservative compared to aircraft industry.

International cooperation is in general of great importance in order to minimize duplication of research and to start at the highest present level. Transfer of technology and exchange of experts are measures to be taken.

The combination of MARINTEKs expertise and testing facilities together with NTH, SINTEF and DnV represent in the international arena a unique and well qualified group which should be quite competitive. Together with builders and operators the necessary basis for a successful program exsists. However, close and faithful cooperation is necessary.

TOOLS FOR PREDICTIONS OF MOTIONS AND SEAKEEPING QUALITIES OF SES AND CATAMARANS

by

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HIGH SPEED MARINE CRAFT, 4-6 May 1988

Kristiansand

ABSTRACT

Important sea loads and motions of surface effect ships (SES) are reviewed. Limitations of numerical prediction models for SES and catamarans are pointed out. Model tests are shown to be useful except for prediction of heave resonance accelerations.

INTRODUCTION

The "High Speed Marine Vehicles" Committee of ITTC (International Towing Tank Conference) defined "high-speed marine vehicles" to mean:

a) vehicle types which sustain most of their weight, at design speed, by means other than hydrostatics,

b) unconventional displacement ships, and

c) semi-displacement ships which operate in a speed range Fn $\geq$ 0.5 (Fn = $U/\sqrt{Lg}$, $U=$ ship speed, L = ship length, g = acceleration of gravity).

Examples of high-speed marine vehicles are shown in Fig. 1.

We will particularly discuss prediction tools to evaluate seakeeping qualities of Surface Effects Ships (SES), catamarans and SWATH (Small-waterplane-area-twin-hull-ships). The Surface Effect Ship (see Fig. 2) is an air-cushion-supported vehicle where the air cushion is enclosed on the sides by rigid sidewalls and on the bow and stern by compliant seals of the bag and finger or planing type. When the SES is off cushion it behaves very much like a conventional catamaran.

Important variables in evaluating the seakeeping qualities are:

a) vertical motions and accelerations,

b) roll angles,

c) relative vertical motion and velocity between the vehicle and the waves,

d) deck wetness,

e) wave impact loads (slamming),

f) added resistance in waves, and

g) structural loads between the hulls.

In head sea, the vertical motion of any point P on the vessel can be obtained by a combination of the vertical motion (heave) of the center of gravity, pitch angle and the longitudinal distance between the center of gravity and the point P. The vessel's accelerations are important in assessing sea sickness and an individual's effectiveness in performing operations. Occurrence of deck wetness is strongly influenced by the relative vertical motion. The level of slamming is determined by both the relative motion and velocity as well as the local structural form where slamming is occurring. Large relative vertical velocities will cause large added resistance of the vessel.

Important parameters in assessing the level of vessel motions are the resonance periods, damping level and wave excitation level. Relatively large motions are likely to occur if the vessel is excited with oscillation periods in the vicinity of a resonance period. It is, therefore? of importance to know the resonance periods of heave, pitch and roll (for an unmoored vessel there are no (uncoupled) resonance periods in surge, sway and yaw.) However, if the damping is high or the excitation level is relatively low due to cancellation effects, it may be difficult to distinguish the response at resonance periods from the response at other periods. If no artificial damping is introduced, roll motions of a ship in a beam sea and vertical accelerations of a SES are examples where resonance oscillations can be clearly seen.

Obviously the motion of a vessel will be influenced by the vessel's form. We will not discuss this in detail here, but we will give some examples. Salvesen (1973) has made a comparative study of the seakeeping characteristics of SWATH, catamarans and monohulls. His conclusions are that the SWATH is superior to conventional monohulls and catamarans in moderately severe head seas. This can be clearly seen in Fig. 3. A simple way to explain why the catamaran has a higher vertical motion than the monohull is that the catamaran has lower damping in heave and pitch. This is caused by wave trapping between the two hulls. A reason why the SWATH ship has particularly low vertical motion is cancellation effects in the wave excitation loads. However, a SWATH ship may not be superior in all sea conditions. In following seas, the SWATH will pitch more if not equipped with foils. It should be noted that the examples presented above are for moderate and low vessel speed. A special seakeeping feature with-Surface Effect Ships is according to Butler (1985) that "SES can operate safely at higher speeds in higher sea states than equivalent length monohulls."

Heave and pitch are critical motion modes of a SES. Resonance heave motions may cause excessive vertical accelerations and make it unpleasant and limit operations on board a SES in cushion-borne condition, but nonresonant heave motion and pitch motion are also important for operational effectiveness and motion sickness indices. Large relative vertical motions between the SES and the waves can cause wave impact loads (slamming) and possible structural damage. This is particularly true in hullborne: condition.

The resonance period of the heave motion of the center of gravity of a cushion-borne SES is much lower than for monohull ships with comparable length. The natural period for the heave of the center of gravity of a cushion-borne SES can be approximated by

$$T_{N3} = 2\pi\left[\frac{h_b}{\gamma g(1 + p_a/p_o)}\right]^{\frac{1}{2}} \tag{1}$$

where γ = specific heat ratio of air (= 1.4)

g = gravitational constant

p_a = atmospheric pressure

p_0 = cushion pressure

h_b = cushion plenum height

This is also the natural period for the cushion and is physically due to compressibility of the air in the cushion. Let us exemplify this formula for the SES-200 vessel (see Table 1). We find that $T_{N3} = 0.5$ sec, which is low relative to wave periods of importance. But ocean waves can very well excite resonance oscillations in heave even if they do not have significant energy for periods around the natural heave period. An important reason is the frequency of encounter effect between the vessel and the waves. Let us explain this by an idealistic situation that can be created in a ship model basin. In one end of the tank there is a wavemaker that creates regular sinusoidal waves of period T_o. On the towing carriage we have mounted the vessel. The carriage is heading into the waves with a constant speed U. Let us concentrate on one point P on the vessel and consider the time T_e it takes for two successive wave crests to pass the point P. This will obviously be less than T_o. By analysis we can find that

$$T_e = T_o/(1 + \frac{2\pi}{T_o}\frac{U}{g}) \tag{2}$$

A real sea can be considered as the sum of many sinusoidal 'waves of different periods, amplitudes and directions. The energy will normally be concentrated around one period. If we call that period T_o and assume long crested waves, we can use equations (1) and (2) to find out qualitatively if different sea states can cause heave resonance oscillations of a SES in head sea. Resonance heave oscillations will occur if

$$T_{N3} = T_e \tag{3}$$

Equation (2) can be generalized to other wave headings by simply replacing U by $-U \cos \beta$. Here, β is the heading angle between the vessel and the wave propagation direction. For instance, head sea means $\beta = 180°$ and beam sea means $\beta = 90°$.

We can use the discussion to plot a diagram like the one shown in Fig. 4. The figure tells, for instance, that period $T_o = 2.5$ sec. associated with sea state I causes heave resonance when the SES-200 is heading with a speed 30 knots against the waves. If the vessel changes direction, for instance to bow sea (135° heading), a resonance heave oscillation will not occur for a speed of 30 knots and wave period $T_o = 2.5$ sec. It would occur if the vessel had a speed of -30 knots/ cos $135° = 42$ knots (see Fig. 4); but this is well above the design speed 28 knots of the SES-200 in calm water.

When resonance oscillations in heave occur it is not the heave motion itself that causes problems. Resonance heave motion may not be more than a couple of centimeters. It is the resulting heave accelerations that can cause problems. This is illustrated in Fig. 5, which presents results from full scale measurements with the SES-200 at full power in head sea (Adams and Beverly (1984)). When the Ride Control System (RCS) is off, the significant single amplitude of the vertical accelerations at the longitudinal position of center of gravity comes close to 0.4g, which is the limiting value set for intolerable conditions for the individual(s).

If we want to relate Fig. 5 to the discussion that followed Fig. 4 about sea states that cause resonance heave motions, we should have in mind that a sea state does not consist of one regular wave train of period T,. It is the sum of many regular wave components of different periods. This means that for sea state II there may be important wave energy for a wave component with period $T_o = 2.5$ sec. In addition we should note that the vessel does not only respond in heave at the resonance period. Further, it should be pointed out that the heave excitation is proportional to the significant wave height $H_{1/3}$ and that higher sea states obviously imply higher values of $H_{1/3}$ ($H_{1/3}$ means the mean height (crest to trough) of the one third highest waves).

From Fig. 5 we note that the Ride Control System (RCS) has been effective in damping resonance oscillations in heave. The RCS system used is shown schematically in Fig. 6. Vent valves and fan inlet guide vanes (IGV) are used to modulate the mass of the air in the cushion. For lower sea states, (IGV) are most effecient. We note from Fig. 5 that the RCS system is not working effectively outside resonance conditions. The reason is that the most important effect of the RCS system is to increase the heave damping and that damping has the most significant effect on the response in the vicinity of the resonance period.

In Fig. 7 are shown full-scale results for the vertical accelerations on the bridge. In particular for higher sea states, the acceleration on the bridge are higher than for the center of gravity of the SES. The reason is that pitch contributes to the vertical acceleration on the bridge. The resonance period of pitch is 3 sec. Compressibility effects of the cushion does not have a significant effect on the pitch motion.

All the results presented so far are for head sea and full power. Lower speed and other wave headings will generally mean lower vertical accelerations. This is illustrated in Fig. 8 for sea state III.

Adams and Beverly concluded that SES-200 could operate on cushion without excessive motion and wave impact loads up to sea state IV; i.e., a maximum significant wave height of 7 feet. The significant single amplitudes of pitch and roll motions were always less than 3° and 5°, which are motion limits set for helicopter operations. The significant values of the lateral accelerations were less than the 0.2g limit set for individuals. The speed reduction in different sea states is shown in Fig. 9.

Increasing the length of the SES would imply that the vessel could operate in higher seas at a higher speed. Adams and Beverly estimated that a 4000 ton scaled version of SES-200 could operate up to a 19 feet significant wave height. The projected speed varied from 46 knots in calm water to over 35 knots in sea state VI. The SES-200 may not be optimized from a seakeeping point of view. Other SES designs at comparable weights may results in higher limiting sea conditons for SES operations on-cushion.

The effect of waves on the roll stability of SES does not seem much addressed in the literature. For monohulls it is known that critical conditions may occur due to:

a) large roll motions in combination with deck wetness,

b) effect of breaking waves,

c) broaching or loss of directional stability in following sea, and

d) loss of static roll stability moment due to waves.

For SES it is not likely that situation a) will occur. Item b) i.e., breaking waves have been the cause of capsizing of many small vessels and cannot be outruled as a critical situation. Situation c) and d) could occur if the frequency of encounter between the waves and the vessel were small, *i.e.,* if the SES follows the waves. By generalizing equation (2), to following sea and consider and idealized case with regular following waves, we will find that $T_e = \infty$ (or the frequency of encounter between the SES and the waves is zero), if

$$T_o = \frac{2\pi U}{g} \tag{4}$$

For the SES-200 at speed 12" /sec, this means T_o = 7.7 sec. This is a representative wave period for sea state IV (see Fig. 4). If broaching will occur in this situation depends, for instance, on the position of the vessel relative to the wave crest, on the rudder size and the afterbody design of the SES hull. Consequent capsizing due to broaching depends, for instance, on the roll restoring moment. We have insufficient documentation to speculate if this can represent a critical situation.

NUMERICAL PREDICTION MODELS FOR SES

Taylor and Moran (1988) have surveyed features of four different numerical simulation models for hovercrafts (see Table 2). The four models were:

(1) Nonlinear time domain model developed by Oceanics, Inc. (Kaplan, et al (1985)),

(2) Nonlinear time domain model developed by David Taylor Research center (DTRC) and ORI, Inc. (Moran (1976)).

(3) Nonlinear time domain model developed by Textron Marine Systems (Moore and Neilan (1987)), and

(4) Frequency domain model developed by Maritime Dynamics, Inc. (1986).

Three of the models are nonlinear. An important nonlinear effect is air leakage through the seals. In the linear model this is approximated by equivalent linearization. An advantage of a linear frequency model relative to a nonlinear time domain solution is that statistical estimates of extreme values are more easily obtained.

One of the methods have only been applied to air cushion vehicles and is not directly applicable to SES. The reason is that hydrodynamic loads on th.e sidewalls and seals have to be accounted for.

Only one of the listed methods accounts for unsteady free surface deformation due to the vessel. Dependent on what the wave period, vessel length a:nd speed are, this may be an important effect. Moran (1975) has studied the problem experimentally by using a 1/3 scale model of the U. S. Navy's 45-foot XR-5 manned test craft '(length-beam-ratio $L/B \simeq 6.58$) in regular head sea waves. Results for the wave amplitude at different longitudinal positions x/L are presented in Fig. 10. Positive x is in the forward direction of the SES and $x = 0$ corresponds to midships. If the vessel had no effect on the incident waves, the wave transfer function η/ζ presented in the figure would be 1. This would be the case for very low and high frequency of encounter. Let us translate the results in Fig. 10 to the SES-200 ship. At heave resonance $\mu = \frac{2\pi}{T_e}\sqrt{L/g} = 28$. This is far outside the tested frequency range, but it is reasonable to assume $\eta/\zeta = 1$ i. e., that the SES has no effect on the incident wave system at the heave resonant period. But if we examined a wave period $T_o = 6.5$ sec., and a vessel speed of 25 knots, we will find that $\mu_e = 4.8$. According to Fig. 10 wave deformation is important. Both $T_o = 6.5$ sec. and $U = 25$ knots are representative values for sea state III (see Fig. 4 and 9); but also for higher sea states, wave deformation would be important. This means that a model that adequately describes air leakage due to large relative motion between the SES and the waves should incorporate unsteady free surface deformation due to the vessel as one of its features. The spatial pressure distribution in the cushion and the details of the dynamic air and water flow at the leakage areas are also likely to matter in this case.

Kaplan, et al **(1981)** seem to be the only ones that have presented extensive comparisons between theory and experiments in a refereed and easily available publication. **They** compared their computer program with model test data for six different designs. Details in terms of transfer functions are given for three surface effects ships. Actually, they are able to predict satisfactorily the motion transfer motion of the XR-5 model without accounting for free surface deformation (see Fig. **11, 12** and **13).** This may not be inconsistent with Fig. **10** and the accompanying discussion, the reason being that the heave and pitch motion is a consequence of an integrated pressure effect on the hull. This means that local effects as presented in Fig. 10 may not be that pronounced when integrated over the whole hull and combined with other physical factors. For one of the vessels used in their comparative **study,** the agreement between theory and experiment **was not completely satisfactory** (see Fig. **14).** Kaplan, et al **(1981),** have not provided any information how well **heave** acceleration resonance oscillations are predicted.

From a hydrodynamic point of view, the numerical models **presented in Table 2 seems to be** less advanced scientifically than methods used for engineering calculations of loads and motions of monohull ships.

The discussion of numerical prediction models so far has been relevant for a cushion-**borne SES. In** extreme weather conditions, a SES would be hull-borne and the hydrodynamic **analysis becomes** very much similar to that for **a catamaran. This will** be addressed in the following chapter.

NUMERICAL PREDICTION MODELS FOR CATAMARANS AND SWATH

The traditional way to calculate wave induced motions and loads on catamarans and SWATH is to use extensions of strip theory programs for monohulls. When it comes to monohulls, extensive comparisons between theory and experiments have been performed and one has good knowledge of the limitations of strip theory calculations. The same limitations should apply to catamarans and we will, therefore, present them in the following text.

Strip theory is a high frequency theory. That means it is more applicable in head and bow sea waves than in following and quartering seas for a ship at forward speed. The Seakeeping Committee of the 16th ITTC (International Towing Tank Conference) reports, for instance, substantial disagreement between calculated results and experimental investigations of vertical wave loads in following waves.

It should also be noted that strip theory is a low Froude number theory. It does not properly account for the interaction between the steady wave system and the oscillatory effects of ship motions. To the author's knowledge there is a lack of systematic investigations that show how good strip theory is at high Froude numbers; but care should be shown in applying the theory for $Fn = \frac{U}{\sqrt{Lg}} >\sim 0.4$. Here U is the ship speed, L is the ship length and g is the acceleration of gravity. One exception to this may be when the frequency of encounter between the ship and the waves is very high and the free surface deformation due to the hull does not matter.

Another limitation of strip theory is the assumption of linearity between response and incident wave amplitude. This means it is questionable to apply in high sea states with ship slamming and water on deck occurring.

Strip theory is also questionable to apply for ships with low length to beam ratios. The reason is that strip theory is a slender body theory. On the other hand, the Seakeeping Committee of the 18th ITTC concludes that strip theory appears to be remarkably effective for predicting motions of ships with length-to-beam ratios as low as 2.5.

Strip theory neglects all viscous effects. The most severe consequence of this is poor predictions of roll and torsional moment at roll resonance. In practical calculations, empirical viscous roll damping terms are added. For a catamaran, viscous effects will not matter much for roll predictions. But if hydrofoils are introduced between the hulls, viscous effects on the hydrofoil may influence the heave and pitch predictions. For a SWATH-ship viscous effects on the pontoons may influence the prediction of the vertical motions.

Even if strip theory has its limitations, it should be realized that strip theory in many cases gives good correlation with experiments and is extensively used in engineering applications. The comparative study reported in the 16th ITTC for the S-175 ship model shows, for instance, that the agreement between strip theory and experiments is good for

a) Pitch for all wave headings,

b) Surge for head and bow waves,

c) Sway and yaw for bow waves,

d) Vertical and longitudinal accelerations for all headings, except beam sea,

e) Lateral accelerations for all wave headings if autopilot effects are accounted for,

and

f) Vertical shearing forces and bending moments based on STFM for all headings except quartering and following seas.

STFM mentioned in item f. stands for the strip theory developed by Salvesen, Tuck and Faltinsen (1970). The other type of strip theory is often referred to as OSM and differs from STFM (or NSM) in the way the forward speed is accounted for.

An important hydrodynamic effect for catamarans is the interaction that occurs between the two hulls. This has, for instance, been studied in the zero speed case by Nordenstrom, Faltinsen and Pedersen (1971). For certain frequencies. the wave energy may be trapped between the two hulls and cause small damping in pitch and heave motion. Nordenstrom, Faltinsen and Pedersen showed poor agreement between theoretical and experimental values of wave bending moments between the two hulls. The correlation between theory and experiments was fair for pitch, vertical shear force, and pitch connecting moment between the two hulls.

The forward speed effect in catamaran seakeeping engineering predictions is accounted for in the same manner as for monohulls. The same limitations listed earlier for monohulls apply also for catamarans. In addition, it should be realized that the interactions between the two hulls that occur at forward speed becomes more complicated than for zero speed. This is exemplified in Fig. 15 where one numerical prediction method accounts for hydrodynamic interaction and another one neglects it all. It is the theory that neglects hydrodynamic interaction that agrees best with experiments. This means that the theory that accounts for hydrodynamic interaction is not correct. To my knowledge, there is not available a numerical method that properly accounts for the interaction between the two hulls when a catamaran has a non-zero forward speed. This implies that the prediction of, for instance, the wave amplitude between the two hulls may be in significant error. The consequence of this is, in general, poor prediction of wave impact loads and wave-induced dynamic loads between the two hulls. However, it should be noted that Hadler et al (1974) were able to show good agreement between theory and experiment for the Hayes catamaran in head waves at 10 knots for heave, pitch and relative motion by neglecting all hydrodynamic interaction and assuming the structure had no influence on the incident waves. The rationale for doing that is questionable and more extensive comparisons betwen theory and experiments are necessary.

MODEL TESTS OF SURFACE EFFECTS SHIPS (SES)

Model tests are a common way to evaluate the seakeeping qualities of a SES. In Fig. 16 and 17 are presented results from a comparative study of model and full scale values of heave and pitch motion of SES 100B. The agreement is generally good, except for low frequency pitch motion. The reason for the disagreement is not likely to be scaling problems within the model tests.' Without having detailed information on the model tests and the full scale experiments, it is difficult to elaborate further on the disagreement. A more general comment would be that it is easier to do controlled tests in model scale then in full scale. For that reason, one should put more faith in the model test results.

Due to difficulties in scaling compressiblity effects from model tests to full scale, there are, in special cases, difficulties in interpreting model test data for SES. This can be explained in a simple way by Eq. (1). Since atmospheric pressure p_a is not normally scaled in model tests, and since $p_a / p_o \gg 1$, Eq. (1) implies that

$$\frac{(T_{N3})_{mod}}{T_{N3}} = \frac{L_{mod}}{L} \tag{5}$$

The index "mod" in (5) means model scale and L is the length of the vessel. Let us take SES-200 as an example. We have earlier said that T_{N3} is 0.5 sec in full scale. If we use a model scale length that is 1/25 of full scale, Eq.(5) says that $(T_{N3})_{mod} = 0.02$ sec. When model tests are finished and the results for all wave periods are going to be translated to full scale values, we would use Froude scaling. This means that th.e ratio between full scale time T and the model time T_{mod} is

$$\frac{T}{T_{mod}} = \frac{\sqrt{L}}{\sqrt{L_{mod}}} \tag{6}$$

Equation (6) implies that we translate the full scale resonance period in heave to be 0.1 sec, and not 0.5 sec, which it should be. Since heave resonance accelerations are significant for a SES, this obviously causes errors in predicting acceleration levels in certain sea states. However, this scaling problem may be considered an isolated problem associated with heave resonance acceleration. One may put more faith in numerical results than model test results in these special cases. But, it should be realized that numerical methods are not that advanced that they can properly model all physical effects that matter for SES. For instance, we have found no documentation on how well numerical methods predict heave resonance acceleration in full-scale. Examples where model tests are particularly needed and superior to numerical method is evaluation of slamming pressure on the wet deck, structural loads between the two hulls, and effect of waves on the stability of a SES.

CONCLUSIONS

Tools to predict seakeeping qualities of Surface Effect Ships (‘SES) and catamarans are discussed. Even if numerical methods in several cases show good correlation with model tests for vertical motion predictions, it should be realized that numerical methods are not advanced enough to properiy model all physical effects that matter for a SES or a catamaran. Examples where numerical methods may show bad correlation with model tests and full scale experimental values are wave impact loads (slamming) and structural loads between the hulls. Model tests are a useful way to evaluate seakeeping qualities of a SES and a catamaran. One exception is prediction of heave resonance accelerations of a SES.

ACKNOWLEDGEMENT

Discussions with Lt. P. W. Taylor and Prof. P. Kaplan are appreciated.

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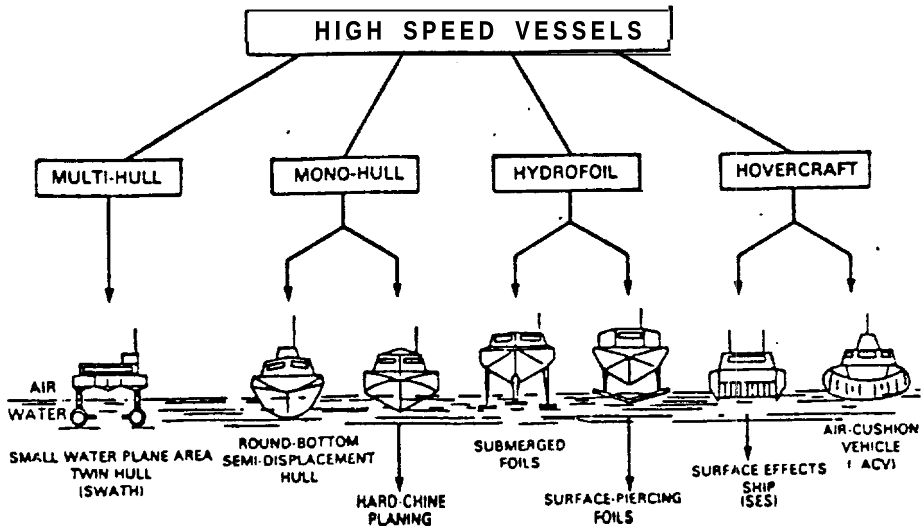


Fig. 1. Example on **High Speed Vessels** (ITTC(1.981))

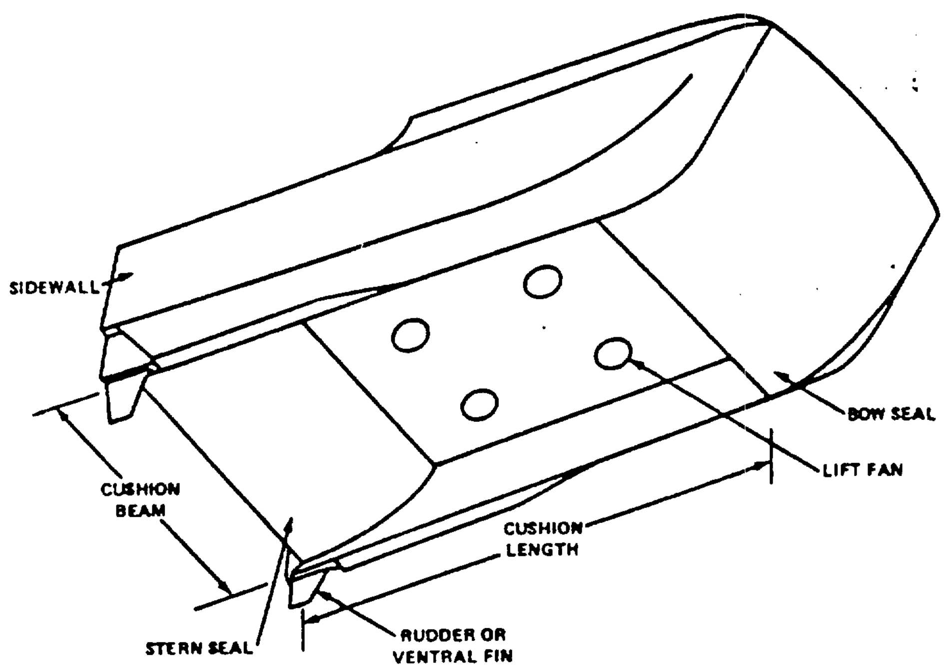


Fig. 2. Surface Effect Ship (SES) (ITTC(1981))

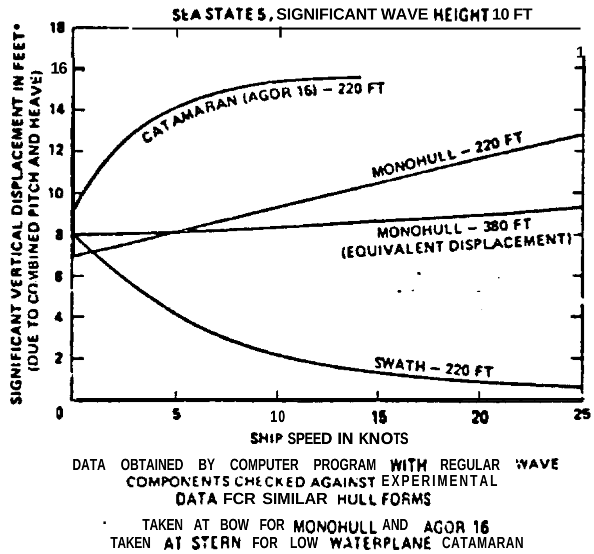


Fig. 3. Curves of significant vertical displaement vs. ship speed comparing a small-waterplane-area-twin-hull ship, a monohull, and a catamaran (AGOR 16)(Salvesen (1973))

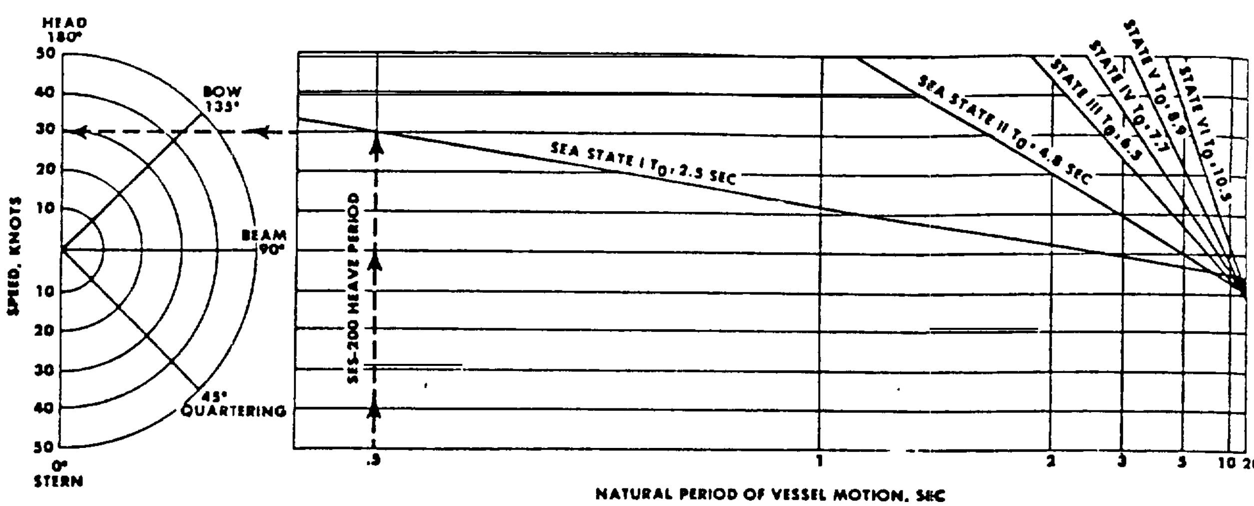


Fig. 4. SES-200 Heave Period (Butler (1985))

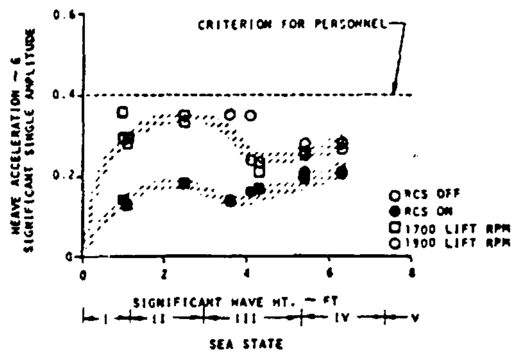


Fig. 5. Vertical Accelerations at Full Power in Head Seas at the Center of Gravity (Adams and Beverly (1984))

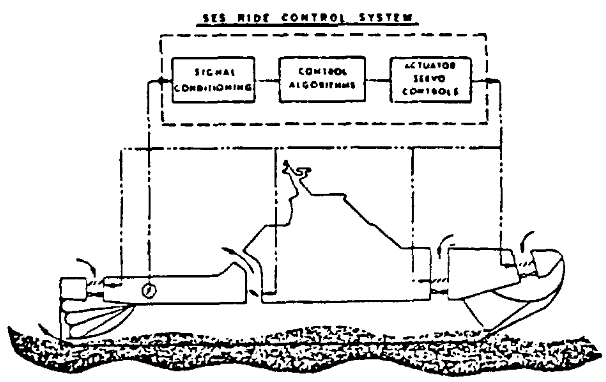


Fig. 6. SES Ride Control System (Butler(1985))

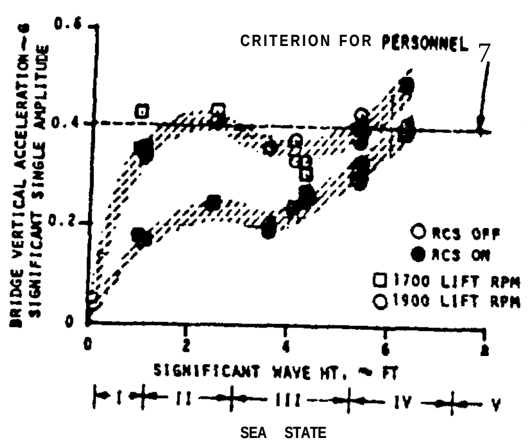


Fig. 7. Heave Acceleration at Full Power in Head Seas at the Bridge (Adams and Beverly (1984))

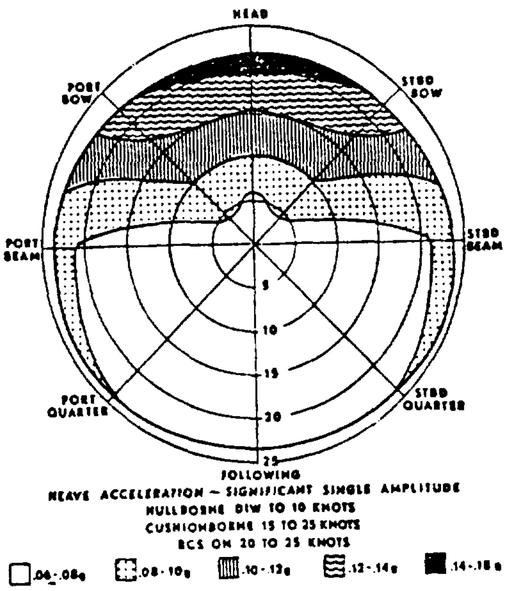


Fig. 8. Sea State III Heave Acceleration Speed Polar Diagram (Adams and Beverly (1984))

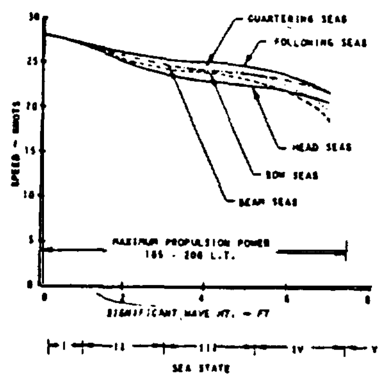


Fig. 9. SES-200 Speed versus **Sea State (Adams and Beverly** (1984))

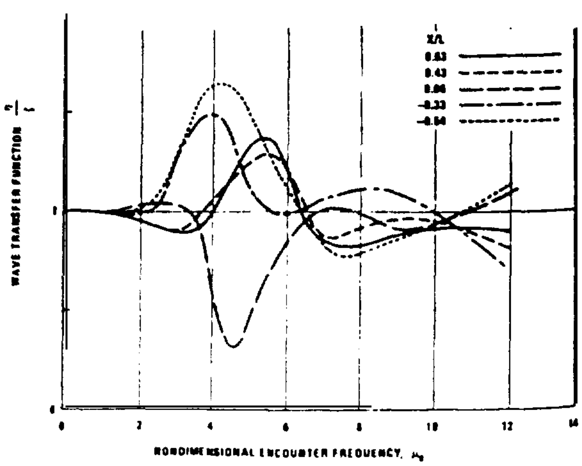


Fig. 10. **Wave** amplitude transfer **function at several locations in** the cushion of the model for regular waves (Moran, et al (1977))

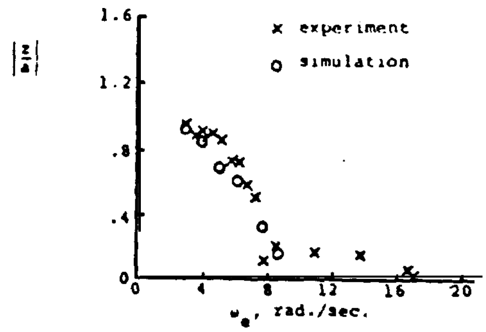


Fig. **11.** Heave response in regular waves (model tests and simulation) L/B = 6.54, Fn = 0.72 (Bentson and Kaplan (1979))

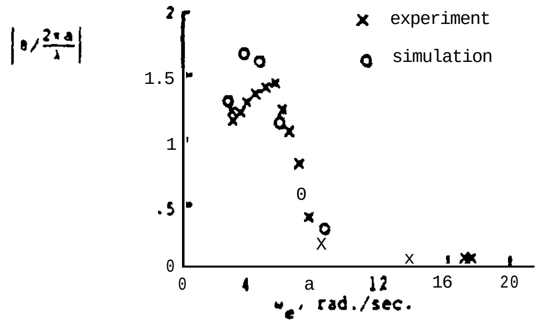


Fig. 12. Pitch response in regular waves (model tests and simulation) L/B = 6.54, Fn = 0.72 (Bentson and Kaplan (1979))

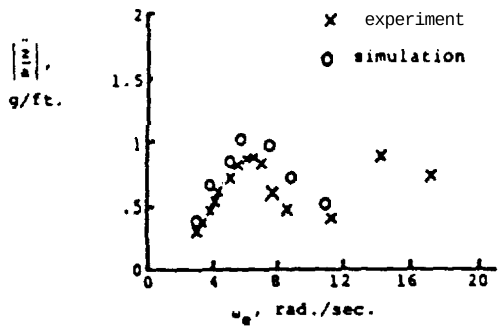


Fig. **13.** Heave acceleration response in regular waves (model tests and simulation) L/B = **6.54,** Fn = 0.72 (Bentson and Kaplan (1979))

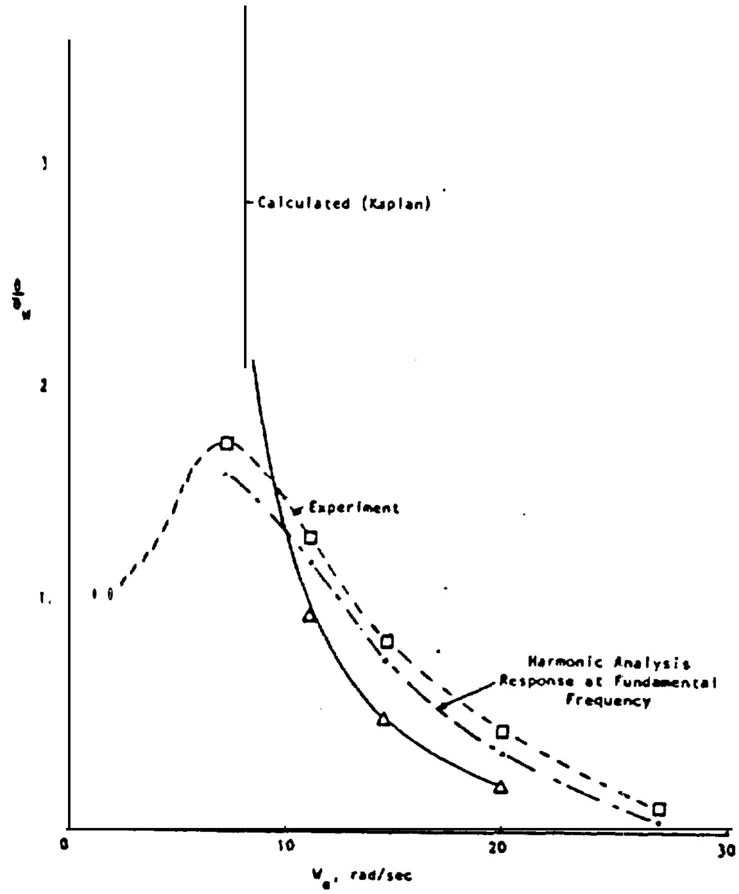


Fig. 14. PLS pitch responsein regular waves, U=28.3 fps (Kaplan, et al (1981))

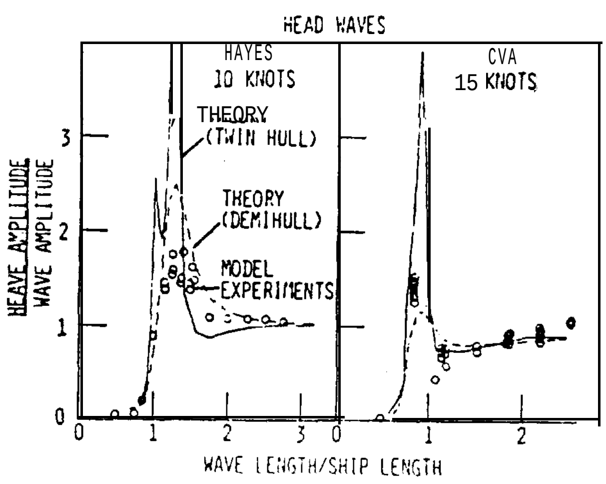


Fig. 15. Comparison of twin and demihull theory with model experiments (Hadler, et al (1974))

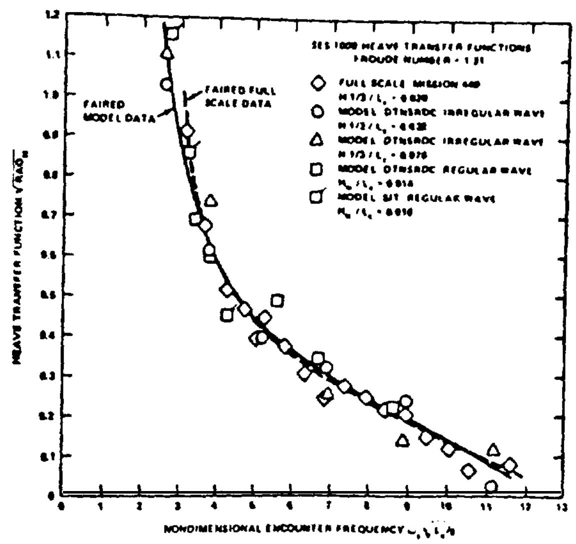


Fig. 16. **SES-100B** model and full-scale heave response comparison at a **Froude** number of 1.31 **(ITTC (1987))**

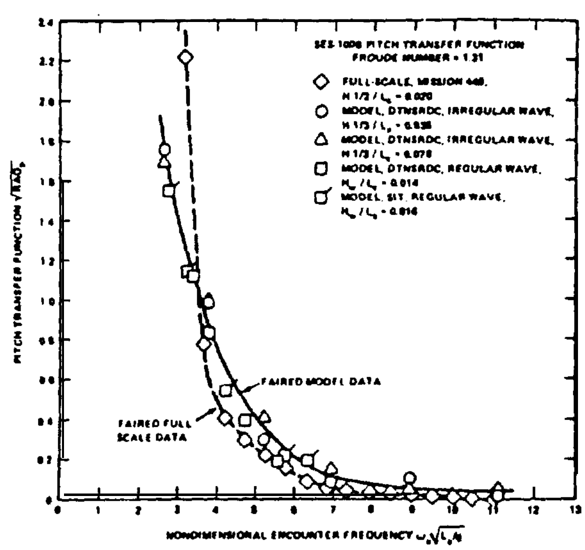


Fig. 17. **SES=100B model and full-scale pitch response comparison at a Froude** number-**of** 1.31 **(ITTC (1987))**

WEIGHTS:	
Displacement(100%Fuel&Supplies)(LT)	205
Fuel (LT)	59.6
Light Ship (LT)	128
DIMENSIONS:	
Length Overall (L_{OA})(ft)	159.1
Beam Overall (B_{OA})(ft)	39.0
Depth Molded (ft)	15.2
Mast Height Above Keel (ft)	41.5
Navigation Drafts (incl. rudder(ft)):	
Hullborne FLD	9.3
Cushionborne FLD	5.5
Wet Deck Height (ft):	
Bow	1.5
Stern	5.0
Effective Cushion Length (L_c)(ft)	133.3
Effective Cushion Beam (B_c)(ft)	11.5
Cushion Length/Beam Ratio	4.25
Nominal Cushion Pressure @ FLD (psf)	52

Table 1 Main particulars for the SES-200 ship (Adams and Beverly (1984))

PROGRAM	(1) OCEANICS	(2) DTRC/ORI	(3) TEXTRON MARINE	(4) MARITIME DYNAMICS
TYPE	Nonlinear Time Domain	Nonlinear Time Domain	Nonlinear Time Domain	Freq. Domain Linearized About Mean Op. Cond.
SHIP TYPE	SES	ACV: JEFF (A) JEFF (B)	SES: 100 B 2K SES ACV: LCAC JEFF (D)	SES: SES 200
Rigid Body Degrees of Freedom	6	5 (No Yaw)	6	5 (No Surge)
Free Surface Deflection	No	Yes (Relaxation Equation)	Yes - Steady State Deflection (Empirical)	No
Seal Flow Shutoff	Yes	Yes	Yes	Yes (Statistically Accounted For)
Quasi-Steady Air Flow thru Seals	Yes	Yes	Yes	Yes
Cushion Compressibility (Adiabatic)	Yes	Yes	Yes	Yes
Spatial Press. Distribution in Cushion	No	No	NO	No
Quasi-Steady Fan Curve Used	Yes	Yes	Yes	Yes
Seal Dynamics Modelled	Yes	No	No - Empirical Seal Force Data	No
Accommodates Ride Control Simulation	Yes	No	Yes	Yes

Table 2 Comparison of four hovercraft simulation programs (Taylor and Moran (1988))

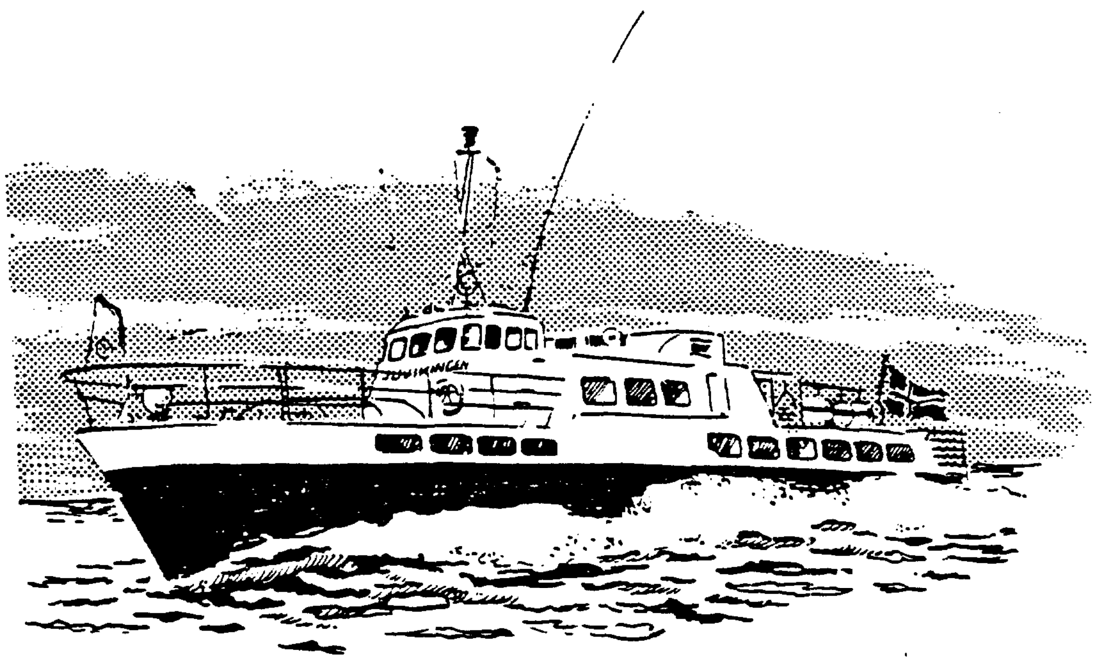
SINTEF - The Foundation for Scientific and Industrial Research
at the NORWEGIAN INSTITUTE OF TECHNOLOGY

Division Director Birger Elvestad, MSc. ENG

MAY 1988

HIGH SPEED COMMERCIAL CRAFT IN COSTAL WATERS

- ECONOMICAL ASPECTS



Norwegian Society of Chartered Engineers Conference on High-speed
Marine Craft - Technology update and market potentials
4 - 6 May 1988, Kristiansand, Norway

1. INTRODUCTION

This lecture is concerned with the economic aspects of operating scheduled services with fast passenger vessels in coastal waters. The economic issues that will be dealt with are confined to the transport-related side of such operations based on material from services in Norwegian coastal waters and fjords. I will also discuss the cost of the scheduled service and carefully evaluate the potential earnings and costs.

The traffic data covers the changed traffic volumes following the qualitative changes in service offered by faster vessels, the changes in travel time, frequency and the way in which such craft are used by passengers.

The cost of the scheduled service is related to the actual and potential use of fast passenger vessels by shipping companies in Norway.

2. SUMMARY OF DEVELOPMENTS

Fast passenger vessels first came into service in the 1960s as a supplement to conventional coastal vessels. The new vessels were mainly restricted to summer services. In the 1970s, new types of fast vessels were introduced: Catamarans, hydrofoils and vessels with naval gunboat hulls. These vessels permitted a fast passenger service throughout the year, and led to an overall improvement in local passenger services. As the operational speed of fast vessels was between 24 to 30 knots, about 80% of all routes were able to increase the sailing frequency. The remaining routes had unaltered frequency levels. About half of the shipping companies were able to replace one or more of their conventional ships by fast passenger vessels.

This first revolution in coastal services led to considerable improvements in traffic volume. Passenger volumes increased by between 40 to 80% in most areas and the majority of these passengers were newcomers to coastal passenger services.

The greatest impact was felt in rural communities of less than 300 inhabitants without roads. There was an annual passenger increase per inhabitant of 70%. This meant an annual average of between 4 to 5 trips per inhabitant.

In more populous rural areas and communities with upto 8 000 inhabitants served by fast passenger vessels, there was a per capita increase of between 0.2 to 3 trips in to the regional centre annually.

The changes in the scheduled service had little influence on the connections between rural areas and their respective community centres. On the other hand, the rural and community centres were provided with a completely new service in to the regional centres. The introduction of fast passenger vessels, however, made it less easy to send goods by coastal shipping.

The new vessels improved the standard of the scheduled service. Travel times were cut by 50 to 60% in most cases. The schedule was altered to allow day trips so that passengers from rural areas were not obliged to stay overnight when visiting their regional centre.

The most frequent users of the fast passenger vessels live in rural areas or community centres. For such groups "necessity" was the most common reason for taking these vessels. In contrast, passengers who lived in the regional centres or those from outside the route network, generally took the service for the purpose of "recreation".

For the majority of passengers, the regional centre was either the start or destination of trips by fast passenger vessels.

Most passengers were satisfied with the departure and arrival times for these vessels. Their high speed made it easier than before to find a suitable schedule.

It seems to be conclusive that the traffic increase is due to the improved standard and the new travel possibilities. It has not been possible to find a traffic increase model which could be applied for the calculation of traffic for planned new services.

We are on the verge of a new generation of vessels. Vessels such as hovercraft, combined hovercraft/catamaran and ones with modified hulls with efficient propulsion units will bring the operational speeds up to between 35 and 50 knots.

This raises a number of questions:

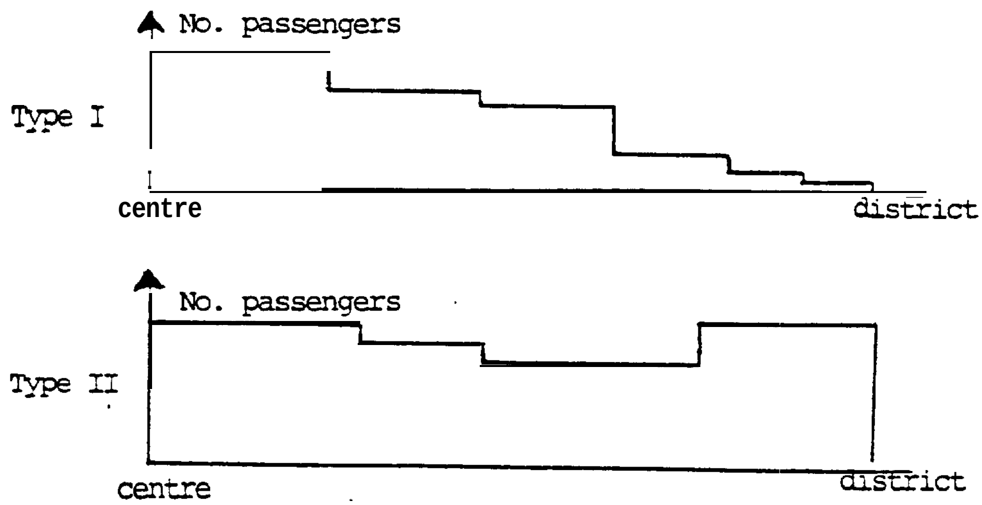
Which traffic market these vessels suit?

What is the potential for the generation of new traffic and the transfer of traffic from other modes of transport?

How will the financial side of shipping operations be altered if such vessels are brought into service?

These matters are central economic issues related to scheduled passenger services, they will now be considered more carefully taking a concrete case from coastal traffic in mid-Norway.

3. THE INCOME CHARACTERISTICS OF THE ROUTE STRUCTURES



In route structure I, travel is primarily from the district to the centre, though there is some traffic in both directions. The frequency is not as important as allowing passengers a reasonable amount of time in the regional centre. The generation of trips is governed by the number of inhabitants, ranging from 0.5 to 2.0 round trips per inhabitant annually.

The lowest generation of trips is found in local centres with >8 000 inhabitants situated far from regional centres, the highest figures were in small stops with <300 inhabitants which were close to the regional centre. En route traffic between stops was rare. Here the growing competition from private cars was apparent. Car ownership is growing in rural areas and the standard of roads and connections are improving. Since the fjords are not as advantageous as the "straight line" of the road, car ownership is a competitive transport alternative in such rural areas.

We know that in route structure I the change from conventional vessels to the 1st generation fast vessels led to an increase in passengers at some stops of between 20 to 90%. In addition, most travel times were almost halved. Many people were given the possibility of using coastal transport on a day-return basis, making the overnight stays in the regional centre unnecessary. A further reduction in travel time is likely with the new generation of fast passenger vessels. These vessels will be capable of speeds of 35 to 50 knots, and will reduce the time people have to be absent to take a round trip from the district to the regional centre. Nevertheless, as this reduction will not be as revolutionary as the first one, the share of new traffic will be much smaller. Competition with the private car will also be more noticeable. Though the increased speed could make it possible to offer higher frequency, it is doubtful if there is enough potential traffic to make this viable, unless there is a market for round trips from the regional centre out to the districts.

Road transport by private car has cut back the time advantage of fast passenger vessels. Higher speeds in vessel routes, type I, is a necessity if costal transport is to hold its own, or possibly win back the traffic from the roads lost at the beginning of the 1980s. A minor share of traffic transfer is feasible.

The districts are suffering from depopulation. The reduction in the number of inhabitants is clearest in the trip generation figures. For those living in rural areas with few inhabitants, an average trip generation of 10 - 15 trips per inhabitant, annually is common with the 1st generation fast passenger vessel. Calculations concerning the effect of the 2nd generation fast passenger vessel on the trip generation figures show a one-time increase of 10%. If depopulation continues for 3 - 4 years with just a few per cent, this trip generation increase could soon disappear..

Other changes could be relevant for the trip generation factors: purchasing habits, service consultations, the work market, structure of schooling and trips for leisure/visits. Necessary trips include trips for treatment, work and education, visits-to public offices and business travel. These constitute about half of the market (40 - 45%) for this type of route. Leisure and buying trips or a combination of these, make up the rest of the trip generation. This last category of trip is the one that is most sensitive to reductions in buying power. Nevertheless, this is also the group with the greatest growth potential if the service is attractive and well-directed for periods when the economy is sound.

In route structure II there are two fairly equal regional centres (such as Bergen and Stavanger) which form the basis of the route. The frequency and the number of round trips daily are important here. The speed and the price are also significant compared with other means of transport. Competition is wide, with air, coach and private cars as the most usual alternatives. The generation of trips is mainly regulated by the contact requirements of business.

There are few routes of this type in the material for Norway. The flagship route Stavanger - Bergen has a long tradition. Otherwise, it is the traditional express coastal steamer service which has served these routes until now. New routes are planned. One of these is between Trondheim and the main towns in Møre county: Hristiansund - Molde - Alesund. Though there will be a small amount of traffic en route to and from the coastal areas, this will largely a centre-to-centre route.

The calculations which have been done are based on a fast passenger vessel service with 2nd generation vessels (hovercraft/ catamaran) which at sea only compete with the conventional express coastal steamer. The new type of vessel will have to compete with air, car and coach transport.

As Table 1 indicates, fast passenger vessels have a clear competitive edge over conventional shipping, car and coach transport when it comes to mean travel time. The new type of vessel is also competitive with air over the shortest distances because of the travel time on the ground. Given the present situation, coaches and air transport operate with a higher frequency of departures, and there is no limit to the frequency of private cars.

Table 1 Comparison of different means of transport

	New vessel	Air	Exp. steamer	Coach	Car
A-M	45 min NOK 190 2x day	1h 40min NOK 270 2x day	3h 30min NOK 96 lx day	2h 15min NOK 69 6x day	1h 45min NOK 134 *
M-K	1h NOK 190 2x day	1h 35min NOK 270 6x day	4h 30min NOR 132 lx day	1h 50min NOK 78 7x day	1h 30min NOK 124 *
A-K	2h NOK 220 2x day	2h 50min NOK 310 2x day	8h NOK i73 lx day	4h 40min NOK 173 3x day	3h 10min NOK 258 *
K-T	2h NOK 290 2x day	2h 20min NOK 410 6x day	6h 45min NOK 250 lx day	5h NOK 173 4x day	3h 40min NOK 303 *
M-T	3h 15min NOK 340 2x day	2h 40min NOK 485 7x day	10h 30min NOK 360 lx day	5h NOR 200 5 x day	4h 5min NOK 350 *
A-T	4h 15min NOK 410 2x day	2h 20min NOK 590 7x day	13h NOK 389 lx day	7h 20min NOK 263 5x day	5h 50min NOR 485 *

* private cost of car travel from a Norwegian driving manual (Kjørekostnadshåndboken) per 01.01.1985
+ 7% inflation per annum
+ ferry tickets.

According to calculations the price level of the new vessels should be cheaper than the private cost of car transport. Coach transport should be cheaper and air transport more expensive than the new vessels. This price is mainly determined by the type of passenger category and the competitive advantages over the various means of transport mentioned above. Centre-to-centre route structures will produce a higher share of necessity trips than centre to district (SO-70% opposed to 40-45%).

We find more passengers on business travel and: fewer on buying trips. The chief competitors to the new type of vessels for the business traveller, will be the plane for longer distances and the private car for the shortest ones. For leisure travel, the car is the strongest competitor, and will often be unbeatable if 3-4 travel together.

There is no particularly large group of potential travellers in a market as described here. Consequently, the share of new traffic is restricted.

Transferring traffic from other means of transport has its limitations. The existing prices of various forms of transport are set in a restrictive manner and the prices, are regulated according to existing concessions. There are numerous local variations, however in the present example, the following would be the maximal figures which could advisably be expected.

20 % transfer from air transport
20 % transfer from private car
30 % transfer from coach transport
50 % transfer from the express coastal steamer service

This traffic would represent an income of NOK 10.5 million. This would only cover part of the operating costs of the new service.

Conclusions concerning the income potential

The basis for traffic between the centre and the district is restricted. Thus there will only be a small increase in traffic with 2nd generation vessels which offer higher speeds or more frequent departures.

The competitive situation found in the centre-to-centre routes indicate that the higher speed of the 2nd generation vessels mean that there is a chance of entering the passenger transport market with a reasonable share of the passenger volumes. Nevertheless the means of price determination for the scheduled passenger market may make it difficult to enter this market.

The limited room for manoeuvring regarding earnings may make the cost the decisive factor which will determine the feasibility of a 2nd generation of fast passenger vessels.

4. COST CHARACTERISTICS OF THE ROUTE STRUCTURES

The cost of shipping operations are normally divided into 4 main categories:

1) Vessel costs, these are normally variable and include the cost of fuel, crew, maintenance and insurance

2) Route costs these include commissions, harbour fees, advertising/markerting, operating costs of terminals and local waiting halls, and operating costs for the infrastructure/crew/lodgings, bunker and maintenance facilities.

3) Capital costs for vessels, depreciation and interest

4) Share of common expenses for administration and training, profit.

The respective shares of the costs are illustrated by some examples:

	Route type 1 Districts-Trondheim1)	Route type 2 Ålesund-Trondheim2) Sandnessjeen Tr.heim
Vessel costs	38-45%	55-58%
Route costs	15-22%	8-10%
Capital costs	30-32%	30-32%
Common expenses	8-10%	2-5%
Income/cost ratio	0.4.-0.5	0.3-0.9

1) Hovercraft
2) Ses-catamaran

It is not expected that the vessel and capital costs will be dominant for these two route structures.

The vessel cost share of the total expenses will be less in the district routes (type 1) than in the centre-to-centre routes (type 2). The vessels will have fewer operative hours a year in type 1 routes because they have no combined function. Thus the vessels will be berthed in the centre during the day and at the district departure point at night.

The route costs will be higher in the district routes than in the centre-to-centre routes. This is because of the infrastructure facilities. In the districts, the vessels are usually the only ones to use the local ports, while in larger centres the terminals/quays/waiting halls can also be used by others.

The capital costs amount to about a third for both types of routes. National, regional differences are found concerning financial conditions and funding arrangements.

The vessel costs include the fuel, crew and maintenance as the main categories. The share of the vessel costs for these items varies with the type of vessel. For the 2nd generation of fast passenger vessel there is sharp competition between hovercraft and and hovercraft-catamaran. Here, a trade-off has to be made between fuel consumption and the acquisition/maintenance costs of the respective vessels.

5. FUTURE FAST PASSENGER VESSELS IN SCHEDULED SERVICE

A number of the lectures during this conference have touched upon the technological perspectives for fast passenger vessels.

I will conclude with some of the issues which have to be considered before we can discuss whether such vessels have a future role in scheduled service. I have based this on the income and cost characteristics which have been presented for such vessels. I feel that I must stress that there are other markets and application aspects which are more decisive for the development of such vessels.

The price level and the will to pay for the services of such vessels in scheduled service leave little room for manoeuvre on the income side. On the other hand, new markets could be opened up, but this will not be possible without increased operating costs. Some examples are:

> Charter traffic in different forms can be (combined with scheduled services. Sporadic sightseeing tours in the summer is an irrational concept. Tourism must be integrated into a package which includes transport by such vessels. This type of transport could also be promoted in connection with education (schools, courses/conferences). Such traffic should preferably start in the main centres.
>
> Total transport solutions are becoming increasingly important. The route network should be structured for nodal traffic. Since travellers frequently need additional transport to and from the terminal, a reasonably-priced feeder service could be established using taxis for instance. Through-fares for different means of transport such as sea-air, sea-coach, sea-rail based on regional or national price zones would also be of interest to travellers.

It is on the cost side that the greatest possibilities and challenges are to be found. Perhaps this is my technical way of thinking coming to the fore. Fuel costs must be reduced considerably in relation to the first generation of fast passenger vessels. This must be done without increasing the maintenance costs, rather the contrary, maintenance costs per sea mile must be reduced and be brought into line with the prices that are common in the car industry. Innovations in integrated hull designs, aggregates and propulsion units could half overall costs compared with today's. The challenge involves finding hull modifications, changing the use of materials and developing more efficient aggregates.

Manning costs are another main element. Increased speed will bring increased productivity, but this is not enough. At present such passenger vessels require a crew of 4-5, efforts should be made to make such vessels operative with a crew of 2. Once maintenance requirements are reduced this will allow manning levels to be minimized. If we incorporate more automatic control, automatic ticket systems, accounting, cafe and catering etc., fast passenger vessels could still meet the classification requirements with reduced crews.

Efforts to reduce operating costs could lead to increased capital costs for such vessels. The relationship between operating costs excluding crew and capital costs for scheduled services is biased when compared with competing forms of transport. Scheduled fast passenger vessel services can accept higher purchase prices providing the fuel and maintenance costs at least compensate for this increase.

There is a huge potential for innovation within the organization and administration of scheduled passenger shipping. These are still two costly elements which will not be dealt with here since they are not of decisive importance for the future of fast passenger vessels in scheduled service.

I hope that my argument has indicated clearly enough that it is the marine technological developments that represent the greatest challenge, and that this may represent the motivation that is required for further developing fast passenger vessels as a viable means of transport.

High-Speed Marine Craft

Technology update and market potentials

TOTAL TRANSPORTATION CONCEPT SEA/LAND/AIR

- HOW WILL HIGH-SPEED MARINE CRAFT COMPETE?

by

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IKO LOGISTIKK AS
Oslo. 13. april 1988

Tottrans csp/JEW-kb/22.4.88

Introduction.

The paper that in a limited number of pages shall give an overview of such a vast subject as the total transportation concept, embracing the three possible travelling elements, must be rather brief in its presentation of the various (concepts. The high-speed marine craft will, by definition, be considered a craft with a service speed above 25-30 knots and only surface crafts will be considered.

Likewise, for land transportation the road travelling vehicles, railroad systems above and below surface will be the land competitor and in air - planes, particularly commuter systems and helicopters, will close the transportation triangle - air/sea/-land.

In an attempt to limit the geographical areas under consideration., only Scandinavian and Continental countries, with the exception of the San Francisco Bay area, will be considered.

Focal areas will be limited by the fact that passengers with restricted space on board most high-speed crafts will only endure some 2-2.5 hours travelling time and thus, distance between ports is restricted to some 75-100 n.miles. The English Channel with its multiple crossing lanes between England and the Continent as well as England - Channel Islands, has been the domineering area for introduction of high-speed surface crafts.

Scandinavian waters have so far lagged behind in this development. although this year high-speed catamarans will be introduced in the Aaland routes out of Stockholm, as well as routes on the Swedish East Coast and Gotlanti. The Øresund network of ferries embraces a certain number of high-speed crafts, particularly operated by Øresundselskapet out of Copenhagen. across the straights to Malmø.

The long Norwegian coast has in the latter years seen a large number of high-speed crafts be introduced, starting on the West Coast of Norway, and today embracing practically the complete coast from Kristiansand to the isolated, weather beaten areas of the Finnmark coastline.

Types of surface crafts.

High-speed surface crafts embrace both pinning crafts, in mono hull and catamaran designations, and surface effect crafts defined by hovercraft and surface effect ships (SES).

Hydrofoils are considered a particular development, distinctly different from the two mentioned groups. Ground effect machines will not be considered, but referred to as a possible development in areas of inland water characteristics.

Common to all these categories of crafts is their sensitivity to deadweight increase. By virtue of high speed and the common desire for economy, the crafts have to be designed as lightweight crafts in all engineering aspects. Aluminum hulls and structure, or balanced by composite materials, high-speed, lightweight engines and lightweight equipment for machinery, navigation and outfit are the overruling strict design basis. The loaded displacement will, in the majority of cases, embrace some 75% lightweight, and only some 25% deadweight. The high-speed craft concept is thus limited to carrying passengers and lightweight cargo like passenger cars.

The rather common belief that high-speed crafts are suited at almost any speed for carrying trailers, containers and other types of cargo within these weight categories, must therefore be considered without factual grounding, in as much as the power required for propelling these crafts with this type of cargo would make them oil guzzlers without any financial and economic foundation.

Supporting these weights would, furthermore, drastically increase lightweight, thus even before taking any cargo on board. a heavy penalty on structural weight would penalize engineering power. The domino effect on structure, machinery output, auxiliary equipment etc. makes it clear that deadweight restrictions, as long as commercial activities are considered, will be the predominant limiting commercial factor as regards flexibility of utilization of high-speed crafts.

It is felt prudent that a few technical parameters should be given so as to ease comparison between different types of crafts, as well as quoting a certain technical denominator for sake of comparison between the different modes of transport.

Figure no. 1 (1) gives power requirements in calm water for a number of crafts as stated on the figure. This igure, although some 20 years old, is felt to give a good comparison between power requirements.

The superiority of hydrofoils with submerged foils, as well as the amphibious hovercraft, is readily apparent.

Figure no. 2 (I), which is from the same reference, gives an interesting diagram of sea handling characteristics which evidently, once more shows the superiority of thk submerged foil, hydrofoil craft and the difficulties that amphibious hovercrafts with air propulsion encounter in the slightest seas. This leads to the question of comfort. From the passenger viewpoint the most important operating characteristics when in marine crafts, can probably best be described under the heading of cost, comfort and convenience; speed itself is not necessarily a major attraction for the short routes on which most ferries are likely to operate. And it is thus, in this context, that the characteristics of any high-speed marine craft should be assessed.

Figure no. 3 (1) gives an interesting comparison from the same reference source, comparing vertical accelerations midships with the competing modes of transport under consideration. The in this context gentle ride of the hydrofoil (P.T. 50), compares favorably with the Vickers Viscount aircraft which, some 20 years back was a contemporary aircraft in shuttle and near continental traffic. It is perhaps somewhat surprising to see the poor comparison with busses and trucks on average roads. Admittedly, the standard of roads in the intervening period will most likely ensure that these figures by today's standards are very much lower than shown in this diagram.

Accelerations in waves, as shown in figure no. 4 (I), show again the superiority of the hydrofoil crafts compared with hovercrafts and planing crafts. The rather alarming accelerations, even in moderate wave conditions as experienced with planing crafts is interesting from the viewpoint of comparing mono hull and to a certain extent, catamaran hulls envisaged with higher payloads and comparably high sustained sea speeds. The forces which will be transferred to cargo and passengers in the seaway will, at least for the cargo, necessitate elaborate securing systems to ensure that the cargo stays in place in the seaway. The consequences of the cargo within containers are, of course, an entirely different matter. The admittedly, limited experience with utilizing fairly large catamarans in an attempt on regular schedules across the North Sea with frozen cargoes, underlines the size of forces as illustrated in this diagram.

Figure no. 5 (2) shows an up to date transport efficiency diagram including present day's SES-crafts, mono hulls, catamarans and hovercrafts as well as hydrofoils.

It is interesting from the viewpoint of embracing some 20 years' development as evident from this span of diagrams. that few differences are seen in the broad picture of efficiency between the different types of crafts.

In fact, it must be stated that the SES-craft is a derivative of the hovercrafts designed some 20 years ago, with fixed longitudinal walls and skirts forward and aft. The handling characteristics of the SES-craft wil! most likely be comparable to non-amphibious hovercraft (water propulsion) as shown on figures I and 2.

Figure no. 6 (2) shows a tabulation of different types of crafts based on passengers, installed power, full speed loaded which yields a factor of transport efficiency. This factor

$$\frac{\text{No. of passengers x max. cont. speed at full load (knot)}}{\text{Total installed power (kW)}}$$

proves an interesting comparison when using the same parameters on competitive transport modes in air and on land.

Figure no. 7, based on the same parameters, shows the different modes of sea transport compared with present day standards of Scandinavian bus transport, railroad and air transport.

It is obvious from this comparison that from a transport efficiency point of view, the high-speed surface craft does not compare favorably with rail, bus or aircrafts. Not surprisingly, the busses come out on top under the particular parameters given, which are stated on the figure. For sake of comparison, a high density, channel ferry has been calculated, which indicates that the very best of surface effect ships may compare from a transport efficiency point of view, with the new, large channel ferries recently introduced which, incidentally, are meant to be a competitor to the new Channel Project.

A comparison of a number of state of the arts crafts is shown in figure no. 8. This figure shows the lending particulars of the largest in service and proposed crafts of the various types of high-speed surface crafts under consideration in this paper. As evident from this figure the, in ship terminology main dimensions, reflect small crafts which by virtue of their high speed serve a limited transport demand restricted to passengers and lightweight car of limited amount.

The transport efficiency factors do not change by introducing the larger crafts under consideration.

It is evident that the design push for any type these crafts under consideration is lead by naval demands. From a commercial point of view, it appears that medium speed catamarans, mono hulls - particularly the derivative of hydrofoils into the Mono Stab design - as well as SES-crafts will dominate the larger crafts of tomorrow. The hovercraft appears to have a restricted area of utilization.

When introducing building costs, the comparison between figures no. 6, 7 and 9 becomes interesting. This explains, to a certain degree, why the hovercrafts, due to their high operating costs, are having a somewhat limited market penetration.

Speed is expensive, particularly if anything but passenger transport demanding higher deadweight is considered.

British Channel

The British Channel has been the birth place of high-speed sea travel. The data presented in this paper, particularly as regards hovercraft operation, describes development of the last two decades in passenger transport on the channel routes. Figure no. 10 shows the present percentage capacity by high--speed crafts in the passenger transport across the Channel. As shown, the transport is primarily by hovercrafts, although submerged hydrofoil transport (Boeing jetfoil) for years have had a limited operation in the Northern part of the Channel. The amount of traffic generated by these crafts have had a rather slow increase which appears not to have given the market potential foreseen with high-speed passenger transport across the Channel.

The introduction of commuter planes from city to city (Dockland Airport in the center of London) and drastically reduced waiting time in the airports prior to takeoff, will increase and harden competition with high-speed sea crafts.

The long debated. but now agreed to introduction of an undersea tunnel, although for rail transport only, will furthermore change and broaden the transport alternatives.

It appears that utilization of high-speed crafts in this area will have to compete on cost and convenience, and as such form, a limited part of the transport supply in the major part of the channel traffic.

The Channel Island traffic, on the other hand, is serviced by high-speed crafts both to England and to the Continent. In comparison to the Northern Channel traffic, catamarans and hydrofoils, but not hovercrafts, are utilized for servicing the islands. These crafts are purely passenger carrying crafts without any possibiiity of carrying cars or other types of cargo. The service has so far been entertained by different sizes of hydrofoils, although catamarans are steadily replacing the hydrofoils. Heavier types of traffic; cars, trucks and general cargo, is conveyed between the Islands, England and the Continent by ordinary displacement ships.
7 The development of larger SES-crafts carrying a certain amount of deadweight as regards passenger cars may be introduced, provided the economics are right.

It is obvious that the ordinary vessels utilized in this traffic are having rather high manning expenses. Operational characteristics on these very heavy crafts are also a burden to the operators. Introduction of semi high-speed vessels, like catamarans and other types of displacement crafts having a speed range of 20-25 knots with a fully automated machinery and low manning, may prove to be a development emanating from today's displacement craft operation.

Figure no. I I shows a 20-25 knot version of a composite built 500 passenger - 100 car capacity craft.

It appears that where commuter transport by' road, rail or sea can be directly compared, the conditions for sea transport must be based entirely on transit time which can be superior to road due to low road transit speed, because an appraised alternative which is favorably'competing with road and rail transport is difficult to realize.

High-speed commuter transport within the San Francisco Bay area utilizing catamarans and mono hull constructions, is one area where the sea transport has shown considerable potential and has captured a rather large slice of the commuter market.

For short, limited distances there is no air alternative and, unless a very fine network of rail systems has been developed, the real competitors are cars - busses and high-speed crafts.

Morning and afternoon, high density traffic throughout the world leads to low transit speed, so also in the Bay area. Figure nos. 12 and **13** show **typical high-**speed crafts utilized, public opinion of hover travel, as well as some of the service networks entertained by these crafts.

Scandinavia

Scandinavian waters are in some areas partly hampered by ice formation during the winter time. This is a problem which excludes lightweight, high-speed operating crafts unless some sort of hovercraft principle is utilized. **This** has, in fact, **been** utilized with considerable success in the Øresund scheduled traffic between Copenhagen and Malma. Figure no. 14 shows one of the 2 hovercrafts presently operating in a winter mode, where the only craft to operate during this particular season, was the hovercraft. It is interesting to note that commuter airplanes previously operated between these two cities, were abandoned in favour of the hovercraft regular transit.

The Øresund company of Copenhagen has for years been operating in parallel between the two cities. Hydrofoils were operated in the 70'ies, but for operational and economical reasons, these have been replaced by catamarans which today form the basis for the whole fleet. These high-speed catamarans (figure no. 15) are all passenger type only and operate in the high-speed mode of some 30-35 knots. The regularity of the service, apart from under ice ridden conditions, is satisfactory.

So far, very few high-speed crafts have been operated in the Baltic. The considerable traffic out of Stockholm, principally to Finland, has developed into a luxury passenger ferry fleet surpassing any other fleet in the world as regards quality, luxury and size of vessels.

Signs ar now, however, evident that high-speed catamarans of the Fjellstrand **type** will be introduced between Stockholm and the Aaland Islands. Due to the ice problem:; encountered during winter time, the operations must obviously be restricted to the ice free water time, normally some 9 months of the year. The sea conditions normally prevailing in the Baltic should not give rise to any operational problems.

It shall be of interest to note how these vessels can compare, which by virtue of size alone cannot meet to the standards offered passengers on the ships competing intensively in the Stockholm - Finland traffic pattern. The problems with confined space for passengers on board high-speed crafts have previously been commented upon in this paper, and a direct comparison to passengers' requirements, particularly when competing with some of the most top class, luxury vessels servicing any type of ferry network shall be interesting.

One could expect that the quality of passenger accommodation on board these catamarans must be considerably above the present, where the capacity is based on short time, commuter transport without any recreational facilities given. In this context it is interesting to note that the attempts to penetrate the Caribbean market out of Florida with similarly sized catamarans failed, primarily for operating reasons due to the vessel's inability to cope with the prevailing weather and wave conditions. It is, however, anticipated that the standards offered the passengers should also be carefully kept in mind when summing up the experiences from this operation.

In the Scandinavian waters there appears to be a tendency towards a sea transport system split into three groups.

High-speed passengers craft only
Semi high-speed passenger/passenger car capacity
Medium speed passenger/heavy loading cargo

In the Baltic between the East coast of Sweden and the island of Gotland, this type of operation will start during this summer with the introduction of high-speed, passenger only catamarans. The service offered passengers between these destinations precludes the use of road and rail traffic and, due to capacity and positioning of airport, the utilization of aircrafts will be less competitive than high-speed surface crafts.

A similar transport pattern is emanating rapidly along the Norwegian coast. Three decades ago the infant steps towards high-speed craft operations were taken in Norway. Hydrofoil crafts were operated in the Oslofjord and on th.: West Coast out of Bergen. These crafts were having some operational problem:. but particularly on the West Coast, opened completely new possibilities for operating between capital cities and the provinces in a sense that not had been possible with roads and ferries due to the lack of bridges and tunnels.

The operational problems experienced with these vessels were many, but lead to the introduction of the Norwegian designed and built catamarans, the Westamarans, which although not operating at the same speed, offered reliability in a different sense than had been experienced previously. The operational economics of operating these very often low density services, could not be commercially viable unless a heavy subsidy was granted by the national authorities. Figure no. 16 (6) shows a typical operational result of a typical traffic scheme on the Norwegian West Coast, with a spread of some 10 years.

The next generation of high-speed craft, the SES-craft, is still in its infant stages, but throws light on operations with speeds hitherto not possible with semi displacement catamaran crafts. The designs in operation so far are all passenger capacity design without any car or commercial cargo capacity. Figure no. 17 shows the SES-craft so far in service, presently by Troms Fylkes Dampskibsselskap in the Troms county. The operation has been very successful indeed, and it compares favorably with road transport.

The proposed next generation of SES-crafts. as shown on figure no. 18, is an ambitious, bold step towards a craft which, in GRP, will be the largest so far designed and built. This will be a combined cargo and passenger craft. The cargo being some 45 passenger cars, thus entering the area of modest cargo capacity in comparison to size of vessel. The data so far released, as shown on figure no. 18, justifies a few question marks as regards the performance and power required.

For sake of comparison, a similar craft is shown on figure no. 19 (5). This craft which has similar characteristics although apparently designed for somewhat higher speeds and sea capability, indicates by dimensions and power drastically higher powers, both for lifting and propulsion than shown on the Norwegian designed SES-craft. Provided the latter is justified from a point of view of design, it is apparent that the operational characteristics for such a craft would demand tremendous power with corresponding fuel consumption. Introducing this to the transport efficiency previously used in this paper, the data do not compare favorably. It is thus fairly obvious that such a craft would have problems obtaining the necessary state subsidies for operation in as much as the cost/benefit factors involved in this calculation do not yield the characteristics under which state subsidies most likely will be given.

By operating at these high speeds, it is obvious that a certain minimum distance between ports is required to obtain operational economy. Thus, the traditional route between Stavanger and Bergen is one where high-speed crafts have been operating for decades. We all know the sad story of the bold, next step intending to introduce high-speed SES-crafts with an average speed on some 42 knots into this route. The next step, replacing the ill-fated, Swedish built crafts, is still open. An introduction of the above mentioned, combined passenger and car capacity SES-craft could perhaps be of interest.

The Norwegian coast is in most aspects well suited for high-speed mode transport by sea. Although all parts of the coastline today is scattered with small airports, the number of people carried by sea is still formidable, and some 45-50 million passengers are still carried each year. The mode of transport is today predominantly by high-speed passenger crafts or ferries. The ferries taking all vehicle cargo, some 200 in operation are scattered around the coast.

The operatidnal pattern will most likely be similar to what is observed in other countries - high-speed passenger boats carrying passengers only competing with bus and air transport and in some parts of the country, also with rail transport. Medium speed passenger and passenger car/truck transport capacity - either by displacement boats having particular seakindliness, operating in the outskirts of the Norwegian coast - or by high-speed ferries. Slower transport for heavy vehicles could form the third leg of the transport pattern.

High-speed boats for servicing in commuter transport systems will, with the rather low density population, form a network of efficient transport means which cannot survive without heavy subsidies.

Comparing this, however, with the often met, local wish for introducing tunnels and bridges makes, on paper, the superiority of high-speed crafts readily apparent. It is felt that the maintenance and operational problems, particularly with tunnels, frequently are underestimated. This means that the toll to be paid by users of bridges and tunnels will only cover part of the expenses to be incurred. The expenses expected to be paid by the state for maintenance and operation may amount to considerable sums. From a hazard point of view, a collision between a petrol truck and a bus in the middle of a three kilometer long tunnel, some 150 meter below sea level, tells it all.

On the question of costs, the operation of any marine craft can be split in four group:;:

Capital costs
Fuel costs
Maintenance and operation costs
Administrative costs.

Traditionally, he operational costs have been dominated by manning costs.

The manning of traditional ferries have accounted for more than some 50% of the total operational costs. By introducing high-speed crafts manning levels have been slashed, thus pointing towards capital costs and fuel costs as the domineering parameters in the cost picture. Fuel costs are a derivative of speed requirements. With speed requirements from the travelling public ever increasing, the fuel costs have shown an explosive increase. Since these crafts are very deadweight sensitive, it is thus obvious that a choice has to be made; either speed with small carrying capacity or larger carrying capacity but slower speed. Thus, the SE-crafts breaks new ground as regards passenger transport - but not deadweight carrying capacity transport. Public definition of requirements endorsed by national authorities will, no doubt, define the fractional relationships between the different types of crafts to be utilized.

The administrative expenses are rather minor in the total operational expenditure framework - however, the maintenance and operational expenses are by no means minor. Time and again voices are heard claiming that operational expenses of high-speed crafts are high, particularly regarding machinery maintenance and in some instances, also structural maintenance. The latter is particularly related to GRP constructions due to, broadly speaking, the trial and error process, I would say, amply describing the state of art.

This leaves the capital expenditures. As long as these vessels are built in Norway and no competition is rendered by foreign companies - a certain level of cost is established. Presently, three nations worldwide are major builders of high-speed crafts - Norway - Sweden/UK and Australia.

The Far East building expenses as regards conventional crafts are some 40-50% ot the cost experienced with Norwegian - European costs.

It is thus to be foreseen, presumably within short, that the more aggressive Far East builders in Singapore, Hongkong and expectedly, Japan and Korea, will introduce crafts on the market built either under licence from established and experienced European - Australian designers, or designed on its own national design basis. This be as it may, however, costs will be slashed, and thus introducing different parameters into the total operating expenditures for this type of crafts primarily along the Norwegian coast.

With the turbulent situation regarding the future of Norwegian coastal passenger boat and ferry operation, it shall be of considerable interest to observe the parameters laid down by the authorities with regard to operational framework for this type of crafts.

Operational qualities will be based on a few basic parameters,

Economy
Reliability
Frequency

The economical factors involved when building locally and abroad are readily apparent and, with the change of operation, will make cost comparisons and cost competitiveness in favour of sea transport when compared with air and land transport. The changes that most likely will take place within the commuter systems in Norway will probably give high-speed boats added value.. The break-even distances between high-speed boats and commuter airplanes is felt to be some 75-100 n.miles. With high-speed passenger boats, this will entail some 2-2,5 hours travelling time between ports which are literally in the center of the cities as compared with some 30-45 minutes travelling time between airport terminals mostly outside the city centers, giving at least some 15 minutes additional travelling time at either end. With commuter planes, checking in before departure has been reduced to some 10-15 minutes which means that the minimum effective time from city center to city center will be some 1 hour 15 minutes or some 50% of the transit time by boat. It is then becoming a question of availability, cost and frequency whether the travelling public prefers boat or plane. The basically different subsidy structure between the two modes of transport will in most cases make it difficult to range one versus the other, however, boat transport will take a lion's share of the market with distances less than the break-even distance versus plane transport.

It is obvious where boats compete in parallel with road transport that this is a loosing battle.

Thus, boat and road transport should be viewed as interlinked transport systems rather than competing systems and the service network should be designed accordingly.

There is basically only one stretch along the Norwegian coast where rail transport can be directly competing with sea transport - Helgelandskysten in the county of Nordland. The frequency offered by the rail system, however, is not comparable to boat frequency. Thus, within the constraints given, this will lead to boats taking their major share compared with rail passenger transport.

Summarizing, sea transport in the high-speed craft mode will have a definite position in the transport service network along the Norwegian coast.

High-speed boats will due to their deadweight sensitive design, be restricted to passengers and very light additional cargo, for instance a limited number of cars per unit.

Semi high-speed boats and slow boats (ferries) will take the remainder of transport required in commuter traffic consideration.

Surface effect ships will develop along the lines given towards speeds of about 60-70 knots which will perhaps broaden the break-even distances quoted for passenger transport by sea versus air.

It all boils down to a question of economy. Because of the scattered population and vast distances to be covered, no operator, whether building the crafts in Norway or abroad, can make ends meet unless vast price hikes are imposed upon the travelling public, something which will be neither understood nor accepted by the passengers involved.

High-speed passenger boats within these give constraints are thus here to stay and prosper.

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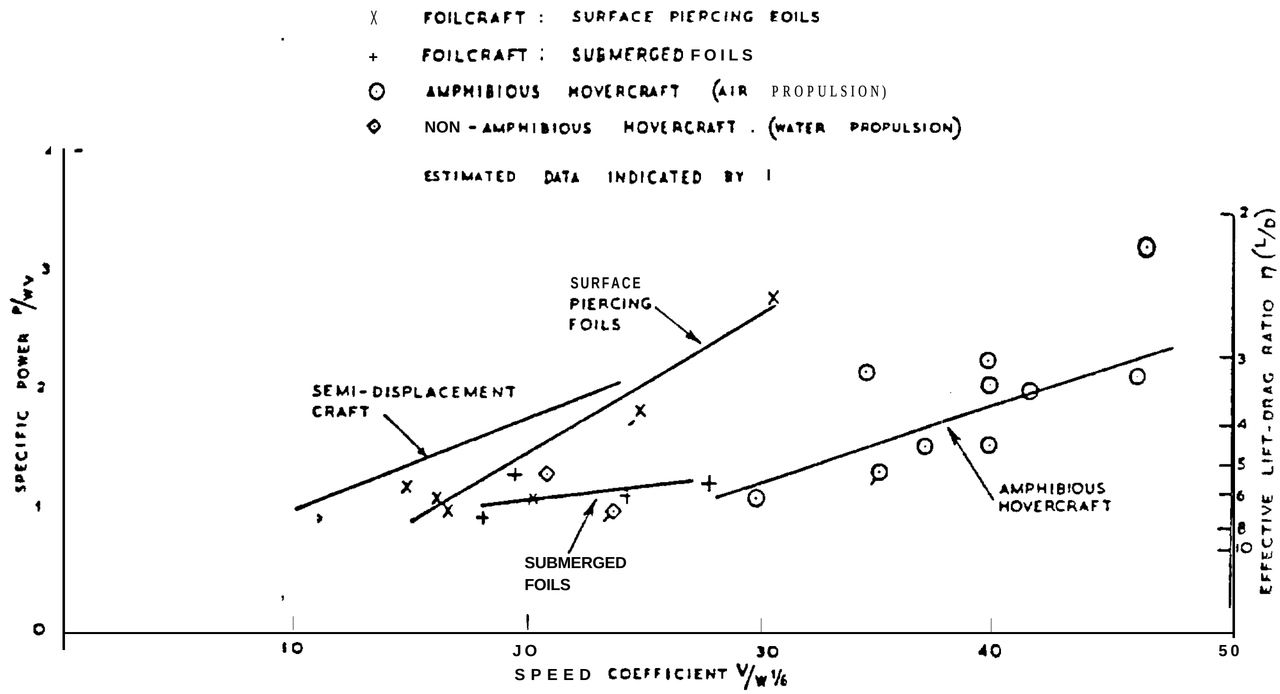


POWER REQUIREMENTS IN CALM WATER

P- TOTAL INSTALLED POWER (h.p.): W- ALL-UP WEIGHT (tons): V- SUSTAINED SPEED (knots)

L - TOTAL LIFT : D-TOTAL DRAG : η -OVERALL PROPULSIVE EFFICIENCY

Fig.no. 1

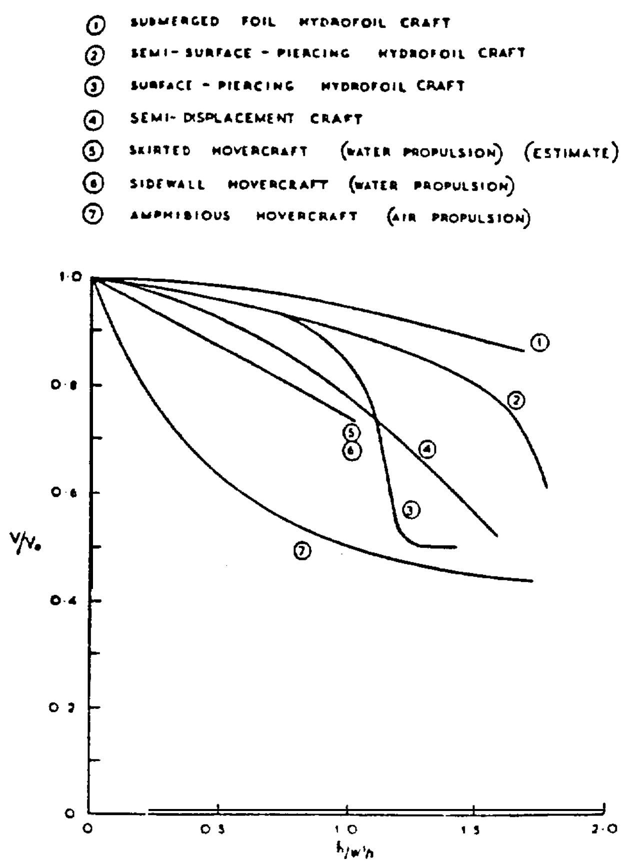
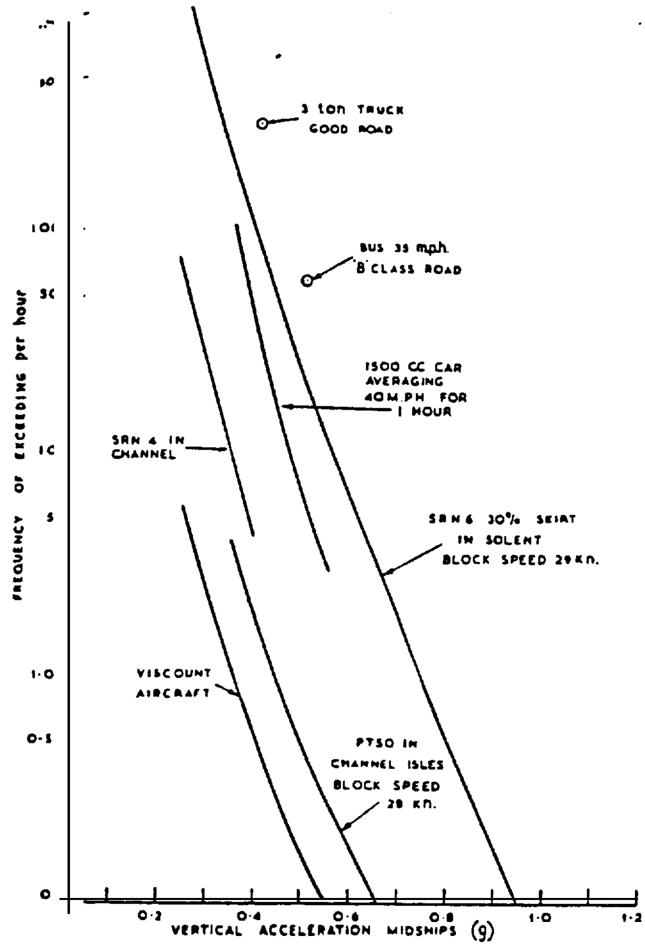


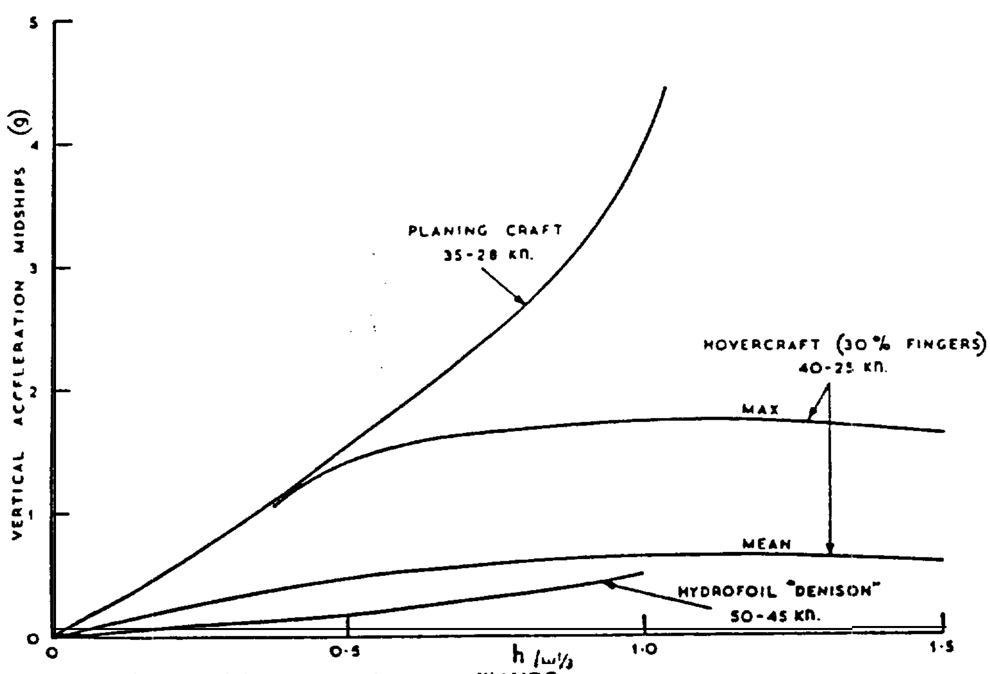
Fig. no. 2

SPEED-WAVE HEIGHT CHARACTERISTICS



MEASURED LEVELS OF VERTICAL ACCELERATION

Fig. no. 3



VERTICAL ACCELERATION IN WAVES

Fig. no. 4

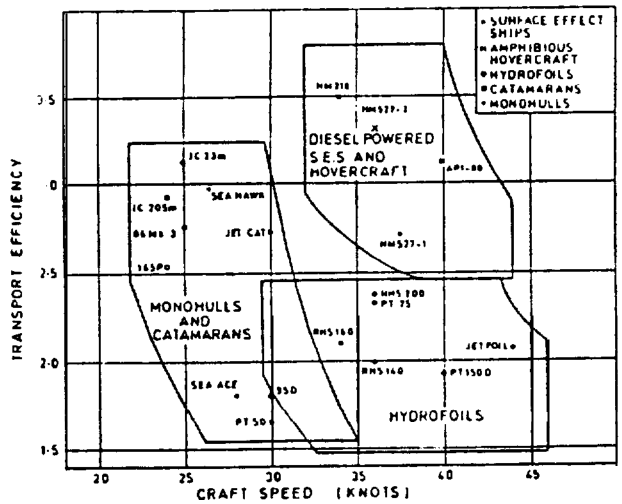


Fig. no. 5

The transport efficiency of High Speed Marine Crafts

#PASS X V / KW

Manufacturer	Type	Number of Passengers	Installed Power (kW)	Speed at Full Load (knots)	Transport Efficiency
Surface Effect Chips					
Vosper Hovermarine	HM218	64	616	34	3.49
Vosper Hovermarine	HM527-1	200	2764	37.5	2·71
Vosper Hovermarine	HM527-2	250-260	2625	36	3·2-3·31
Amphibious Hovercraft					
BHC	AP1-88	88	1128	40	3·12
Hydrofoils					
Boeing / Rodriquez Marine Systems / Cantiere Navale	RHS140 / 929-100	[illegible]	[illegible]	[illegible]	[illegible]
Rodriquez Cantiere Navale	RHS160	180	2909	34	2·10
Rodriquez Cantiere Navale	RHS200	250	3775	36	2·38
Supramar Hydrofoils	PT50	111	2013	30	1·65
Supramar Hydrofoils	PT75	160	2462	36	2.34
Supramar Hydrofoils	PTI SOD	270	5600	40	1·93
Catamarans					
Westamaran	86Mk3	181	1640	25	2.76
Westamaran	95D	178	2953	30	1·81
Jetcat Marketing	JC-Fl	215	2360	30	2.73
Fjellstrand	165P	169	1789	24	2·54
International Catamarans	20·5m	100	620	25	2·93
International Catamarans	23m	150	1193		3·14
Monohulls					
Mitsubishi	Sea Ace	230	3560	20	1·81
Mitsubishi	Sea Hawk 2	400	3560	26.5	2.98

Fig. no. 6

High Speed Passenger Ferry Transport Efficiency

Type	Number of passengers	Installed power (kW)	Speed at full load knots	Transport efficiency
Rail	400	2600	70	10.77
Bus	50	150	50	16.67
Air	130	10000	460	5.98
English Channel Ferry[x]	2000	15000	22	2.93

Fie. no. 7

[x] No account taken for cargo carrying capacity.

Type	Length, Beam (m)		Passengers	Cargo	Installed power (kW)	Speed (kn)	Transport efficiency
SES	60	25	500	76 cars	24200	54	1.12
Hover-craft	56	25	400	55	11100	>60	2.16
Wave piercer	70	28	500	90	> 10000	>35	1.75
Catamaran	50	14	400		4000	26	2.6

Fin. no. 8

Type	Number of passengers	Instd. power (kW)	Speed at full load knots	Transport efficiency	Capital cost per pass.knot US $
Hovercraft 8 HC	88	1128	40	3.12	994
Boeing Jetfoil	260	5532	44	2.07	1748
Fjellstrand 38.8	300	3600	32	2.66	469
SES	250	3500	40	2.85	450
Rail	650	3000	70	15.16	242
Bus	50	150	40	13.33	200
Air	130	11500	460	5.2	334
Channel Ferry	2000 (no freight incl.)	15000	22	2.93	909

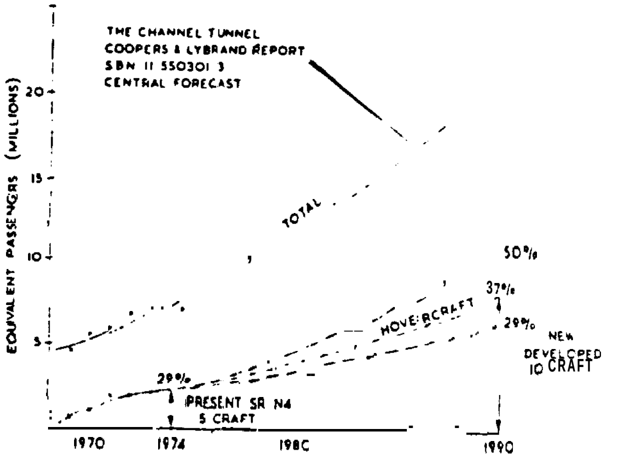
Fie. no. 9

Single Hoverspeed crossings scheduled March-October 1985–86

	Boulogne 1986	Boulogne 1985	Boulogne % change	Calais 1986	Calais 1985	Calais % change	Total 1986	Total 1985	Total % change
March	230	190	+21.05	404	340	+18.82	634	530	+19.62
April	272	268	+ 1.49	588	494	+19.03	860	762	+12.86
May	274	262	+ 4.58	694	600	+15.67	968	862	+12.30
June	300	258	+16.28	792	724	+ 9.39	1,092	982	+11.20
July	320	342	− 6.43	1,138	1,008	+12.90	1,458	1,350	+ 8.00
August	330	346	− 4.62	1,188	1,144	+ 3.85	1,518	1,490	+ 1.88
September	288	254	+13.39	832	808	+ 2.97	1,120	1,066	+ 5.46
October	186	186	–	522	410	+27.32	708	596	+18.79
Totals	2,200	2,106	+ 4.46	6,158	5,528	+11.40	8,358	7,634	+ 9.48

1985 English Channel fast ferry services

	Hovercraft 1985	Hovercraft 1984	Hovercraft % change	Jetfoil 1985	Jetfoil 1984	Jetfoil % change
January	384	221	+54.47	104	80	+30.00
February	308	290	+ 6.21	86	90	− 4.44
March	483	447	+ 8.05	160	176	− 9.09
April	629	686	− 8.31	198	202	− 1.98
May	851	760	+11.97	230	220	+ 4.55
June	964	928	+ 3.88	274	288	− 4.86
July	1,202	1,328	− 9.49	302	306	− 1.31
August	1,186	1,351	−12.21	286	308	− 7.14
September	908	914	− 0.66	278	260	+ 6.92
October	590	487	+21.15	226	184	+22.38
November	430	348	+23.56	146	190	−23.16
December	424	394	+ 7.61	144	210	−31.40
Totals	8,333	8,154	+ 2.07	2,434	2,514	− 3.18



Predicted Cross Channel Traffic-Short Sea Route

Fig. no. 10

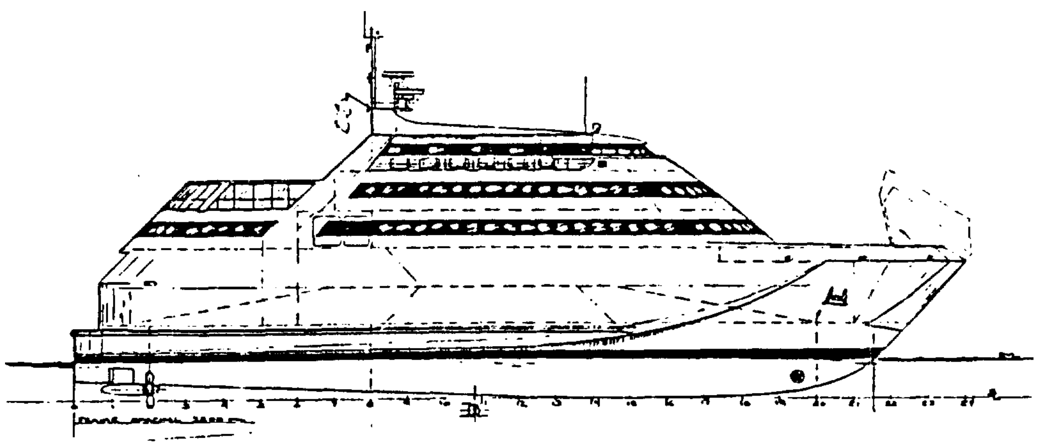
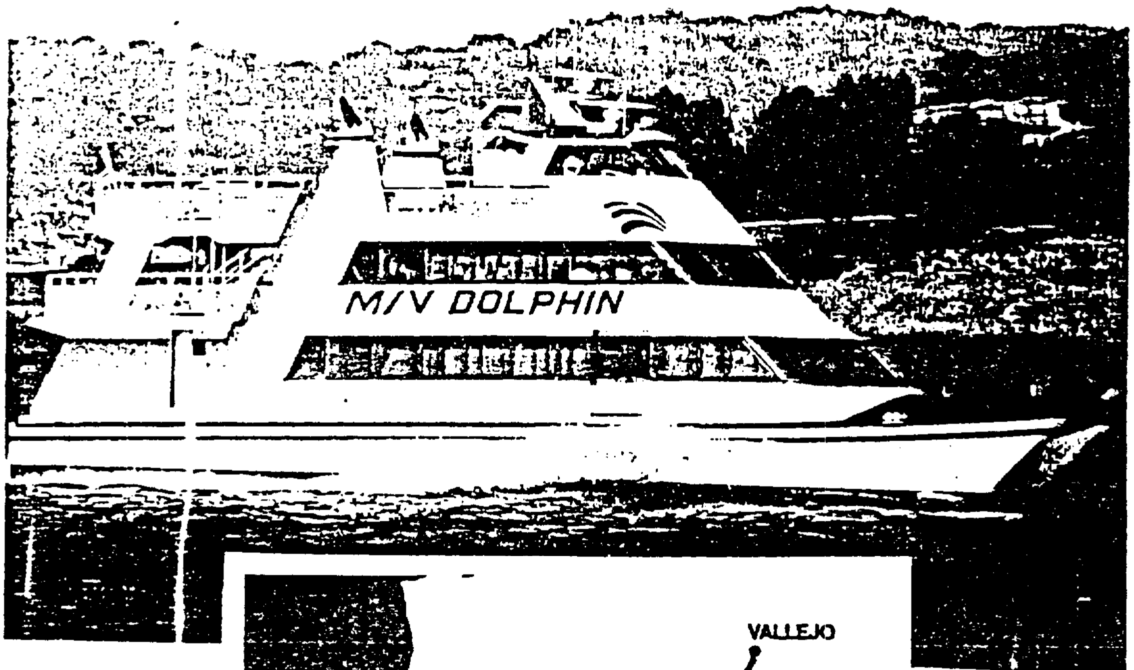
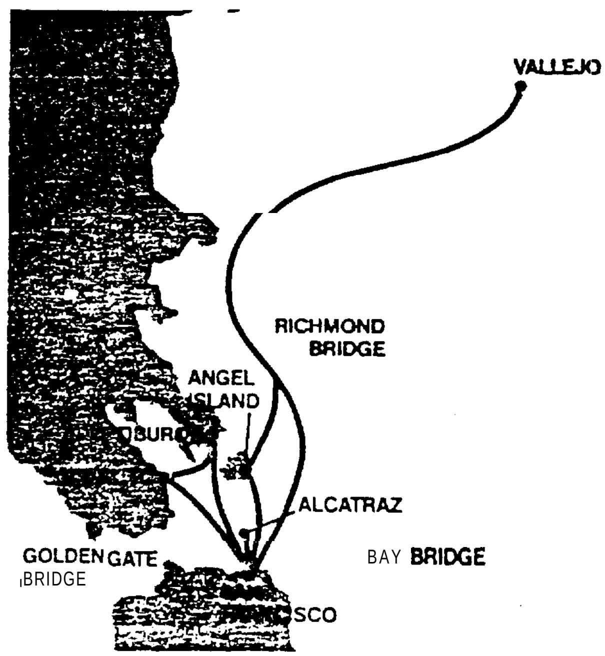


Fig. no. 11

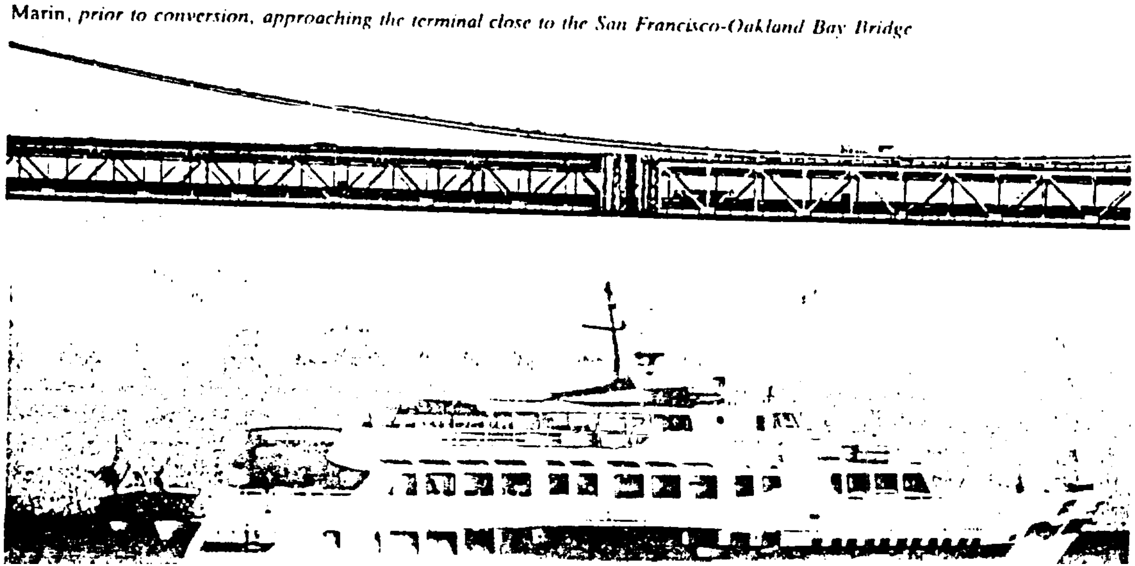


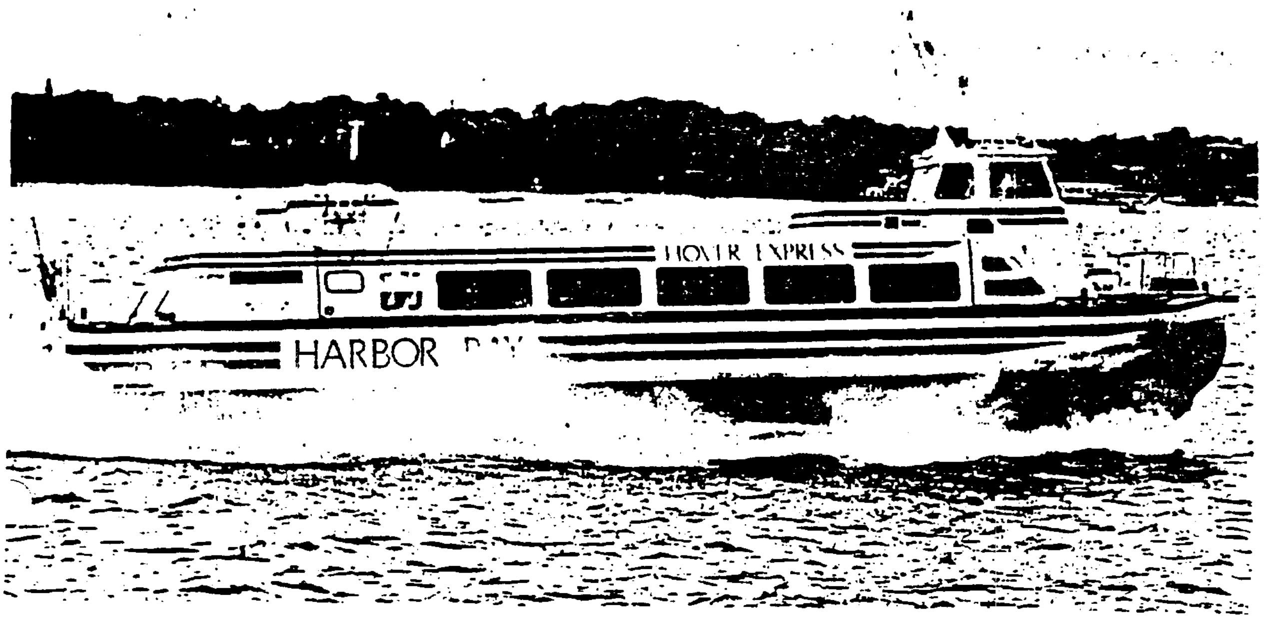


Red and White's scheduled services and the cruising area for its catamarans

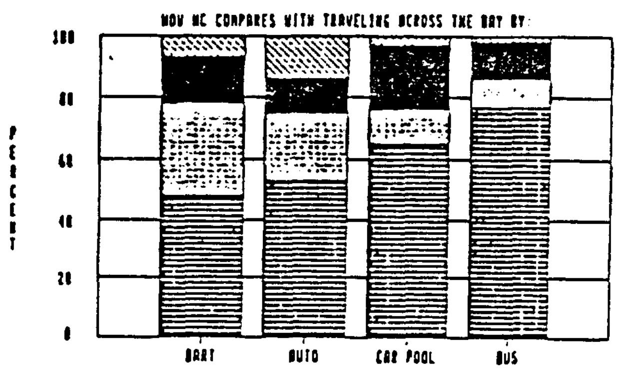
Fig. no. 12

Marin, *prior to conversion, approaching the terminal close to the San Francisco-Oakland Bay Bridge*





HARBOR BAY



How the guests rated the Hovermarine 218 compared with existing methods of crossing San Francisco Bay

Fig. no. 13

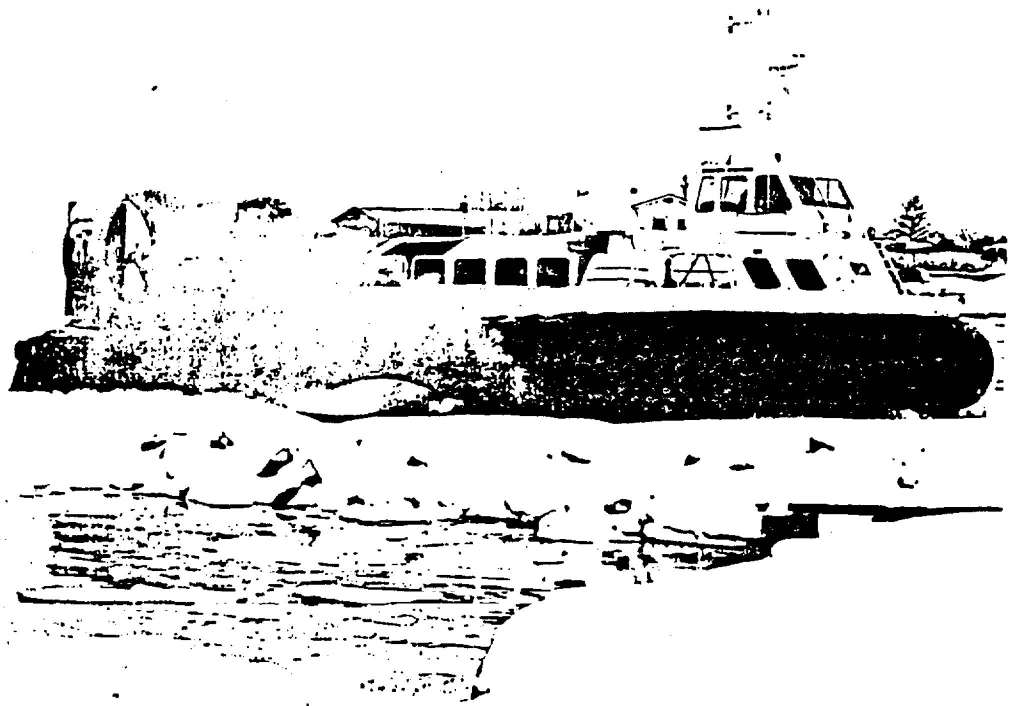


Fig. no. 14

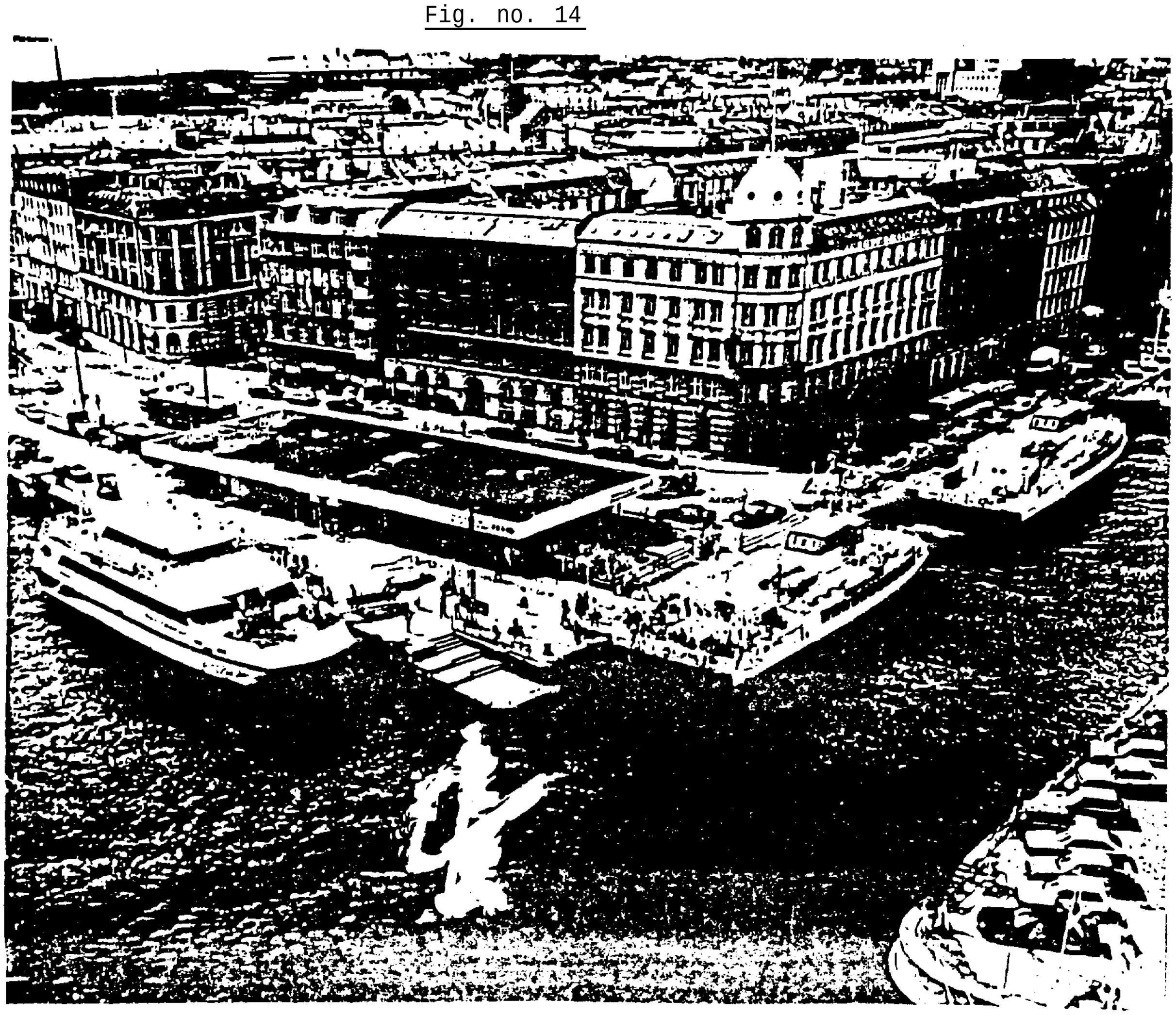
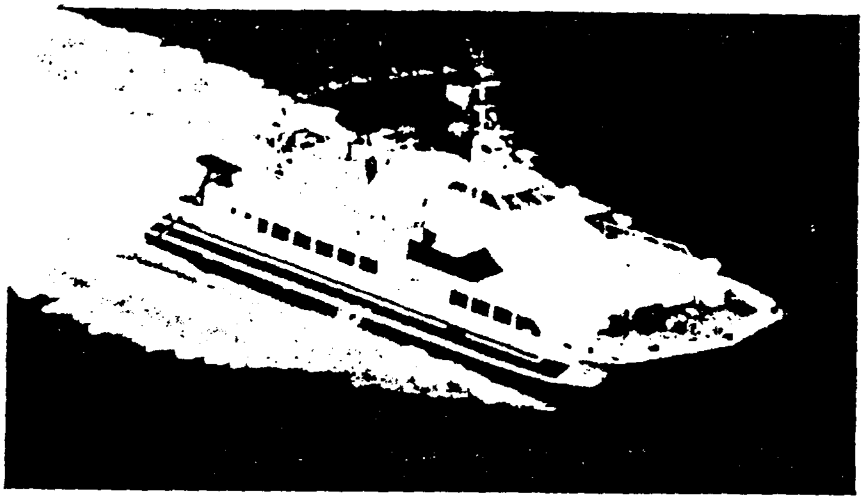


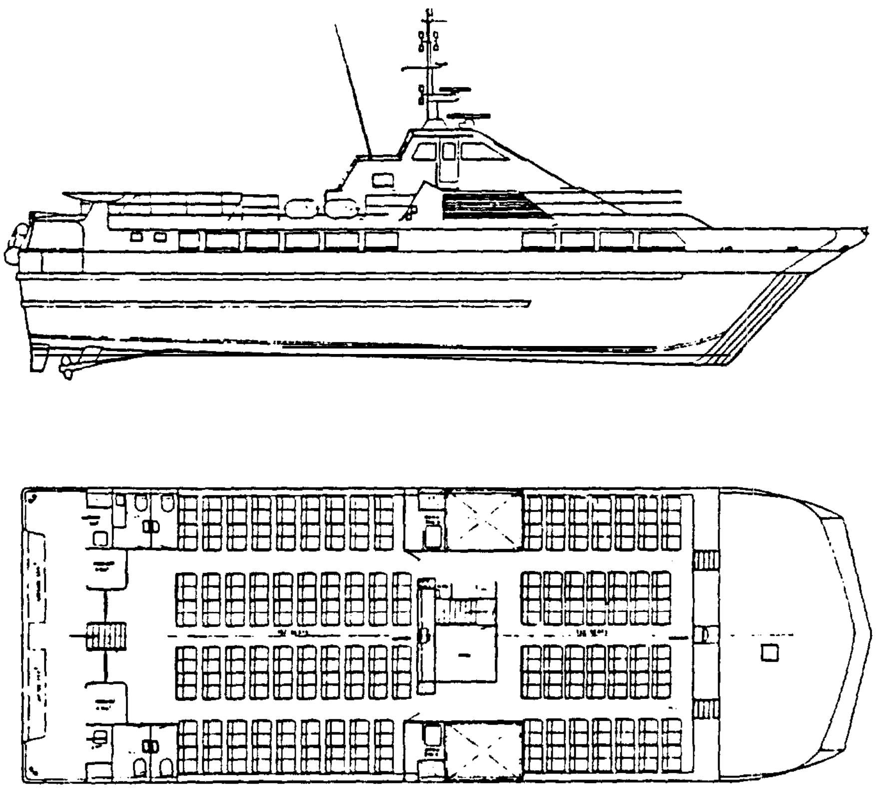
Fig. no. 15

Year	Income (NOK mill)	Cost: Operating vessel	Cost: Schedule cost	Result
1976	10.46	10.77	4.24	– 4 55
1978	12.34	12.77	1.15	- 1.58
1987	19 ≤7	22.53	≤.59	– 9 45

Fig. no. 16

Modifications made to the first CIRR 105P SES before delivery to TFDS included upper deck crew quarters





Outboard profile and passenger deck, SES Norcat showing original propeller installation

Fig. no. 17

Fie. no. 18

Lengde	55 m
Bilkap.	45 PBE
Passasjerer	300
Dødvekt	150 tonn
Hovedmotor	8000 HK
Fart	> 40 knop

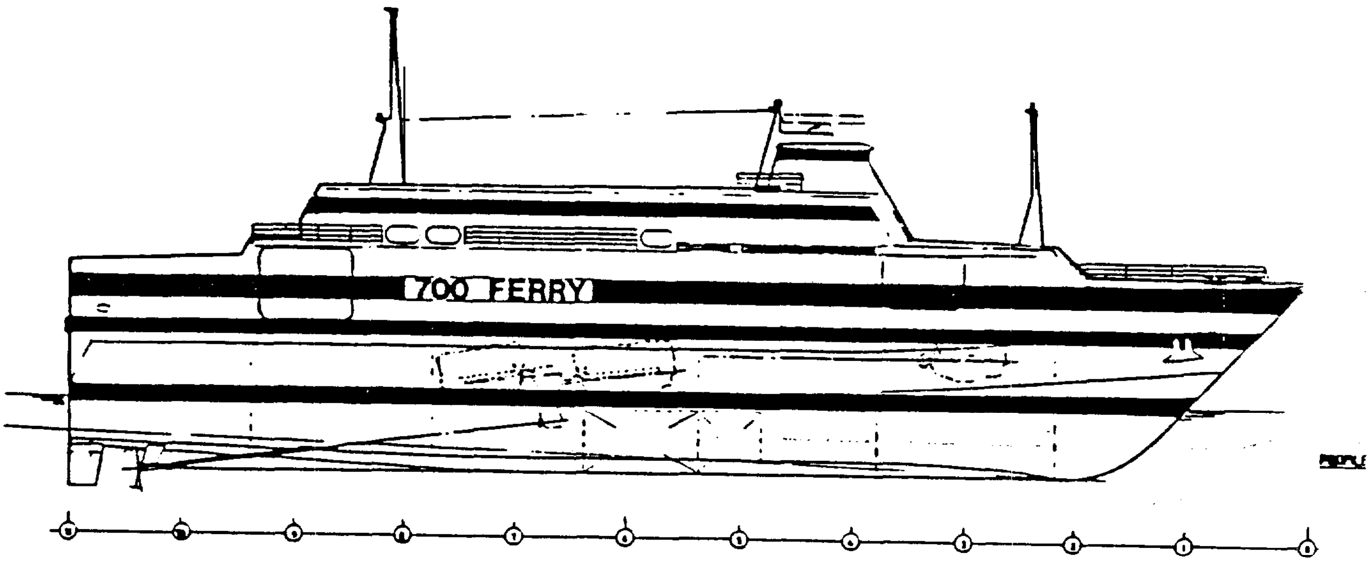


TABLE V - OUTLINE SPECIFICATION FOR DCC FAST FERRY

Length overall.	60.0M. (197 ft)
Beam overall.	25.051. (82 ft)
Cushion depth.	6.5M. (21 ft)
Draught on-cushion.	2.75M (9 ft)
Hull borne draught	4.50M (14.8 ft)

Capacity: 500 passengers and 76 cars
(or 60 cars and 4 coaches)

All-up-weight: 600t (590 tons)

Calm water performance: 54 kts

Speed in waves: 40 kts in sea state 5 to 6.

Installed power: Prop: 2 x 6100 kW (8175 bhp) MTU diesel
2 x 3000 kW (4020 bhp) MTU diesel

Lift: 2 x 3000 kW (4020 bhp) MTU diesel

Range at 50 kts: 750 nm.

Fig. no. 19

HOW TO CHOOSE A CONSEPT FOR HIGH SPEED MARINE CRAFT

Kåre Rygg Johnsen, Båtservice Verft A/S.

The present generation high-speed marine craft opens up for new commercial opportunities regarding seaway transportation.

The current development within areas such as:

* hull form and size
* air cushion and foil techniques
* propulsion systems
* materials

may have a significant influence on the total transportation picture.

There seems to be an increasing need for imfonation and rational methods which enable the operator to arrive at efficient transport concepts. In this paper such a method is indicated.

1. VESSEL CONCEPTS

The most common types of high-speed marine craft are:

* Monohull
* Small Water Area Twin Hull (SWATH) .
* Catamaran
* Surface Effect Ship (SES)
* Air-Cushion Vehicle (ACV)
* Hydrofoil, surface piercing foils
* Hydrofoil, submerged foils

See Fig. 1.

The most predominant qualities for each of these vessels are:

Type	Advantage	Disadventage
Monohull	* Large cargo carrying capacity * Ability to operate in high sea states	* Relatively large propulsional resistance
SWATH	* Favourable motion characterictics * Large deck area	* Poor cargo carrying capacity
Catamaran	* Large deck area * Favourable speed characteristics	* Restricted ocean going operation
SES	* Very good speed performance in still water * Large deck area	* Restricted ocean going operation * Wave and wind sensitive * Poor cargo carrying capacity at conventional cushion pressure
ACV	* Very good speed performance in still water * Zero depth in operation	* Wave and wind sensitive * Poor cargo carrying capacity * Restricted ocean going operation
Hydrofoil	* Favourable speed performance * Favourable motion characteristics	* Poor cargo carrying capacity * Restricted ocean going operation

2. PROPULSION SYSTEMS

Until relatively recently it has been accepted that at speeds slower than about 40 knots, waterjet propulsion is less effecient than conventional screw propellers. Comparisons have been made over the years (e.g. References 1 and 2), of which figure 1 is typical.

However, in recent times some waterjet manufacturers have been claiming slightly over 60% propulsive efficiency, and suggested that the "cross-over" speed is in the 20-25 knot region as shown in Fig. 3 (reference 3). The vessel in question here is a 35 meter SES. Fig. 3 also indicates that there are new types of propeller systems being introduced which claim higher efficiency than the conventional ones.

In order for an operator to select a propulsion system that is optimal with respect to special requirements, technical and economical facts as well as subjective opinion, it is helpful to establish an evaluation matrix, see chapter 12.

The following evaluation is to be regarded as an example on how the evaluation method works. The weight factors are assumed to be filled in by the operator. (The weight factors very often reflects subjective opinions and special operational circumstances.) However, the points given may to some extent be representative for a typical waterjet and a typical CP-propeller.

Example on main propulsion evaluation (SES, L = 50 m, v = 40 knots).

Parametre	Weight factor	Water jet		CP-propeller	
		Points	Score	Points	Score
* Propulsion efficiency at service speed	6	5	30	4	24
* Propulsion efficiency at slow speed	2	3	6	5	10
* cost	6	4	24	3	18
* Weight	4	6	24	4	16
* Reliability	4	5	20	5	20
* Bollard pull	1	3	3	6	6
* Depth under keel	2	6	12	2	4
* Noise & vibration	3	6	18	3	9
* Manouverability	5	5	25	4	20
Total score			162		127

Weight factor: 6 = important, 1 = negligible
Points: 6 = best, 1 = worst (Score = weight factor x points)

In this case the waterjet is considered to be more suitable than a CP-propeller. However, the outcome is totally dependant on the selection of parametres and the weight factors (which will vary from case to case).

3. SPEED PERFORMANCE IN STILL WATER

The speed performance of the various concepts is of course dependent on the individual design. In Fig. 4 and Fig. 5 is shown typical speed performance curves in still water for vessels of L = 50 m (Fig. 4) and vessels of deadweight = 185 tons (Fig. 5), ref. 4.

A general observation for cargo carrying vessels is that the SES has a favourable still water speed performance compared to other hull configarations above 25 knots.

4. MOTION CHARACTERISTICS

In ref. 4 a seakeeping comparison is made between the different vessel concepts.

When motion characteristics are to be evaluated it is important to differ between motions in relatively calm waters (e.g. significant wave height of 2 m) and "rough seas" (e.g. significant wave height of 5 m).

The seakeeping performance of a vessel may be characterized by various parametres such as absolute or relative motions (between vessel and sea), motion angles, velocities and accelerations. In addition to these criterions come other service restricting parametres such as slamming loads, green water on deck, lack of dynamic stability, etc.

In broad terms a ranking of the different types of vessels seems to be as follows:

"Calm seas"

* SWATH
* Hydrofoil
* Catamaran and SES
* Monohull

"Rough seas"

* SWATH (when sufficient air gap)
* Monohull
* Catamaran
* SES
* Hydrofoil

The critical parametre for each design, w.r.t. service retricitons, seems to be:

SWATH : air gap

Hydrofoil : distance btw. keel and surface (or distance btw. foil and keel for submerged foils)

Catamaran : air gap, relatively small buoyance in forebody

SES : air cushion leaks, air gap, relatively small buoyance in forebody

Monohull : bow height, etc.

The service restriction is strongly dependant on the size of the vessel. Generally the seakeeping performance improves with the size of the vessel.

The classification societies have given "Service Restrictions Natations" to all "Light craft" vessels. The service restriction will in addition to the limited operational range from refuge specify reductions in speed versus sea state.

In addition to classification rules there are other seakeeping standards of which some deals with comfort for passengers or working conditions for the crew.

The IS0 criteria for encounter frequencies of 1.0 Hz and above for example indicates for example that light manual work is impared at vertical acceleration of 0.4 g and larger. If this criterion is used to evaluate the operational restriction of different vessel types with a displacement of about 200 tons and a service speed around 40 knots, the soeed should have to be reduced at the following wave heights if the work took place in the bow area (see, ref. 4):

Restrictions due to accelerations at bow:

Ship type	Speed	Sign. wave height
* SWATH	30 knots	3,6 m
* Hydrofoil	45 knots	2,6 m
* Catamaran	35 knots	1,8 m
* SES	45 knots	1,1 m
* Monohull	20 knots	1,0 m

It should be noted that the monohull has relatively excessive bow accelerations. If the midship area had been the site in question, the results would have been:

Restrictions due to accelerations midship:

Ship type	Speed	Sign. wave height
* SWATH	30 knots	4,0 m
* Hydrofoil	45 knots	3,2 m
* Monohull	20 knots	2,7 m
* Catamaran	35 knots	2,5 m
* SES	45 knots	1,5 m

5. SPEED REDUCTION IN HEAD SEAS

An important parametre for most high speed craft is the speed reduction qualities in head wind and seas. At high speeds the aerodynamic resistance may amount to a considerable part of the overall resistance. This is particularly true for SES and ACVs.

The same vessels as mentioned in ch. 4 have the following speed reduction characteristics in head seas:

Speed reduction characteristics:

Ship type	Speed still water	Speed sign. waveheight 3 m	Reduction in percent
* SWATH	30 knots	26 knots	13%
* Hydrofoil	45 knots	36 knots	20%
* Catamaran	35 knots	27 knots	23%
* Monohull	20 knots	15 knots	25%
* SES	45 knots	18 knots	60%

6. MANOUVERABILITY

The manouverability is of course dependent on propulsion system, rudder arrangement, etc. In general terms, however, the following ranking is mostly true with respect to mdnouverability in a slow-speed situation:

* SES (on cushion)
* Catamaran
* Monohull
* SWATH
* Hydrofoil

At high speeds the ranking would have been:

* Catamaran
* Monohull
* SWATH
* SES
* Hydrofoil

The SES has a favourable manouverability at low speeds due to large transverse separation of the propulsion units. This arrangement also improves the position holding capacity. Due to the small draft, relatively large areas exposed to wind and small frictional resistance and drag forces, the SES is, however, relatively sensitive to wind (and less sensitive to current).

The catamaran has the same adventage as the SES w.r.t. the distance between the propulsion units and thus a relatively high degree of manouverability.

7. COMFORT ONBOARD

The comfort onboard is often related to Tuxurious interior. However, qualities such as

* noise & vibration
* accelerations

are essential w.r.t. comfort.

The noise and vibration conditions onboard is often related to the position of the main engine and the propulsion system. A waterjet produces less noise and vibration than a propeller under normal circumstances.

The following vessel-types are relatively easy to control w.r.t. noise and vibration:

* SWATH
* Monohull
* Catamaran

whereas

* SES
* Hydrofoil

may be somewhat more complex to deal with.

a. BUILDING MATERIAL

There are two predominant groups of material normally applied for high speed craft, namely aluminium alloys and fibre reinforced plastic.

If the fire protection is done properly and if the desigh - both 'globally and locally -is well taken care of along with the workmanship, both material groups are well suited for building of fast craft. Both aluminium and FRP or FRP/sandwich is accepted by-the classification societies on an evenly basis.

In table 8.1. an evaluation of steel, aluminium, FRP/single skin and FRP/sandwich is carried out. (For method of evaluation, see ch. 14.)

Evaluation of materials for a high-speed craft:

Parametre	Weight factor	Steel		Aluminium		FRP/single		FRP/sandwich	
		oints e---e-	score	points	score	points	score	points	score
* Weight	2 x 6		12	5	60	3	36	6	72
* cost	3	:	18	5	15	3	9	4	12
* Production QA/QC	3	5	15	4	12	3	9	4	12
* Fire resistance	4	3	12	2	8	6	24	4	16
* Impact resistance	2	6	12	4	8	4	8	4	a
* Ice perform	2	6	12	4	8	4	8	3	6
* Fatigue properties	3	4	12	4	12	6	18	6	18
* Water degradation	3	6	18	5	15	6	18	6	18
* Noise	3	2	[illegible]	2	6	4	12	6	18
* Thermal insulation	3	1	3	1	3	4	12	6	18
* Damage detection	3	6	18	6	18	4	12	3	9
* Maintenance & repair	4	6	24	5	20	4	16	4	16
Total score			162		185		181		223

Weight factor : 6 = important
1 = negligible
Points : 6 = best
: 1 = worst
: o = unacceptable
Score : Weight factor x points

Table 8.1.

As the weight is the most predominant parametre for high-speed craft, the weight factor has been doubled in this evaluation. As explained earlier, the weight factor is based on a somewhat subjective opinion. It may vary according to type of trade, exposure to rough seas, frequency if impact-loads at quay, climate (ice), distance to repair facilities, etc.

Nevertheless, it seems that FRP/sandwich is a material well suited for light craft whereas steel, due to its high specific weight; is not too well suited. In many cases there may be adventageous to use a combination of different materials in order to utilize the best qualities available.

Fig. 6 shows the weight distribution of a typical passenger catamaran. It is seen that the hull weight accounts for more than half of the light ship. This explains the double weight factor in table 8.1. Fig. 7 shows that approximately half of the hull weight is plates, the other half internal structure. If steel plates are used instead of aluminium the plate thickness may be reduced in the order of 1 mm for aluminium thickness of 5 mm at an average. A greater reduction is not realistic due to requirements to bucking, weldability, etc. Taking this reduction into account along with the specific weights of steel and aluminium, the weight of the plates will increase 2.3 times. Even if the rest of the structure will not increase accordingly it is expected that going from aluminium to steel will approximately-double the hull weight.

A similar comparison between aluminium and GRP/sandwich shows that a 10% weight reduction is typical by going from aluminium to GRP/sandwich. In real life these weight figures have a distribution dependent on weight efficient design as shown in Fig. 8.

Fig. 9 shows typical cost per weight figures for GRP/sandwich, aluminium and steel when the work hours are included. Fig. 10 shows a typical distribution of construction costs for a 40 m catamaran in aluminium. This figure reveals that the material cost is relatively small compared to other costs. Therefore new, efficient FRP/sandwich technology should be considered even if the unit price is high.

S-glass woven roving in a polyester matrix is for example about 3 times as expensive as normal E-glass roving/polyester (per weight unit). However, the ultimate tensile strength is in the order of that of steel and if this material is used in 10% of the most exposed parts of the structure, the total price of the vessel will increase by only about 1%, whereas the overall strength may have improved considerably.

9. CARGO CARRYING CAPACITY

The cargo carrying capacity can be ranked as follows:

* Monohull
* Catamaran
* SES
* SWATH
* Hydrofoil

The SWATH has favourable motion characteristics; but its cargo carrying capacity is poor due to the small water area. The only way to compensate this is to allow operation at pontoons, or to take onboard ballast in transit condition. Either the favourable motion characteristics are lost or fuel is burnt to transport ballast.

The SES has reasonable cargo carrying capacity up untill a certain point. If increased capacity is needed and the L/B/d-figurations are to be kept within reasonable limits, the only way to achieve extra load carrying capacity is to increase the cushion pressure. This turns out to be relatively expensive.

All in all one may say that the monohull is best suited for transportation of cargo with high specific weight. Catamaran and SES are well suited for light weight cargo, cargo which needs large deck areas and passengers.

10. BUILDING COSTS

Fig. 11 shows relative vessel costs per unit weight as a function of speed (ref. 5).

The NSFI group system split the ship into 8 groups:

* General
* Hull
* Equipment for cargo
* Ship equipment
* Equipment for crew and passengers
* Main engine w/equipment
* Auxiliary engine w/equipment
* Ship systems

Fig. 10 shows the price distribution of a 50 m lang monohull for speeds of 20 knots and 30 knots. As it is basicly the main engine that increases in price as the speed goes up, the figure demonstrates that speed is a costly parametre for a monohull.

A catamaran would have been less sensitive to the same speed increase, The SES even less, however, when size and cargo carrying capacity is increased, the cushion generating machinery will increase relatively rapidly.

It is difficult to give a general cost comparison between the different vessel types. However; in the following table it is indicated where each type seems to have the relatively best performance when relatively calm seas are assumed (conf. ref. 4).

Displacement \ Speed	0-20 knots	20-40 knots	40-60 knots
0 - 500 t	Mono	Catamaran SWATH SES	SES Hydrofoil
500 - 1000 t	Mono	Catamaran SWATH SES	SES Hydrofoil
1000 - 5000 t	Mono	Catamaran SWATH	SES

11. OPERATIONAL COSTS

The operational costs can for example be split in:

* capital costs
* fuel costs
* crew costs
* maintenance and repair
* harbour and canal toll

The capital cost is reflected by the table shown in ch. 10. The fuel costs are of course dependent on the present oil price. However, for fast craft the fuel is a predominant cost. A vessel of 20.000 hp in continous service burns fuel for around 10.000 USD per day.

The corresponding figure for a passenger vessel with a 5000 kW main engine and 10 hours operation per day would be around 1.500 USD.

12. EVALUATION METHOD

When various technical solutions and trade concepts have been analysed it often appears to be difficult arriving at a final conslusion due to the complexity of the matter. Although a variety of comparable technical and economical facts are at hand, the problem is normally to extract the important factors from the less important ones and to take into account parametres which are difficult to quantify.

In these circumstances an evaluation matrix is of great help. The technical and economical parametres are listed vertically. Each parametre is then to be given a weight factor according to the relative importance and points according to relative quality or capacity. The product of weight factor and evaluation point makes the score of the parametre in question. The alternative with the highest total score (sum of all scores) is then considered to be the most favourable one.

Parametre	Weight factor	Alt 1 Points	Alt 1 Score	Alt 2 Points	Alt 2 Score
No 1					
No 2					
:					
No i					
:					
:					
No n					
Total score					

Weight factor:

1 2 3 4 5 6

negligible … important

Points:

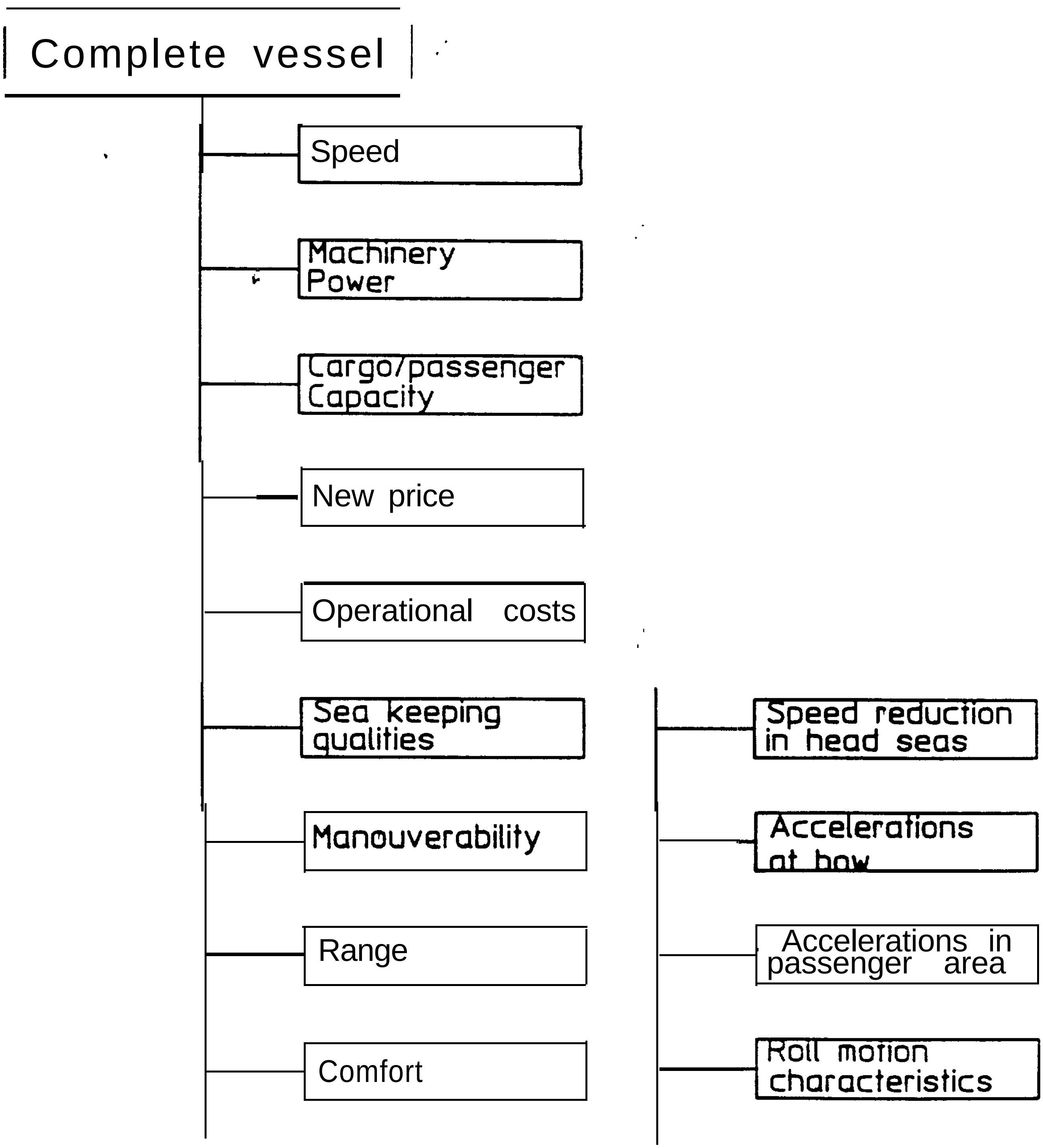
1 2 3 4 5 6

poor performance … top performance

$$\text{Score}_i = \text{weight factor}_i \times \text{points}_i$$

$$\text{Total score} = \sum_{i=1}^{n} \text{score}_i$$

When the total concept, for example a SES versus monohul or catamaran is to be concluded, it is often useful to build up the evaluation matrix hierarchically:



13. EXAMPLE ON SELECTION BASED ON THE EVALUATION MATRIX

The evaluation method outlined in ch. 12 has been used to select a vessel type for transportation of passengers/cargo or a plane cargo vessel. The basic capacities of the vessel are said to be:

Payload	: 185 tons
Range	: 500 nautical miles
Service speed	: Alt. 1: 25 knots Alt. 2: 50 knots

The following vessel data is taken as a basis for the concept evaluation.

Speed = 25 knots

Data / Vessel type	Monohull	Catamaran	SES	SWATH
LOA M.E. power	86 m 7000 hp	78 m 8300 hp	64 m 7000 hp	53 m 18000 hp
Estimated operating costs	11.800 USD/day	12.000 USD/day	12.200 USD/day	19.400 USD/day

Speed = 50 knots:

Data / Vessel type	Monohull	Catamaran	SES	SWATH
LOA Y.E. power	89 m 47000 hp	81 m 36000 hp	67 m 20000 hp	n.a.
Estimated operating costs	38.000 USD/day	31.000 USD/day	20.000 USD/day	

The vessels are assumed to be built in FRP/sandwich. (Aluminium might have been chosen without any significant influence on prices and weights.)

The operating costs include running costs such as crew; husbanding, insurance, harbour toll, consumables, maintenance, minor modifications, etc., as well as capital costs and fuel costs. The operating cost is based on continous operation during the day.

Before an evaluation matrix is to be established some characteristic sea keeping qualities should be estimated. In the following table the speed reduction in seastate 4 and 6 is given (speed reduction due to the environment). Also the maximum significant wave hight in which the vessel may operate according to the 0.4 bow-acceleration criteria is given.

v = 25 knots (head seas is assumed)

Criterion \ Vessel type	Monohull	Catamaran	SES	SWATH
Speed reduction, ss 4	2 knots	1,4 knots	4,0 knots	0,5 knots
Speed reduction SS 6	5 knots	5,0 knots	12,0 knots	1,7 knots
Max. allowable wave height (0,4 g bow)	5 m	4,9 m	3,8 m	7,5 m

v = 50 knots (head seas is assumed)

Criterion \ Vessel type	Monohull	Catamaran	SES	SWATH
Speed reduction ss 4	5,0 knots	3,0 knots	12,5 knots	n.a.
Speed reduction SS 6	15,0 knots	10,0 knots	27,0 knots	n.a.
Max. allowable wave height (0,4 g bow)	4,2 m	4,2 m	3,1 m	n.a.

SS 4 abt. 2,1 m significant wave height
SS 6 abt. 4,5 m significant wave height

The tables show that the SES has a significant speed reduction in head seas.

The accelerations of a SES is of course dependent on the ride control system. The accelerations in the above tables assume the SES being fitted with a reasonably good ride control system..

In the following tables the monohull, catamaran, SES and SWATH is compared by means of an evaluation matrix as describen in ch. 12. Only the highest level in the hierarchy is shown. Two speed, 25 an 50 knots, and two sea states, "Calm weather" and seastate 6, are considered:

v = 25 knots, "calm seas"

Vessel type / Parametre	Weight factor	Monohull Points	Monohull Score	Catamaran Points	Catamaran Score	SES Points	SES Score	SWATH Points	SWATH Score
Building price	6	4		5	30	6	36	2	12
Operating price	6	5	[illegible]	5	30	5	30	3	18
Speed reduction head seas	4	5	20	5	20	4	16	6	24
Motion charac- teristics	2	4	8	5	10	5	10	6	12
Manouverability dead slow	2	3	6	4	8	5	10	3	6
Manouverability service speed	1	5	5	5	5	4	4	4	4
Total score			93		103		106		76

v = 25 knots, seastate 6

Vessel type / Parametre	Weight factor	Monohull Points	Monohull Score	Catamaran Points	Catamaran Score	SES Points	SES Score	SWATH Points	SWATH Score
Building price	6	4	24	5	30	6	36	2	12
Operating price	6	5	30	5	30	5	30	3	18
Speed reduction head seas	5	4	20	4	20	1	5	6	30
Motion charac- teristics	4	4	16	4	16	2	8	6	24
Manouverability dead slow	1	3	3	4	4	5	5	3	3
Manouverability service speed	2	5	10	5	10	4	8	4	8
Total score			103		110		92		95

v = 50 knots, calm seas

Vessel type / Parametre	Weight factor	Monohull		Catamaran		SES		SWATH	
		Points	Score	Points	Score	Points	Score	Points	Score
Building price	6		18	4	24	6	36	n.a.	
Operating cost	6	3	18	4	24	6	36		
Speed reduction head seas	4	5	20	4	16	3	12		
Motion characteristics	2	4	8	5	10	5	10		
Manouverability dead slow	2	'3	6	5	10	5	10		
Manouverability service speed	1	5	5	5	5	4	4		
Total score			75		89		108		

v = 50 knots, sea state 6

Vessel type / Parametre	Weight factor	Monohull		Catamaran		SES		SWATH	
		Points	Score	Points	Score	Points	Score	Points	Score
Building price	6	3	18	4	24	6	36	n.a.	
Operating cost	6	3	18	4	24	6	36		
Speed reduction head seas	5	4	20	5	25	1	5		
Motion characteristics	4	5	20	4	16	2	8		
Manouverability dead slow	1	3	3	4	4	5	5		
Yanouveralibity service speed	2	5	10	5	10	4	8		
Total score			89		103		98		

With the chosen weight factors and points given the following conclusions can be drawn:

* The SWATH would require unrealisticly large engine power for high speeds which makes it unsuitable for the 50 knots-case.

* The SES is the most favourable alternative for both speeds in calm seas.

* The catamaran is the most favourable alternative for both speeds in seastate 6.

It should be mentioned that the required deck area is relatively large compared to the cargo weight (185 tons).

If the deck loading had been increased it is assumed that the monohull would have come out more favourably.

Nevertheless, both the catamaran and the SES appear to be interesting alternatives to traditional, low to medium speed, monohull transportation, particularly for high-value, light cargoes.

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5. "Advanced Marine Vehicles: A Critical Review", P. van Oossanen, MARIN, The Netherlands, CETENA, Genova 1978.

TYPES OF HIGH SPEED LIGHT CRAFT.

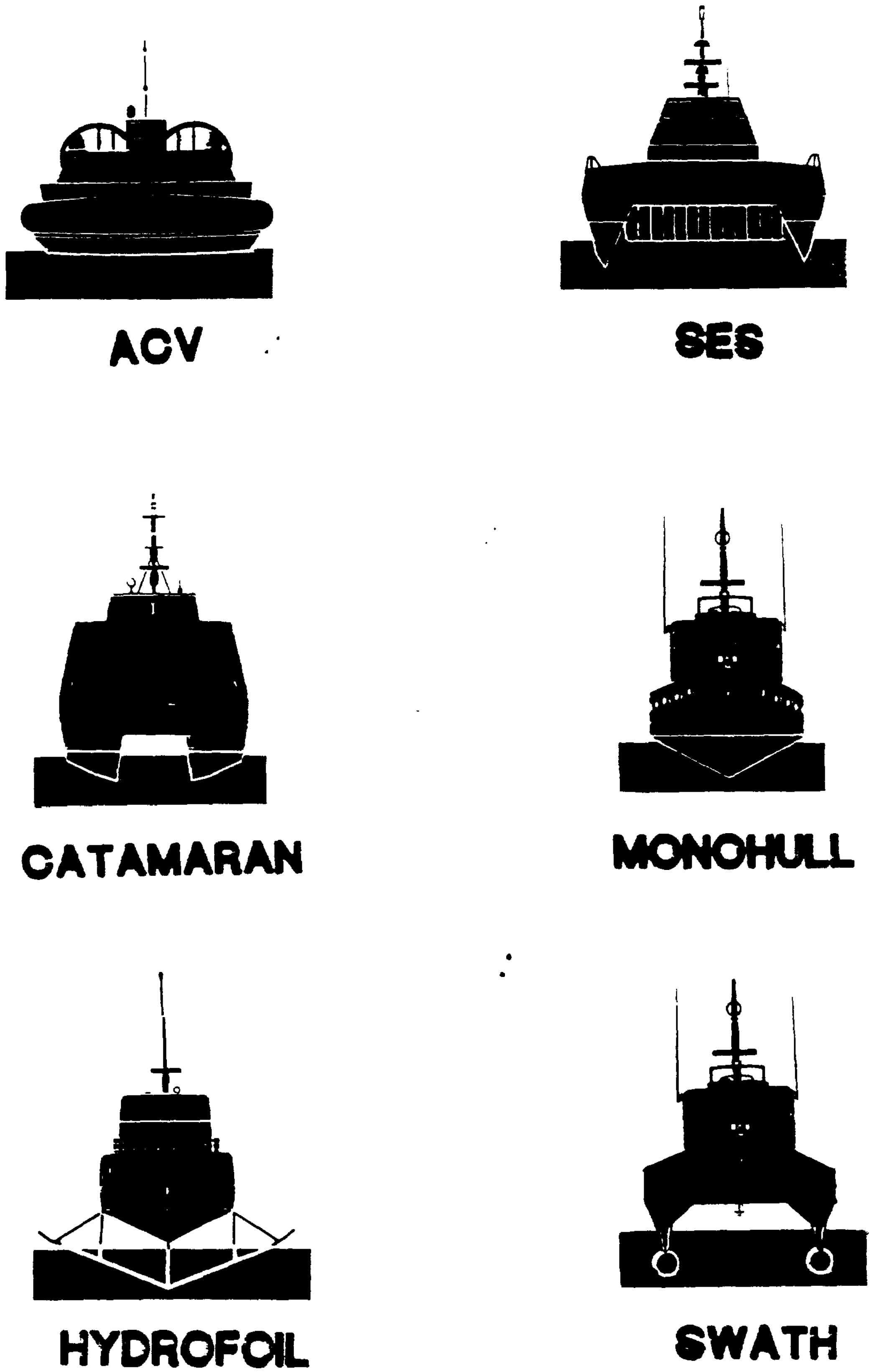


Fig. 1

ATTAINABLE PROPULSIVE EFFIENCY (1967)

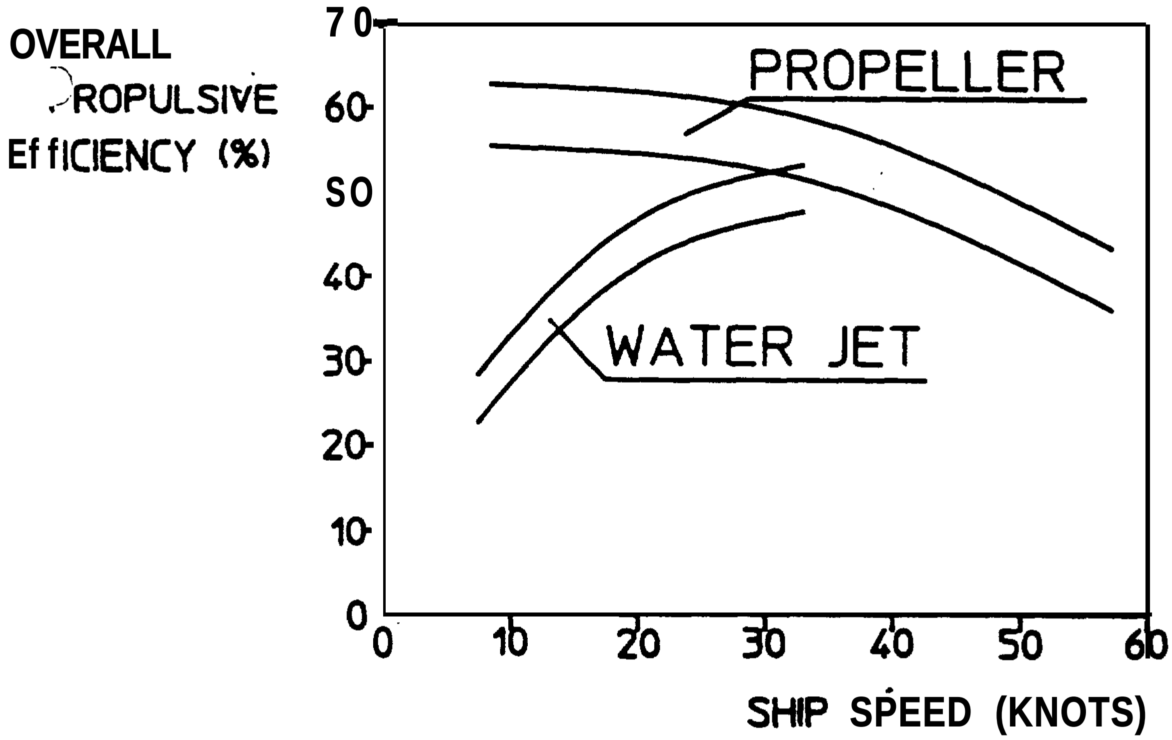
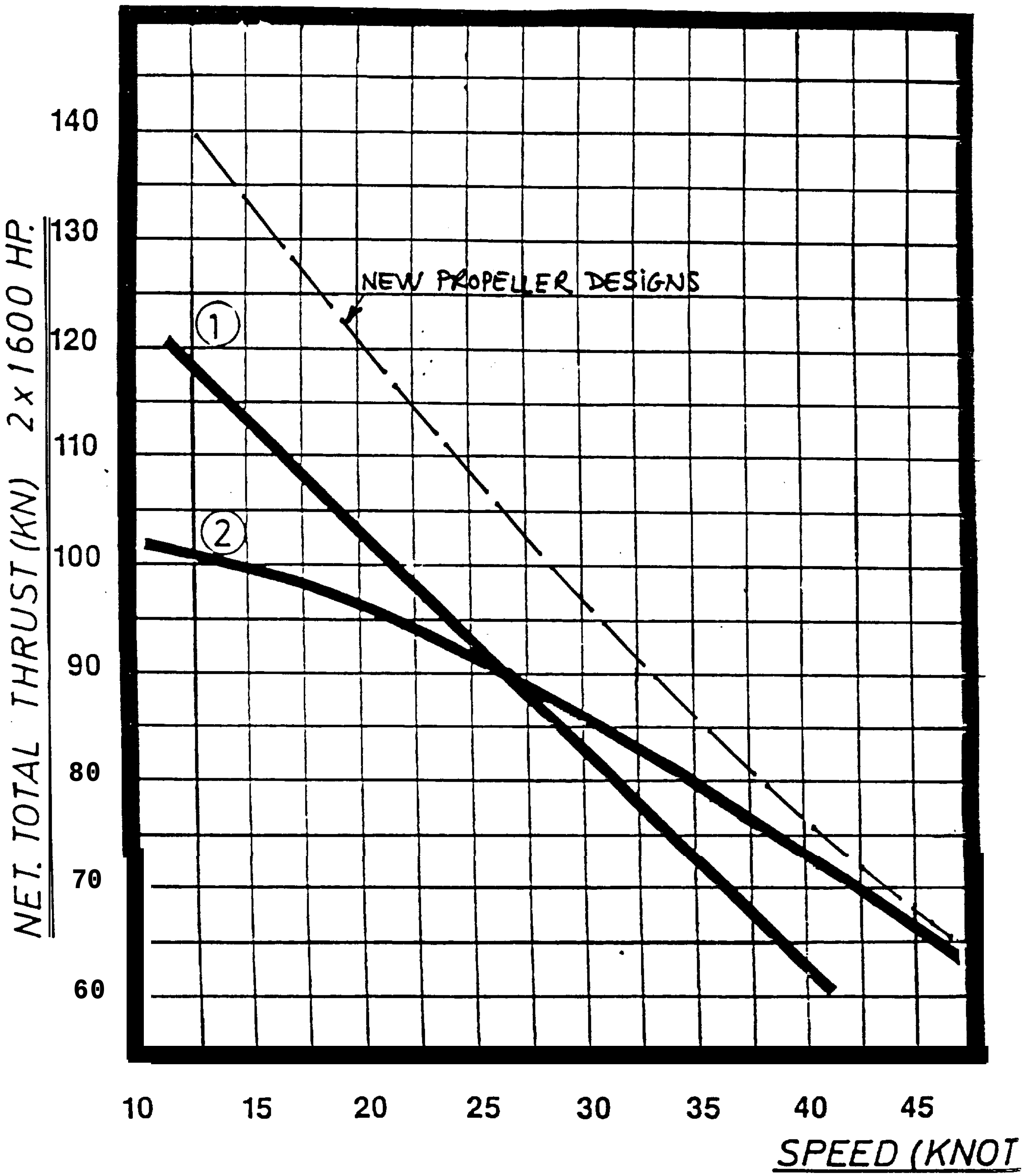


Fig. 2



1. Conventional propeller arrangement.

2. Waterjet.

Fig.3

TYPICAL SPEED - RESISTANCE

CURVES (L = 50r I)

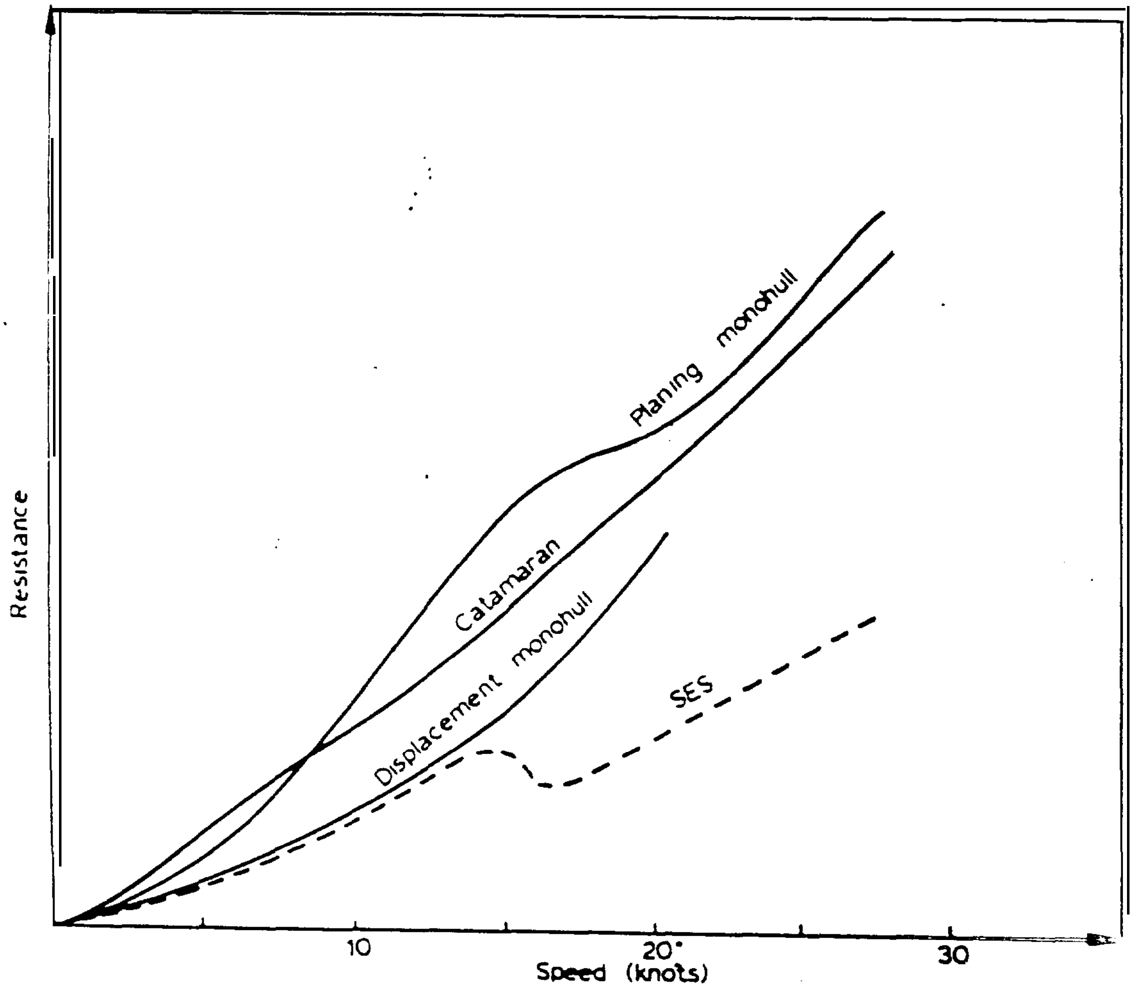


Fig.4

ENGINE POWER / SPEED.

925 m2 - 0.2 ton/m2

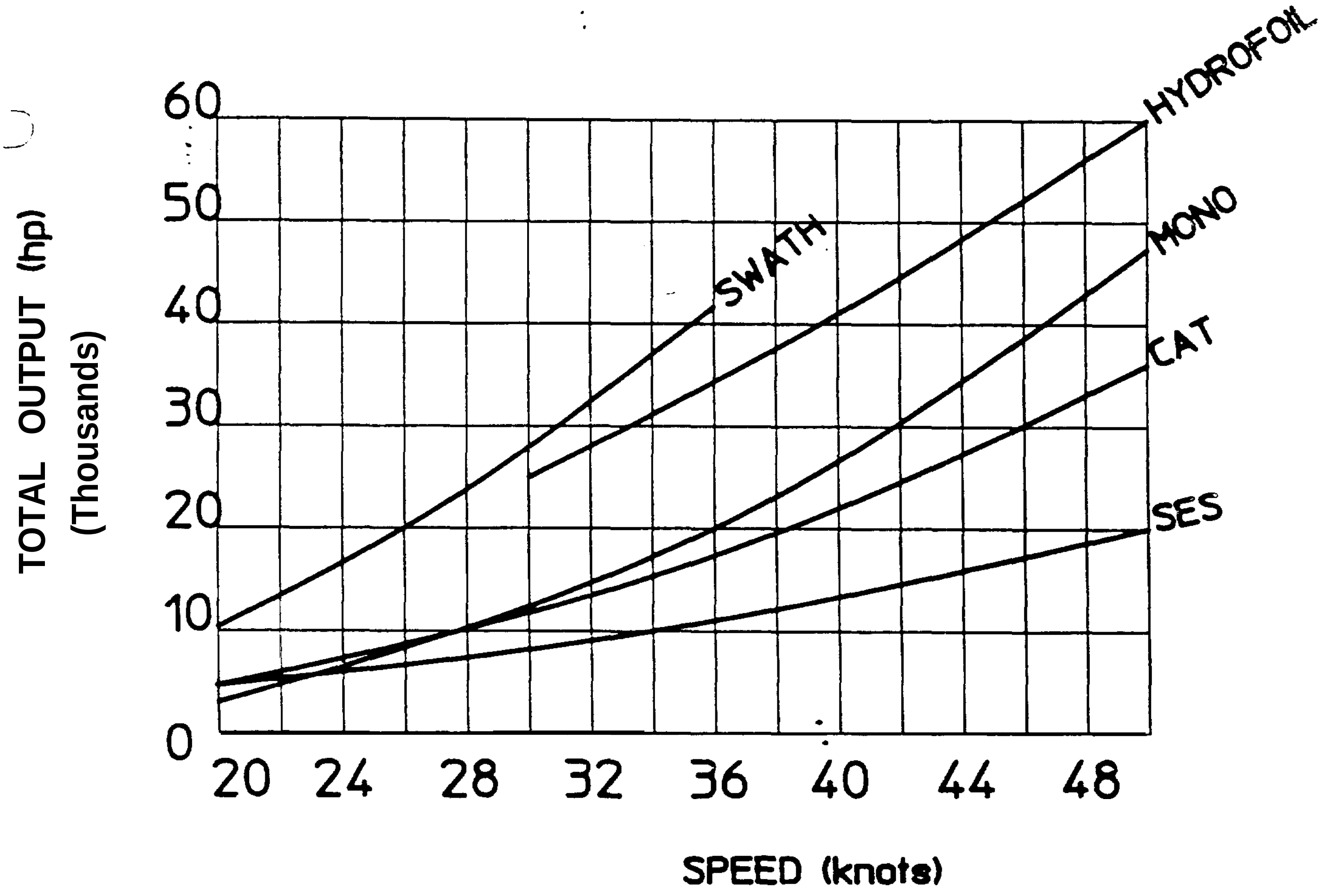


Fig.5

Weight distribution of a typical passenger catamaran:

L = 42 m, velocity = 36 knots.

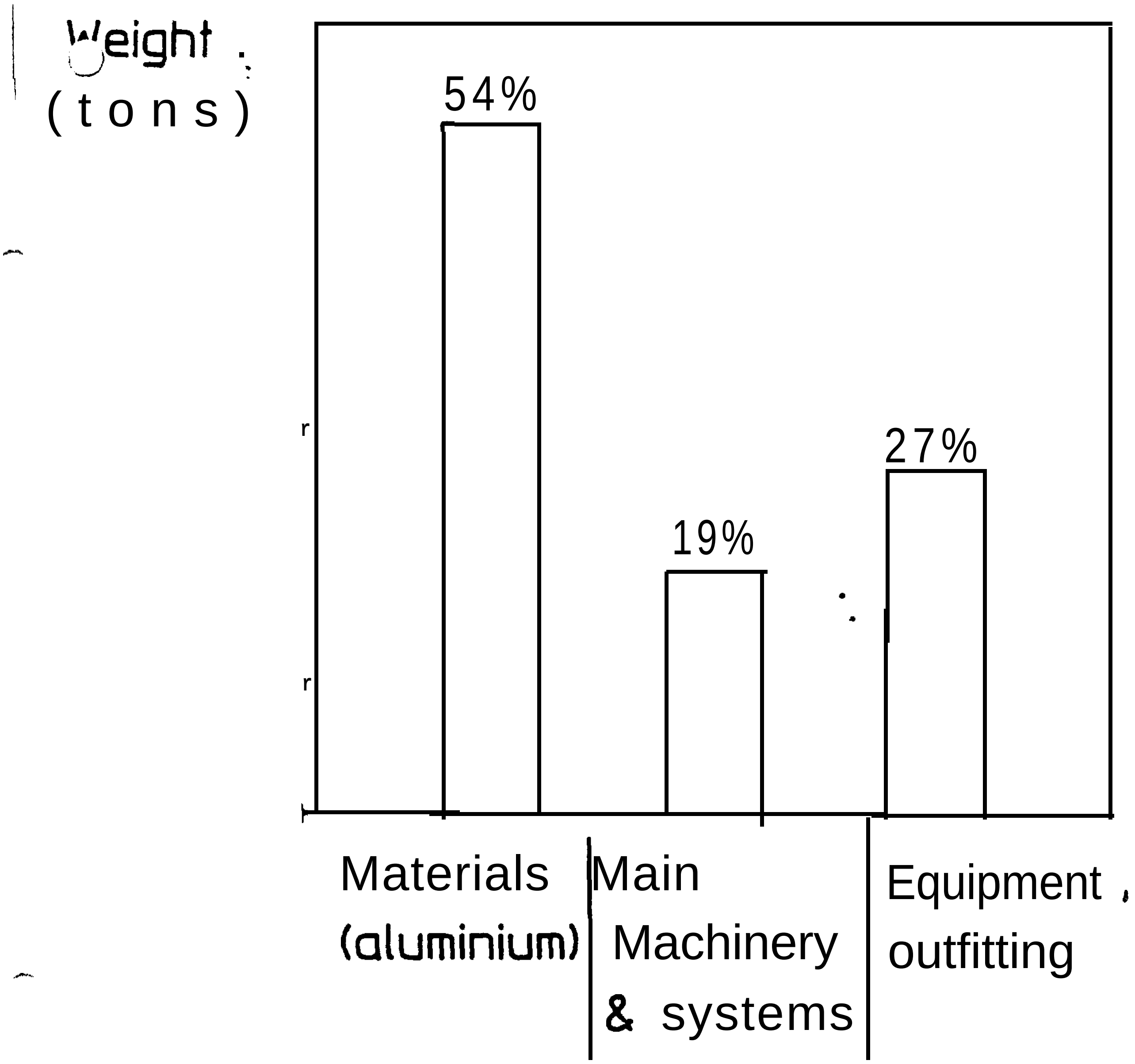


Fig.6

Typical hull weight distribution of a medium size, fast craft.

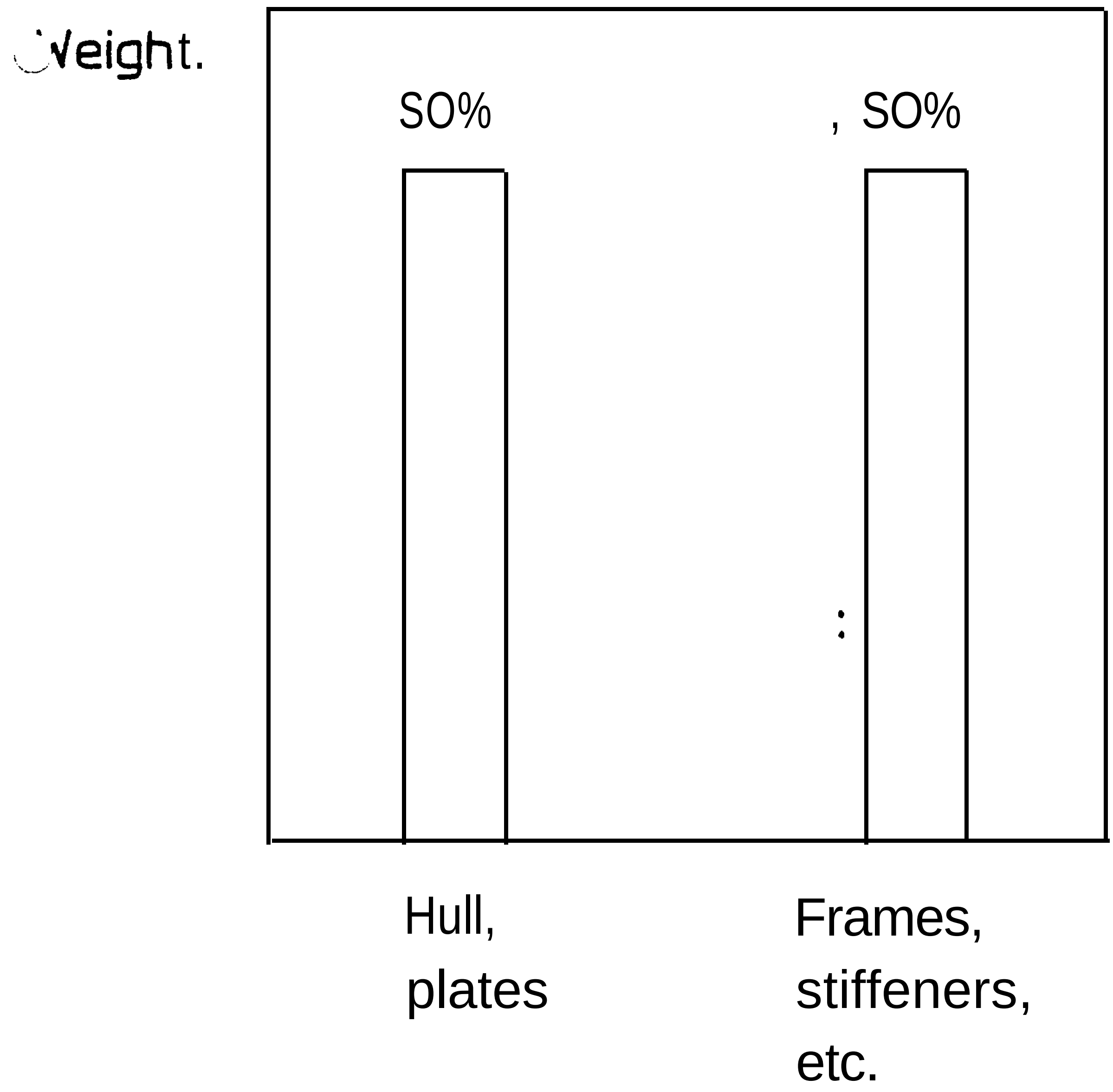


Fig.7

TYPICAL DISTRIBUTION OF HULL WEIGHT FOR DIFFERENT MATERIALS.

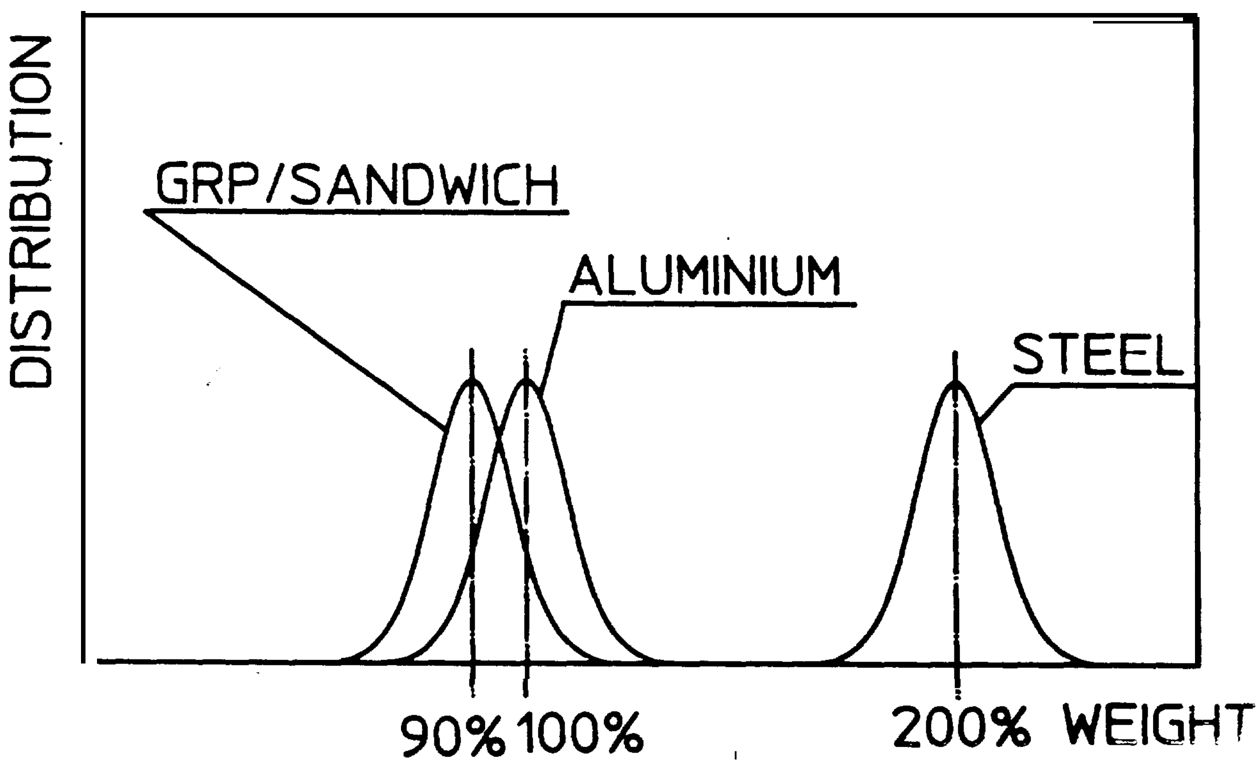
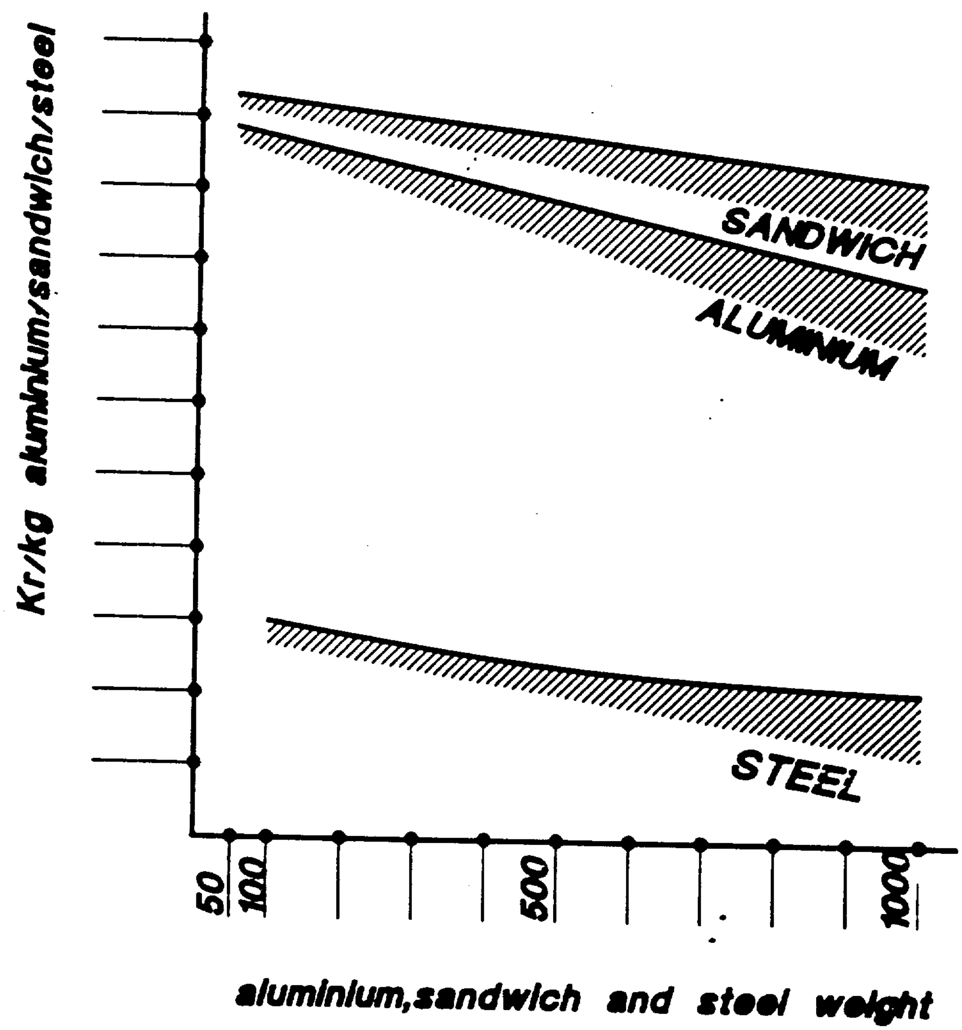


Fig.8



Kr/kg aluminium/sandwich/steel
SANDWICH
ALUMINIUM
STEEL
50
100
500
1000
aluminium,sandwich and steel weight

Fig. 9

Price distribution for a 50 m long monohull car ferry, GRP/sandwich or aluminium hull.

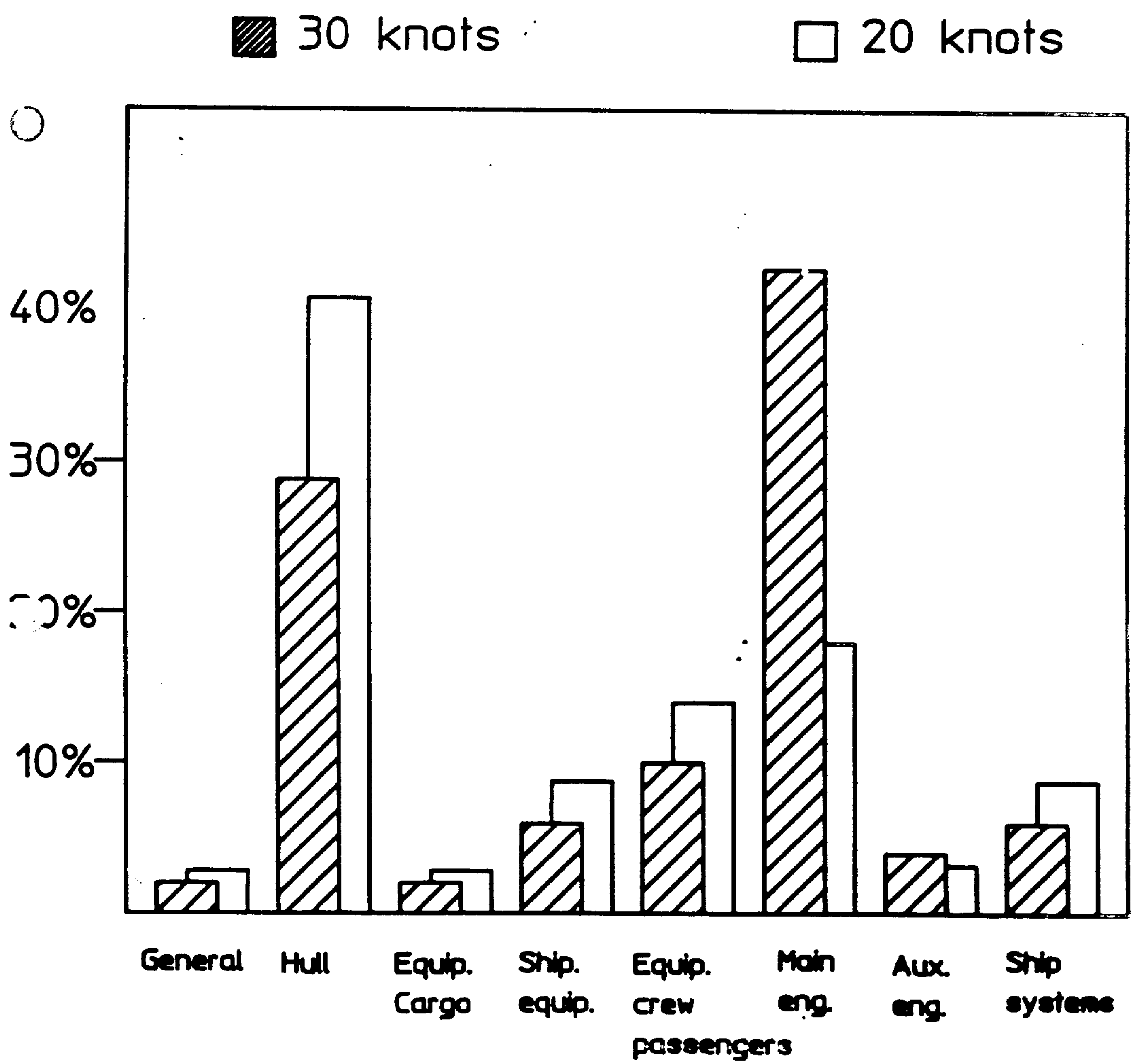


Fig. 10

BUILDING COSTS OF DIFFERENT TYPES OF ADVANCED MARINE VEHICLES AS A FUNCTION OF THE SHIP SPEED.

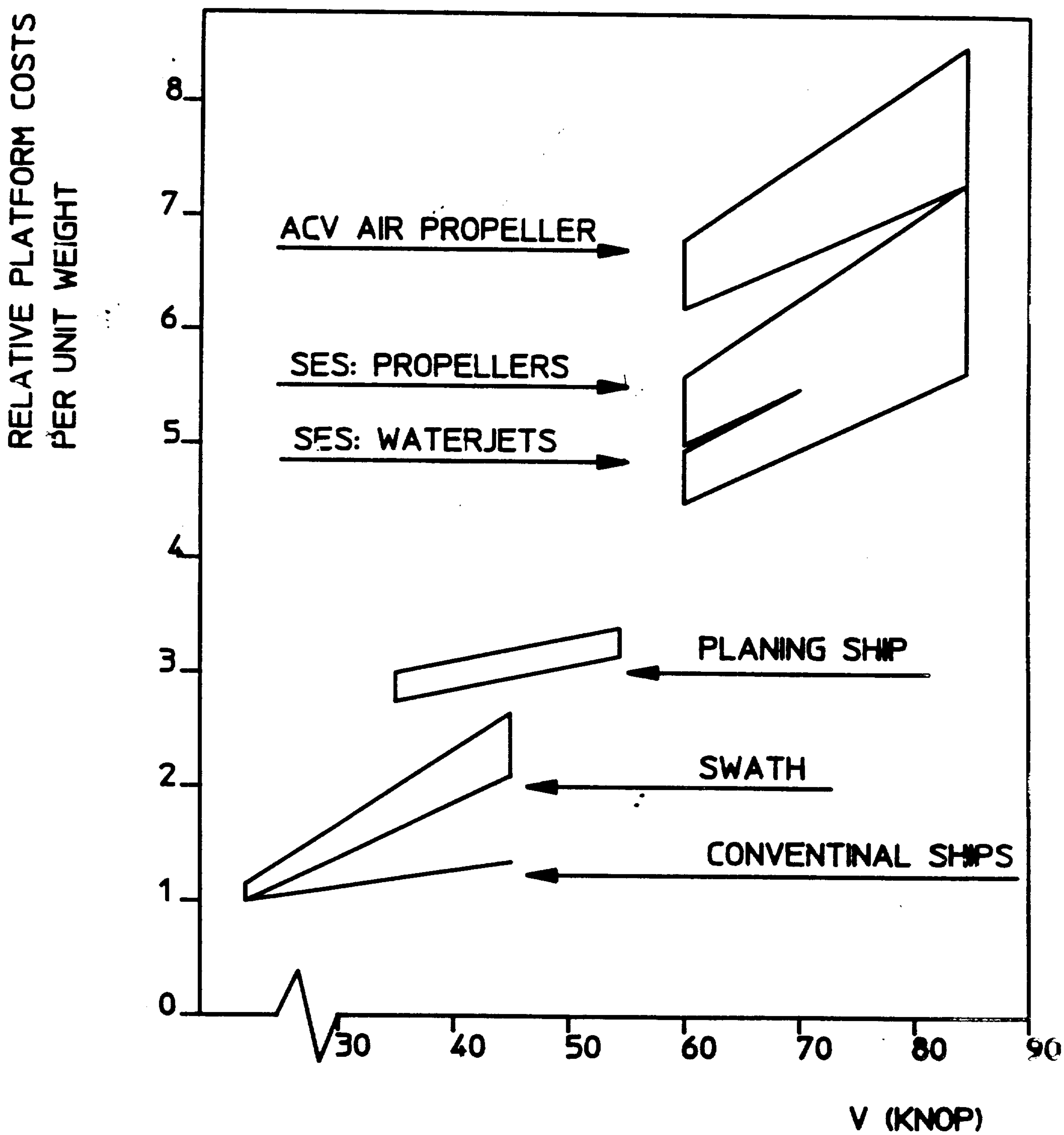


Fig.11

CONFERENCE ON HIGH-SPEED MARINE CRAFT
A - 6 MAY 1988 - KRISTIANSAND - NORDWAY

FAST METHOD FOR DETERMINING AND CHECKING THE MAIN DIMENSION's OF A WATERJET

By:

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R. Admiral (R) Giovanni VENTURINI - CRM Design Office - Milan - ITALY

DESIGN DEPARTMENT MANAGEMENT

FAST METHOD FOR DETERMINING AND CHECKING THE MAIN DIMENSIONS OF A WATERJET

1 - FOREWORD

The user may find it helpful to be able to estimate rapidly and within a sufficient degree of approximation the characteristics required of a waterjet for a specific hull. It is particularly useful to make a summary assessment, during the design stage, with the possibility of direct coupling or by means of a gearbox, to the engine selected for the actual propulsion.

With the cooperation of RUNAVI of Genoa, the Design Department of CRM has therefore drawn a series of curves referred to the unit of power. These make it fairly easy to identify the main design parameters such as capacity, head, r.p.m. and nozzle area, and also to evaluate the efficiency and thrust variations per unit of power which occur when a specific waterjet deviates from optimum conditions.

The diagrams refer to a series of pre-established basic values characteristic of design practices, such as pump efficiency, input efficiency and limit layer recovery.

The relations given in the appendix do however, allow corrections to be made to the values obtained if other parameters are to be taken into consideration.
Of course the diagrams give approximate results, and are of use especially for not very demanding craft, almost always driven by fast diesel engines not exceeding 1000 to 1500 Kw, such as CRM's 18D/SS BR series.

2 - THE DIAGRAMS

The diagrams were obtained using the relations indicated in the appendix under point 2, and are the following:

FIGURE 1

Diagram of T/P, in KN/KW and of Ao/P, in m²/KW, as a function of Q/P, in m³/s.KW, in static conditions.

FIGURE 2

Diagram of Tu/P, in KN/KW, as a function of Q/P, in m³/s.KW, at a constant speed of the craft as a parameter.

FIGURE 3

Diagram of the overall efficiencies η_t, as a function of Q/P, at a constant speed of the craft as a parameter, and of the values of $X_r = V_u/V_a$, again as a function of Q/P.

FIGURE 4

Diagram of the maximum values of $T_u/\underline{P}$ and of the optimum values of $Q/\underline{P}$ in relation to the speed of the craft.

FIGURE 5

Diagram of the values of $A_u/\underline{P}$ in dynamic conditions, as a function of $Q/\underline{P}$.

FIGURE 6

Diagram of the pump head, h_T, in m of H_2O, as a function of $Q/\underline{P}$.

FIGURE 7

Diagram of the value of $n\sqrt{P}$ permissible for the pump, as a function of Q/P, and with the speed of the craft as a parameter.

The curves are obtained by equalling the net head available to that required of the pump.

FIGURE 8

Diagram of the value of the characteristic number attributable to the pump taking cavitation into account, as a function of Q/P and with the speed of the craft as a parameter.

FIGURE 9

Diagram of T_u, as a function of T_u/A_u, with the speed of the craft as a parameter.

FIGURE 10

Net head available on the pump h_{sv}, in m of H_2O, as a function of the speed of the craft.

FIGURE 11

Diagram of the values of QX/PX / QO/PQ Q_x/Q_o, P_x/P_o as a function of h_{svx}/h_{svo}.

The subscript o indicates the design condition and h_{svo} the net head required of the pump at the design point.

3 - USE OF THE DIAGRAMS

3.1 We note that the static thrust T is the static thrust of the waterjet, while the working thrust T_u is equal to the resistance of the craft, at the speed V, without appendages.

The diagrams make it possible to solve all water jet propulsion problems very easily.

It can be seen immediately from figure 3 that the maximum value of the overall efficiency of the hydrojet referred to the resistance of the bare hull, η_T = 0.51, is independent of the speed but is obtained, for the various speeds, with very different Q/P values and which increase as the speed itself drops.

The value of 0.51 refers to the values taken as a basis for the calculation, and which may be considered as mean values for well designed waterjets.

The relations shown indicate that efficiency η_T increases as the pump efficiency η_p and input efficiency η_{imb} increase and as the external loss factor B_1 decreases.

The value of the characteristic number of the intake S = 0.65 refers to very well made pumps, and is rarely achieved in commercial waterjets. The same may be said for the efficiency $\eta_p = 0.88$.

The relations under point 3 of the appendix make it possible to calculate how η_T, $T/_p$, A_u/p and $n\sqrt{P}$ vat-y as η_p, η_{imb}, B_1 and χ_2 vary.

3.2 - The most urgent problem is that of calculating the parameters of the waterjet for a craft whose bare hull resistance at a speed V_o is R_o, while at a speed $V_x < V_o$ it is R_x, so that $R_x/R_o = (V_x/V_o)^m$, where m is considerably less than 2 (planing or semi-planing hulls or operation with a reduced number of propellers).

The optimum value of Q/P and the maximum value of $T_u/P = R/P$ are found in figure 4, from which $P_o = \frac{R_o}{R/P}$ and $Q_o = Q/P.P_o$ are obtained.

If it is felt that the value found for Q is too high and would lead to an excessively bulky water-jet, it is possible to see in figure 3, for the velocity curve V_o, what value of Q/P allows a performance within acceptable limits: then the relevant value of T_u/P is obtained from figure 2. h_{To} and $A_{,}$, are obtained from figures 5 and 6 respectively, while $A_{1o} = Q_o/0.7V_o$.

It should be pointed out that the value found for P_o is the power absorbed by the pump; to find the power of the engine P_B it is necessary to divide by the drive efficiency, which for a waterjet is generally 0.98. From figure 9, in which the curve referred to the speed V_x is entered with T_u/A_u, the T_u/P_x is obtained which makes it possible to get $P_x = \frac{R_x}{T_u/P_x}$.

From figure 2, in addition, Q_x/P_x is obtained, and thus Qx and consequently h_{Tx}.

From figure 11 the value of h_{svx}/h_{svo} for P_x/P_o is obtained which makes it possible, h_{svx} being equal to the value available at the speed V_x, to find h_{svo}, representing the net head for which the pump may be designed to require h_{svx} at the speed V_x with the working thrust of T_{ux}.

This value may be lower than, equal to or greater than that available at the speed V_o given by figure 10.

If it is lower. this means that the pump should be designed less fast than is permitted by the speed V_0: therefore with Q_x/P_x figure 7 gives, for the speed V_x :

$$n_x = n_x \frac{\sqrt[1]{P_x}}{\sqrt{P_x}}$$

and therefore :

$$n_o = n_x \left(\frac{P_o}{P_x}\right)^{1/3}$$

If 15 is equal, this means that the pump may be designed with reference to the speed V_o for the net head, and therefore from figure 7 entering with Q_o/P_o for V_o.

If it is greater, this means that the pump has to be designed again with reference to the speed V_o with Q_o/P_o, but at the speed V_x a net head lower than that available Will be required.

3.3 - In special cases, the waterjet may be designed to give a specific static thrust, after which the speed reached by the boat at constant power is determined.

In this case, the value of T_o/P_o leading to a waterjet of acceptable dimensions is selected in figure 1. The same figure gives the value of A_{uo}/P_o; entering figure 5 with this value different value5 of Q/P are obtained for the various speeds. and from figure 4 the corresponding T_u/P_o values. It is then possible to draw the curve of the working thrust as a function of the speed at the constant power P_o. This curve has to be compared with the bare hull resistance curve of the craft in order to obtain the achievable speed. Note that the T_u/P_o values may be obtained directly also from figure 9.

The design revolutions n_o are given by figure 7 . entering with Q_o/P_o for 'the speed V_o = 0.

3.4 - If we know the capacity Q_o and head h_{TO} of a waterjet at a number of revolutions n_o, within the limits of the coefficients of the diagrams the power P_o is found from the curve in figure 6.

In order to apply this waterjet for a speed V_o. it is necessary to check - using figure 7 - that n_o is lower than the value found entering the curve referred to V_o with Q_o/P_o. The working thrust obtained will be given by figure 2. entering, as usual, with Q_o/Q_o, while the efficiency of the jet will be given by figure 3, and the value of A_u is given by figure 5.

Figure 3 will show whether it is more expedient to work with a greater capacity and a smaller head or vice versa at constant power.

In this case, figure 5 gives the new value of A_u/P_o, while figure 2 will give T_u/P_o. If Q/i', is greater than Q_o/P_o, it is necessary to check n_o

again using figure 7.

3.5 - Given a waterjet with Q_o, h_{To}, T_{uo}, n_o, V_o, A_{uo}, it is possible to use the diagrams to draw the curves of the working thrusts as a function of V and the cavitation limit curve, and, given R = R(V), the power curve as a function of n and V.
The net head required by the pump at n_o h_{*vo} revolutions is obtained from figures 7 and 10.
If this is lower than the net head available at the speed V,, given again by figure 10, the curve of the thurst at constant power P_o may be maintained at least up to the speeii t0 which h_{*vo} corresponds.

in effect, taking into account the drop in capacity as the speed decreases at a constant number of revolutions, it may be maintained up Lo a lower speed, that is to say up to the speed which, entering the diagram in figure 5 with A_{uo}/P_o, corresponds to Q_x/P_o = Q_o/P_o · $(h_{*vx}/h_{*vo})^{x}$.
A thrust curve at the power P_x is obtained immediately from figure 5, entering, for the various speeds, with A_u/P_x and reading the Q/P values and the corresponding T_u/P in figure 2. The curve may be drawn up to a value of Q_x/P_x = Q_o/P_o · h_{*vo}/h_{*vx}.
The curve of the power absorbed along the R = R(V) curve

is obtained for each R as described under point 3.2. The number of revolutions is found with $n_x/n_o = (P_x/P_o)^{1/3}$.

4 - **EXRMPLE**

4. 1- As an example of the use of the diagrams, let us consider a semidisplacement of about 140 t., with the following resistance curve:

V = 15 knots = 7.716 m/s 45.1 KN

V = 20 knots = 10.288 m/s 86.7 KN

V = 25 knots = 12.86 m/s 111.35 KN

At a speed of 15 knots, operating with a force 4 sea adds another 15 KN to the resistance to headway. The propulsion equipment must consist of 3 water-lets. Operation at 15 knots in force 4 seas must be sustained by two waterjets.

Sizing of the waterjet must thus be done for :

37. 12 KN at 25 knots

30 KN at 15 knots

In this case the resistance follows the law R_x/R_o = $(15/25)^{0.42}$.

From figure 4 the optimum values of Q/P = 0.0044 and T_u/P = 0.0398 are obtained, thus :

P_o = 933 KW Q_o = 4.10 m^3/s.

Considering the capacity excessive, from figure 3 it is seen that, losing one point only on efficiency, Q/P

0.0034 and T_u = 0.0392 may be achieved, and consequently P_o = 947 KW, Q_o = 3.22 m^3/s, h_{To} = 25.8 m, X_{ro} = 1.875, a_{uo} = 0 . 1335 m^2.

The following are obtained from figure 9: P_x = 508 KW, Q_x/P_x = $5.04/10^3$, Q_x = 2.56 m^3/s, h_{Tx} = 17.3 m, X_r = 2.46.

From figure 11 the following is obtained :

$$h_{svx}/h_{svo} = 0.66$$

entering with P_x/P_o = 0 . 5 3 6 .

h_{svx} being 1 3 . 7 m, it follows that H_{sv} = 17.7 m, greater than h_{sv} = 14.5 m referred to a speed V_o = 25 knots.

The pump may therefore designed for a maximum number of r-evolutions obtained entering figure 7 with σ_o = 0.0034 and V_o = 25 knots.

n_o = 1 5 . 2 7 r.p.s. is obtained, equal to 916 r.p.m.

The waterjet designed in this way will be capable of developing the static power given by figure 11 for h_{svx}/h_{svo} = 8.8/14.5 = 0.607. that is to say P_x/P_o = 0.475, P_x = 450 KW.

Entering figure 1 with A_u/P_x = 0.1335/450 = $29.66/10^5$ m^2/KW the follwing is obtained :

Q/P = 5.04 $\qquad$ Q = 2.27 m^3/s

T/p = $8.95/10^2$ $\qquad$ T = 40.28 KN

5 - CONCLUSIONS

The graphic method referred to above makes it possible to solve rapidly all the problems related to waterjets in average design coniiitrons.

The relations qiven in point 3 of the appendix allow the values found to be corrected rapidly if different parameters are used.

This method is of use for a preliminary assessment of the application of a waterjet to a specific propulsion problem or of the performances achievaperformance ob-tainable from a waterjet already installed.

It is understood that the development of a project en-tails a far greater commitment, which remains the province of the manufacturers.

6 - ACKNOWLEDGEMENTS

The authors wish Lo thank CRM Milan and RUNAVI Genoa for placing at their disposal. the ini'ormat ion and data processed on the basis of their application to waterjets coupled with CRM 18D/SS BR series engines on fast boats.

SYMBOLS

A_1	=	intake area	m^2
A_μ	=	nozzle area	m^2
B_1	=	external loss constant	
C_D	=	intake resistance factor	
h_{sv}	=	net head on intake	m of H_2O
h_T	=	total pump head	
K	=	chafacteristic c number	
n	=	pump revolutions	r.p.s.
N_s	=	number of stages in series	
P	=	power absorbed by the pump	KW
P_s	=	power of engine	KW
Q	=	capacity of pump	m^3/s
R	=	bare hull resistance	KN
S	=	characteristic number- at intake	
T	=	thrust	KN
T_μ	=	working thrust of Jet	KN
V_u	=	output speed of jet	m/s
V_o	=	speed of craft	=
x_r	=	V_u/V_o	
η_{imb}	=	input efficiency	
η_p	=	pump efficiency	
η_T	=	overall efficiency of waterjet	
χ_1	=	motion quantity factor	
χ_2	=	kinetic energy factor	
φ_μ	=	nozzle loss factor	
δ	=	density	Kg/m^3

REFERENCES

For development and a discussion of the relations given in the appendix, reference may be made to :

G. Venturini, CRM Design Department - Technical report on hydrojet propulsion

APPENDIX

The relations on which the curves in the diagrams are based are the following :

$$\frac{T_u}{P} = \frac{\rho\, Q}{10^3\, P}\left[\left(\frac{2\cdot\eta_P}{1{,}025\,\frac{Q}{P}\,(1+\psi_u)} + \frac{\alpha_2\cdot\eta_{int}\cdot V_0^2}{1+\psi_u}\right)^{\frac{1}{2}} - B_1\cdot V_0\right] \qquad (1)$$

$$h_T := \frac{\eta_P}{1{,}025\cdot g\,\frac{Q}{P}}\,, \qquad X_2 = \frac{V_u}{V_0} = \left(\frac{2\,\eta_P}{1{,}025\,\frac{Q}{P}\,(1+\psi_u)} + \frac{\alpha_2\,\eta_{int}\cdot V_0^2}{1+\psi_u}\right)^{\frac{1}{2}}\frac{1}{V_0} \qquad (2)$$

$$\frac{A_u}{P} = \frac{\frac{Q}{P}}{\left(\frac{2\,\eta_P}{1{,}025\cdot\frac{Q}{P}\,(1+\psi_u)} + \frac{\alpha_2\,\eta_{int}\cdot V_0^2}{1+\psi_u}\right)^{\frac{1}{2}}} \qquad (3)$$

$$h_{sv} = 10{,}1 + \alpha_2\cdot\eta_{int}\,\frac{V_0^2}{2g} \qquad (4)$$

$$n\sqrt{P} = \frac{(g\,h_{sv})^{\frac{3}{4}}\cdot S}{\sqrt{\frac{Q}{P}}} \qquad (5)$$

$$\eta_T = \frac{T_u\cdot V_0}{P} \qquad (6)$$

$$\text{con}\quad B_1 = \alpha_1 + \frac{C_D}{2\,X_i} \qquad (7)$$

$$K = \frac{N_s^{\frac{3}{4}}\cdot S\,(h_{sv})^{\frac{3}{4}}}{\left(\frac{\eta_P}{1{,}025\,g\,\frac{Q}{P}}\right)^{\frac{3}{4}}} \qquad (8)$$

$$\frac{q_i}{P} = \frac{Q}{\underline{P}\; X_i\, V_0} \qquad (9)$$

$$\frac{\dfrac{Q_x}{P_x}}{\dfrac{Q_0}{P_0}} \doteq \left(\frac{P_x}{P_0}\right)^{-\frac{2}{3}} = \frac{h_{svo}}{h_{svx}} = \frac{n_o^{\,2}}{n_x^{\,2}} \qquad (10)$$

$$\frac{h_{svx}}{h_{svo}} = \left(\frac{P_x}{P_0}\right)^{\frac{2}{3}} = \left(\frac{Q_x}{Q_0}\right)^{2} = \left(\frac{n_x}{n_0}\right)^{2} \qquad (11)$$

$$\frac{P_x}{P_0} = \left(\frac{Q_x}{Q_0}\right)^{3} = \left(\frac{h_{svx}}{h_{svo}}\right)^{\frac{3}{2}} = \left(\frac{n_x}{n_0}\right) \qquad (12)$$

In static conditions the value of T/P was found with :

$$\frac{T}{P} = \frac{\rho Q}{10^3 P} \left(\frac{2\,\eta_p}{1{,}025\,\frac{Q}{P}\,(1+\varphi_a)} - \frac{\Sigma\varphi_i\;Q^2/P^2}{\frac{A_i^2}{P^2}} \right)^{\frac{1}{2}} \qquad (13)$$

approximated in :

$$\frac{T}{P} = 1{,}025 \left(\frac{Q}{P} \cdot \frac{2\,\eta_p}{1{,}175} \right)^{\frac{1}{2}} = 1{,}255 \left(\frac{Q}{P} \right)^{\frac{1}{2}}$$

2 - The calculations for the curve!, were done using the following mean values :

η_p	=	0.88
η_{imb}	=	0.59
χ_z	=	0.9
B_1	=	1
$1+\varphi_a$	=	1.02
$\varepsilon\varphi_i$	=	0.7
$\varkappa_i$	=	0.7
S	=	0.45 .

The following relations are thus obtained:

$$\frac{T_M}{P} = 1{,}025\,\frac{Q}{P} \left[\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521\,V_0^2 \right)^{\frac{1}{2}} - V_0 \right] \qquad (1')$$

$$h_T = \frac{0.0875}{\frac{Q}{P}} , X_z = \left(\frac{1,683}{\frac{Q}{P}} + 0,521 \; V_0^2 \right)^{\frac{1}{2}} \frac{1}{V_0} \qquad (2')$$

$$\frac{A_u}{P} = \frac{\frac{Q}{P}}{\left(\frac{1,683}{\frac{Q}{P}} + 0,521 \; V_0^2 \right)^{\frac{1}{2}}} \qquad (3')$$

$$h_{sv} = 10,1 + 0,0266 \; V_0^2 \qquad (4')$$

$$n \sqrt{P} = \frac{3,6 \; h_{sv}^{\frac{3}{4}}}{\sqrt{\frac{Q}{P}}} \qquad (5')$$

$$\eta_T = 1,025 \; \frac{Q}{P} V_0 \left[\left(\frac{1,683}{\frac{Q}{P}} + 0,521 \; V_0^2 \right)^{\frac{1}{2}} - V_0 \right] \qquad (6')$$

$$K = 4,04 \, . \, N_s^{\frac{3}{4}} \; h_{sv}^{\frac{3}{4}} \left(\frac{Q}{P} \right)^{\frac{3}{4}} \qquad (8')$$

$$\frac{T}{P} = 1,255 \left(\frac{Q}{P} \right)^{\frac{1}{2}} \qquad (13)$$

$$\frac{A_u}{P} = 0,817 \left(\frac{Q}{P} \right)^{\frac{3}{2}} \qquad (14)$$

(found by equating $\frac{\rho Q^2}{10^3 \frac{A_u P^2}{P}} = 1{,}255 \left(\frac{Q}{P}\right)^{\frac{1}{2}}$

3 - The expressions which make it possible to calculate the variation of T_u/P, η_T, A_u/P for a variation in the parameters by which they are influenced are :

$$\frac{\Delta \frac{T_u}{Q}}{\Delta \eta_P} = \frac{1}{2} \cdot \frac{1{,}951}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521\, V_0^2\right)^{\frac{1}{2}}} \tag{15}$$

$$\frac{\Delta \eta_T}{\Delta \eta_P} = \frac{1}{2} \cdot \frac{V_0 \cdot 1{,}951}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521\, V_0^2\right)^{\frac{1}{2}}} \tag{16}$$

$$\frac{\Delta \ell_T}{\Delta \eta_P} = \frac{1}{1{,}025 . g . \frac{Q}{P}} \quad , \quad \frac{\Delta x_r}{\Delta \eta_P} = \frac{1}{\frac{Q}{P} . 2 . V_0} \cdot \frac{1{,}904}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521\, V_0^2\right)^{\frac{1}{2}}} \tag{17}$$

$$\frac{\Delta \frac{A_u}{P}}{\Delta \eta_P} = -\frac{1}{2} \cdot \frac{1{,}904}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521 . V_0^2\right)^{\frac{3}{2}}} \tag{18}$$

$$\frac{\Delta \frac{T_u}{Q}}{\Delta \eta_{ind}} = \frac{1}{2} \cdot \frac{0{,}9 . V_0^2 . \frac{Q}{P}}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521 . V_0^2\right)^{\frac{1}{2}}} \tag{19}$$

$$\frac{\Delta \eta_T}{\Delta \eta_{ind}} = \frac{1}{2} \cdot \frac{0{,}9 \cdot V_0^3 . \frac{Q}{P}}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521\, V_0^2\right)^{\frac{1}{2}}} \tag{20}$$

$$\frac{\Delta \frac{A_u}{P}}{\Delta \eta_{imb}} = -\frac{1}{2} \cdot \frac{0{,}878 \cdot V_0^2 \cdot \frac{Q}{P}}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521\, V_0^2\right)^{\frac{3}{2}}} \qquad (21)$$

$$\frac{\Delta \frac{T_u}{P}}{\Delta B_1} = -\, 1{,}025\, \frac{Q}{P} \cdot V_0 \qquad (22)$$

$$\frac{\Delta \eta_T}{\Delta B_1} = -\, 1{,}025\, \frac{Q}{P}\, V_0^2 \qquad (23)$$

$$\frac{\Delta h_w}{\Delta \eta_{imb}} = 0{,}9 \cdot \frac{V_0^2}{2g} = 0{,}0459\, V_0^2 \qquad (24)$$

$$\frac{\Delta n \sqrt{P}}{\Delta \eta_{imb}} = \frac{3}{4} \cdot \frac{0{,}45\, V_0^2 \cdot S}{\left(g\, 10{,}1 + 0{,}266\, V_0^2\right)^{\frac{1}{4}} \sqrt{\frac{Q}{P}}} \qquad (25)$$

$$\frac{\Delta X_r}{\Delta \eta_{imb}} = \frac{1}{2 V_0} \cdot \frac{0{,}878\, V_0^2}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521\, V_0^2\right)^{\frac{1}{2}}} \qquad (26)$$

The variation of the values seen with a variation of χ_2 is obtained using :

$$\chi_1 = \chi_2^{1/2}$$

Giving :

$$\frac{\Delta \frac{T_u}{Q}}{\Delta \chi_2} = \frac{1}{2} \cdot 1{,}025 \cdot \frac{Q}{P} \left[\frac{0{,}576 \cdot V_0^2}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521 \cdot V_0^2 \right)^{\frac{1}{2}}} - \frac{V_0}{0{,}9} \right] \qquad (27)$$

$$\frac{\Delta \eta_T}{\Delta \chi_2} = \frac{1}{2} \cdot 1{,}025 \ \frac{Q}{P} \left[\frac{0{,}576 \ V_0^3}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521 \ V_0^2 \right)^{\frac{1}{2}}} - \frac{V_0^2}{0{,}9} \right] \qquad (28)$$

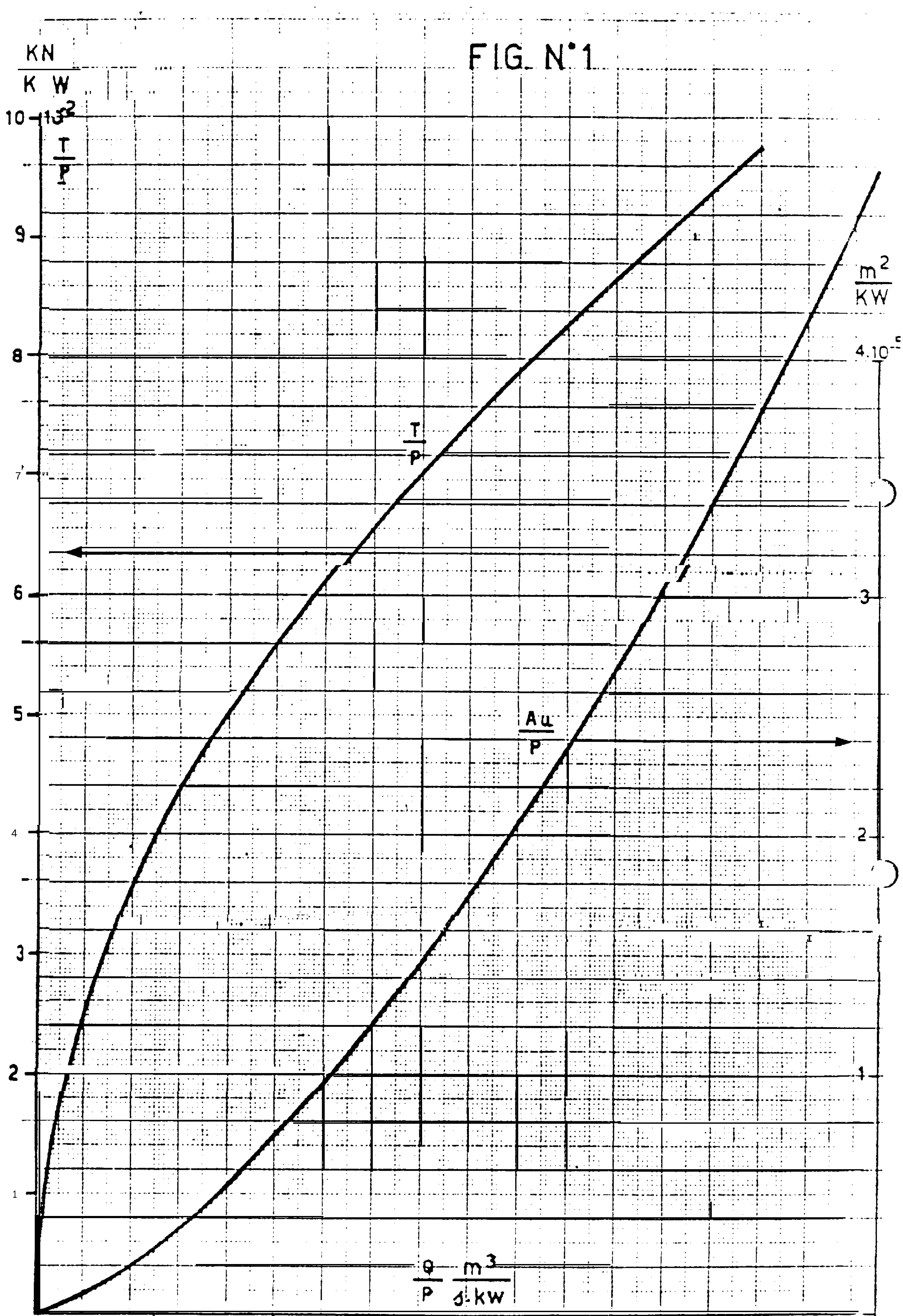
$$\frac{\Delta A_u}{\Delta \chi_2} = - \frac{1}{2} \ \frac{\frac{Q}{P} \cdot 0{,}576 \ V_0^2}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521 \ V_0^2 \right)^{\frac{3}{2}}} \qquad (29)$$

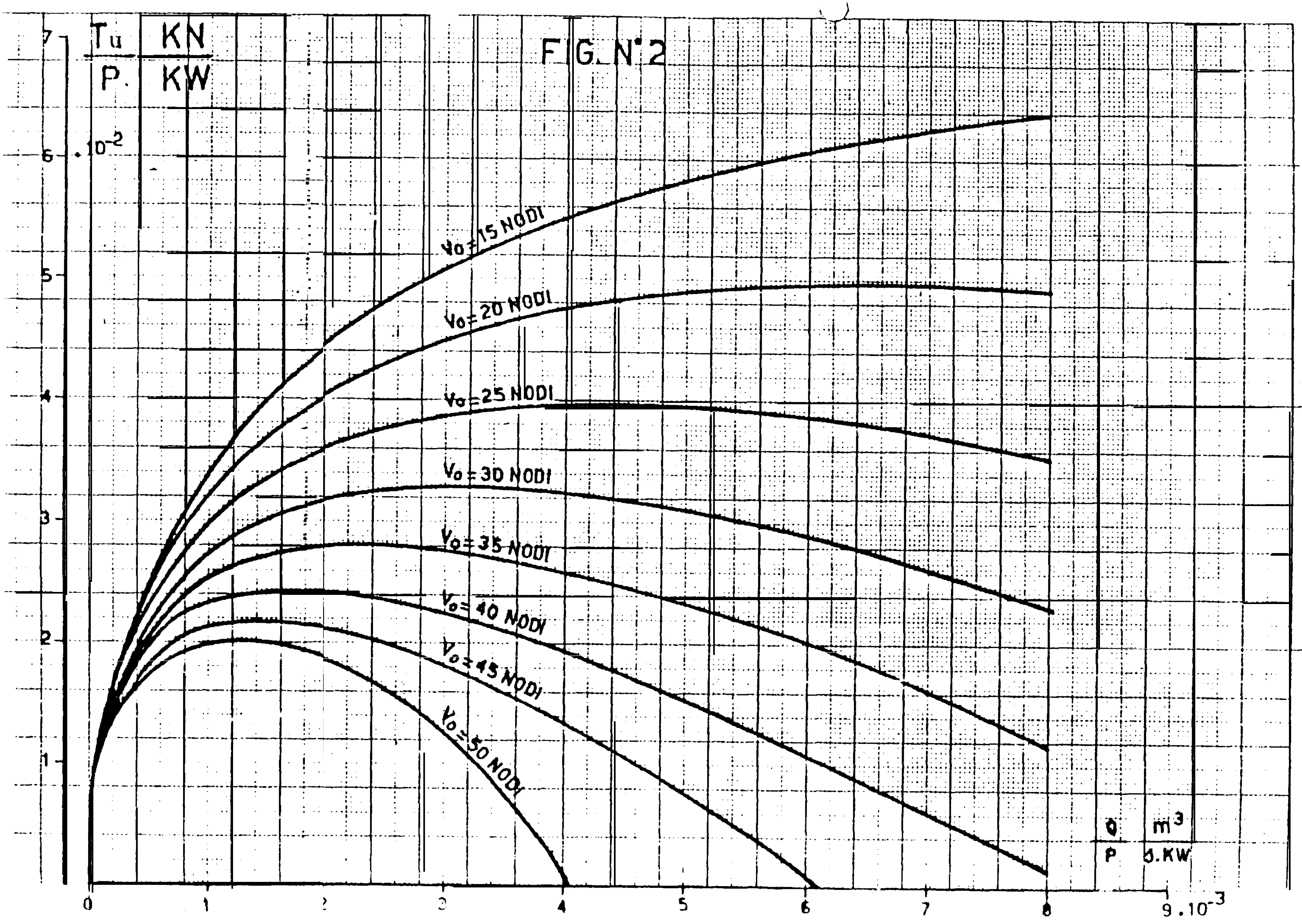
$$\frac{\Delta \ell_{sw}}{\Delta \chi_2} = 0{,}03 \ V_0^2 \qquad (30)$$

$$\frac{\Delta \ n\sqrt{P}}{\Delta \ \chi_2} = \frac{3}{4} \cdot \frac{0{,}295 \ V_0^2 \cdot S}{\left(g \ 10{,}1 + 0{,}266 \ V_0^2 \right)^{\frac{1}{4}} \sqrt{\frac{Q}{P}}} \qquad (31)$$

$$\frac{\Delta \chi_3}{\Delta \chi_2} = \frac{1}{2 V_0} \cdot \frac{0{,}576 \cdot V_0^2}{\left(\frac{1{,}683}{\frac{Q}{P}} + 0{,}521 \ V_0^2 \right)^{\frac{1}{2}}} \qquad (32)$$

FIG. N°1





Tu KN
P. KW
FIG. N°2
.10-2
Vo=15 NODI
Vo=20 NODI
Vo=25 NODI
Vo=30 NODI
Vo=35 NODI
Vo=40 NODI
Vo=45 NODI
Vo=50 NODI
Q m3
P s.KW
9.10-3
0
1
2
3
4
5
6
7
8

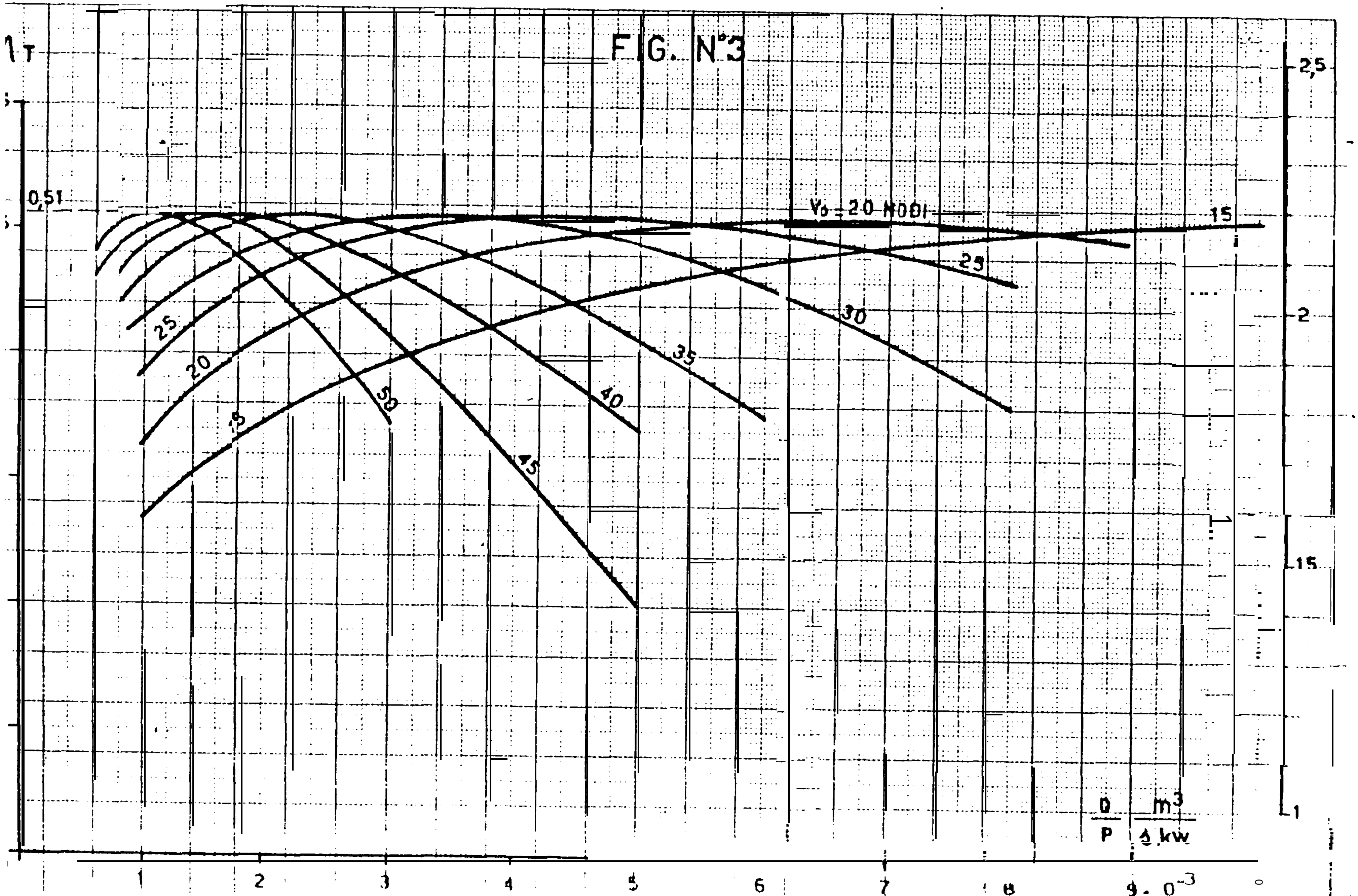
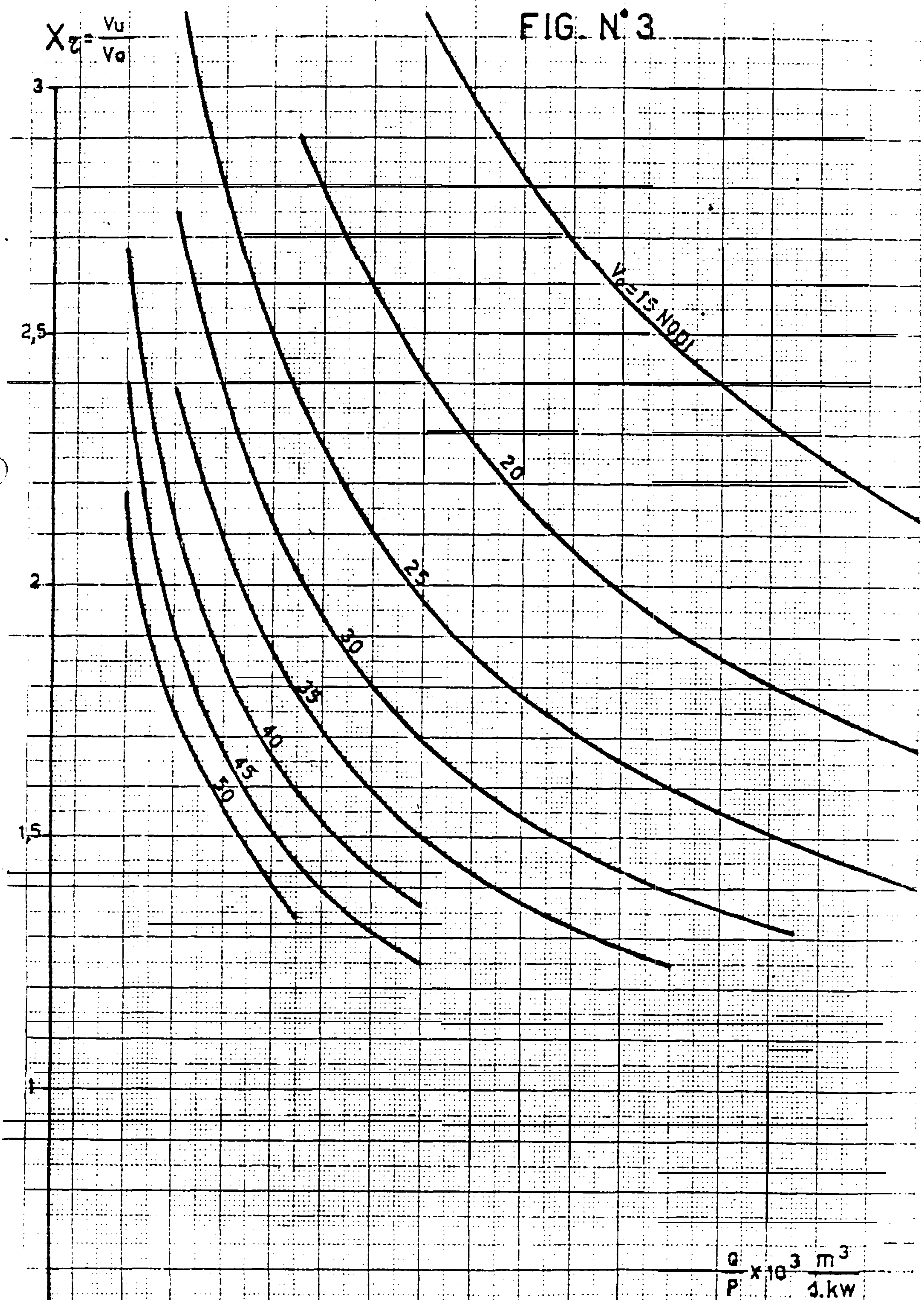


FIG. N°3
η_T
0,51
$V_0 = 20$ NODI
15
25
30
35
40
45
50
25
20
15
2,5
2
1,5
1
1
2
3
4
5
6
7
8
9 . 0^{-3}
$\frac{Q}{P}$ $\frac{m^3}{1\ kw}$

FIG. N° 3



$X_\tau = \frac{V_u}{V_o}$
3
2,5
2
1,5
1
$V_o = 15$ Nog
20
25
30
35
40
45
50
$\frac{Q}{P} \times 10^3 \frac{m^3}{J.kW}$

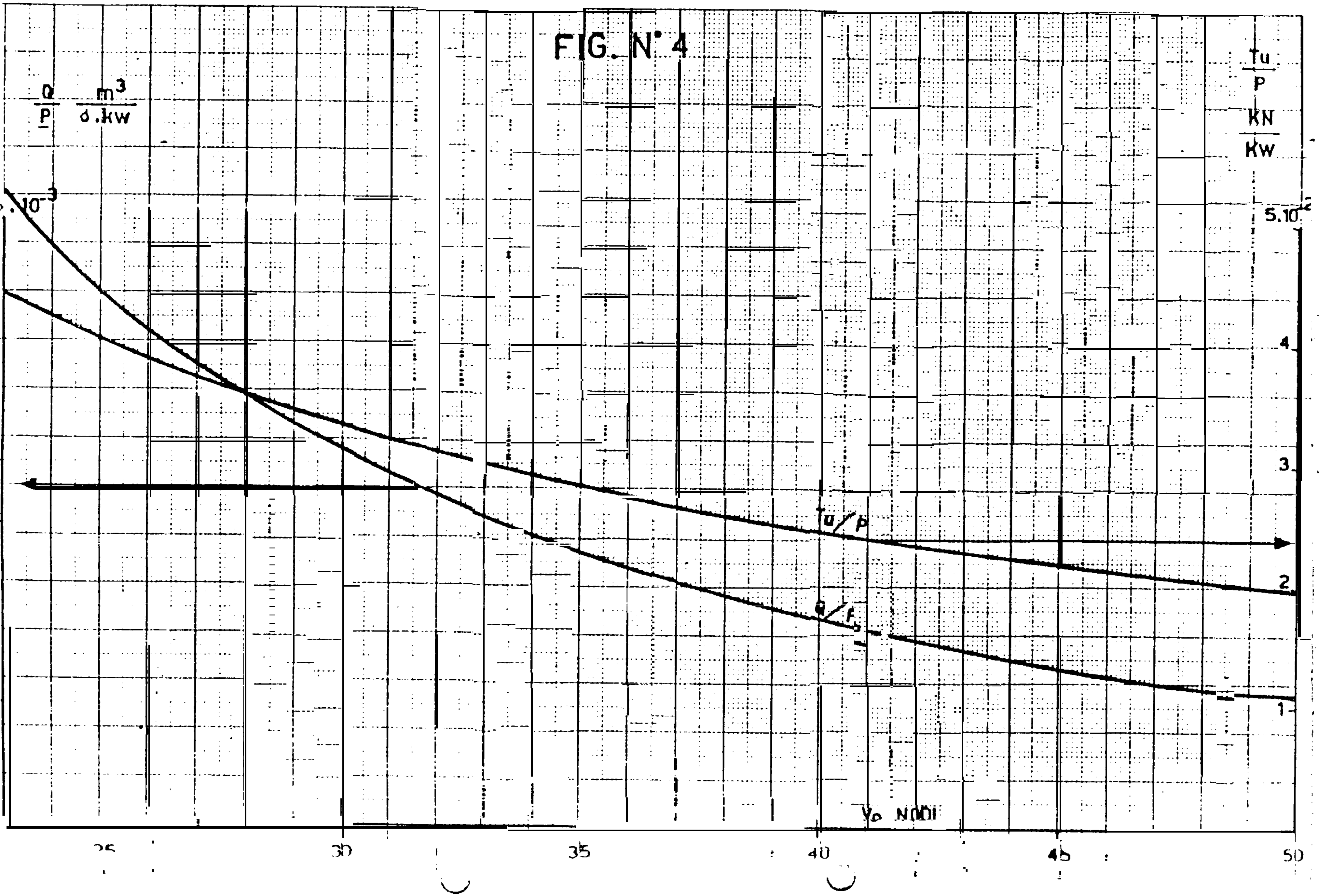


FIG. N° 4
Q/P m³/d.kW
5.10⁻³
Tu/P KN/Kw
5.10⁻²
4
3
2
1
Tu/P
Q/P
Vo N0Di
25
30
35
40
45
50

FIG. N° 5

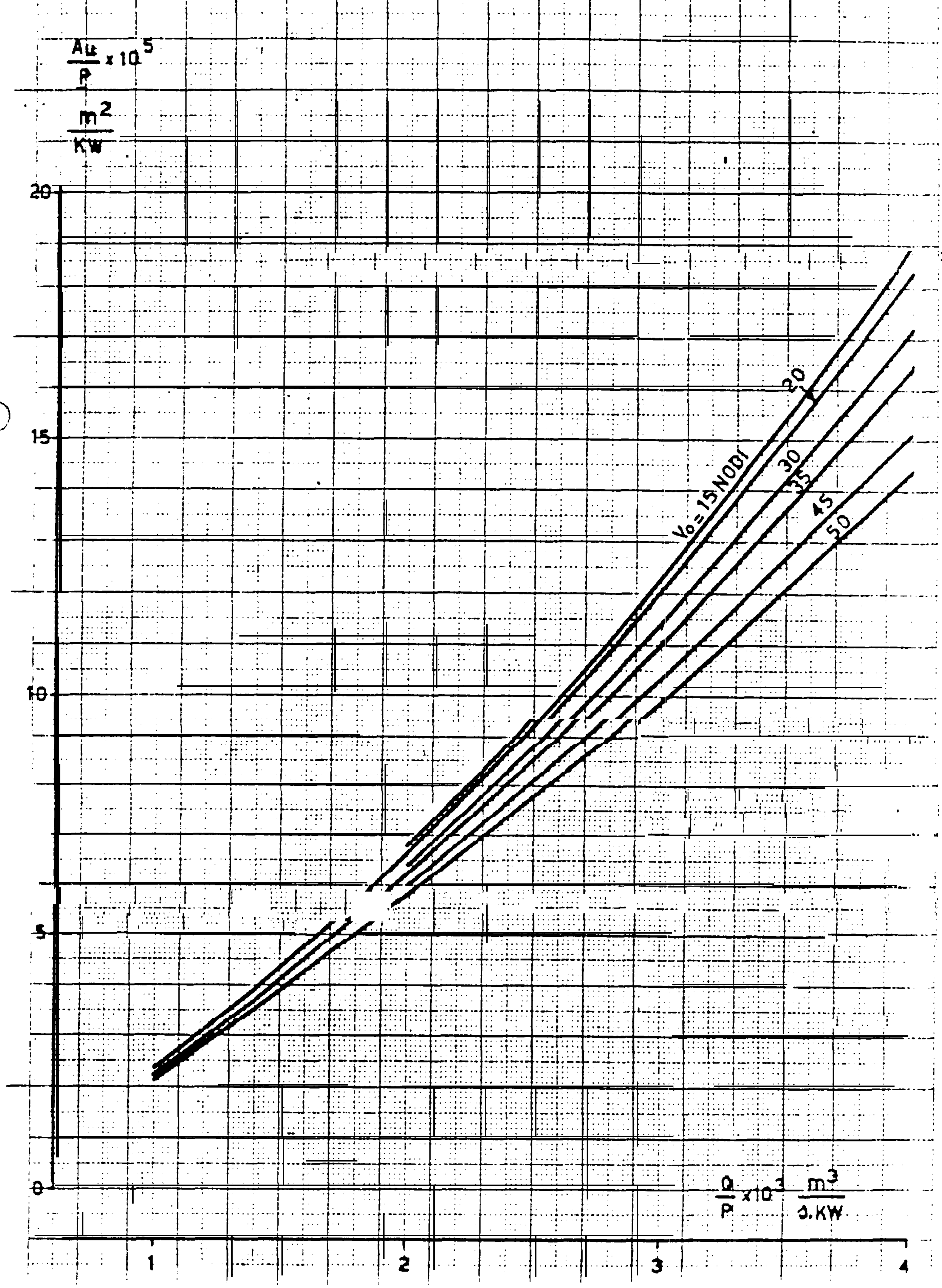
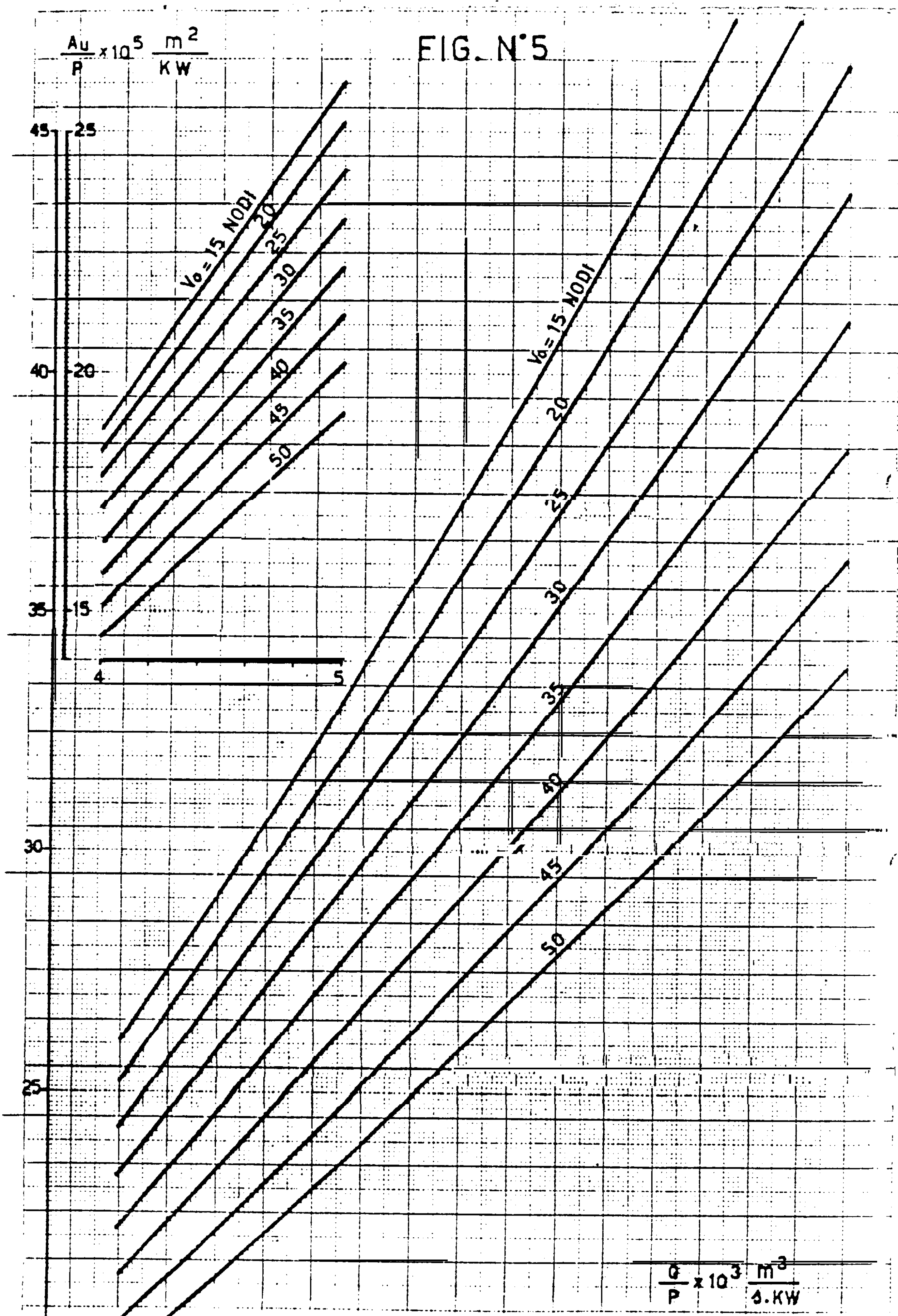
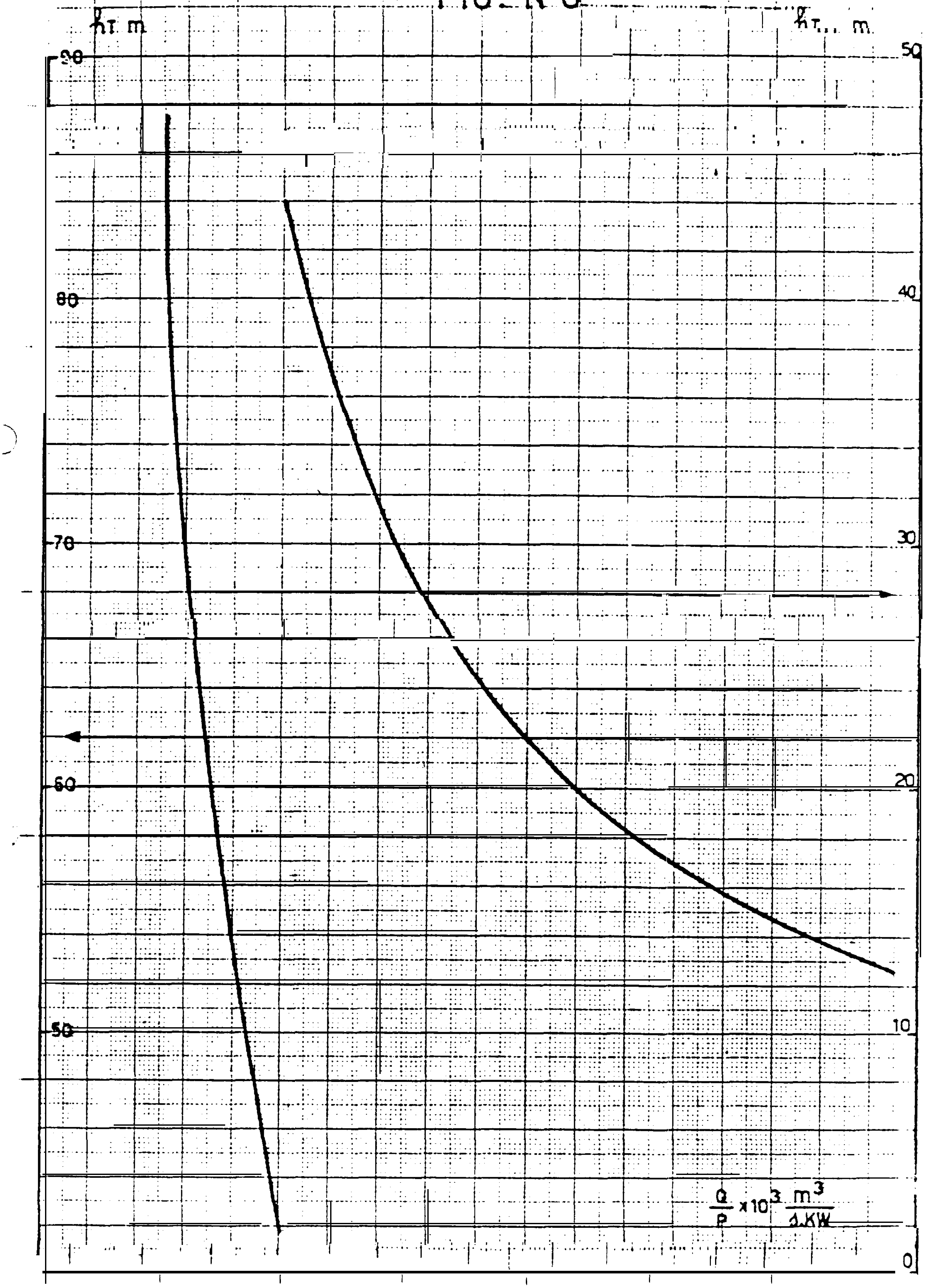

$\frac{Au}{P} \times 10^5$
$\frac{m^2}{KW}$
20
15
10
5
0
$V_o = 15$ NODI
20
30
35
45
50
$\frac{Q}{P} \times 10^3 \ \frac{m^3}{s.KW}$
1
2
3
4

FIG. N°5

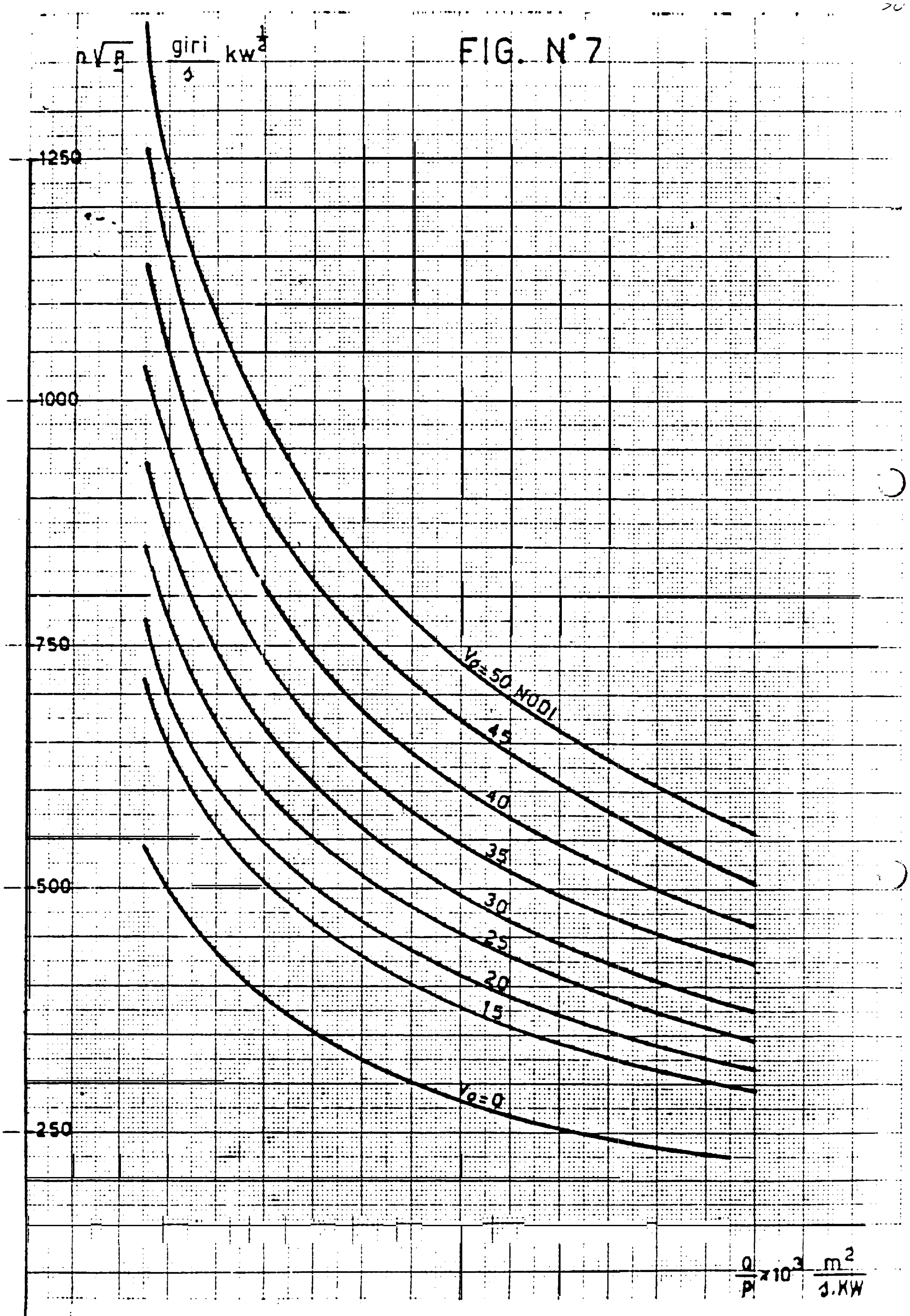


Au/P x 10^5 m²/KW
45
40
35
30
25
20
15
4
5
Vo = 15 NODI
20
25
30
35
40
45
50
Vo = 15 NODI
20
25
30
35
40
45
50
Q/P x 10^3 m³/s.KW

FIG. N°6



h_T m
h_T... m
90
80
70
60
50
50
40
30
20
10
0
$\frac{Q}{P} \times 10^3 \frac{m^3}{1 KW}$



n√P giri/s kW^½
FIG. N°7
1250
1000
750
500
250
Vo=50 NODI
45
40
35
30
25
20
15
Vo=0
Q/P x10³ m²/s.KW

FIG. N° 8

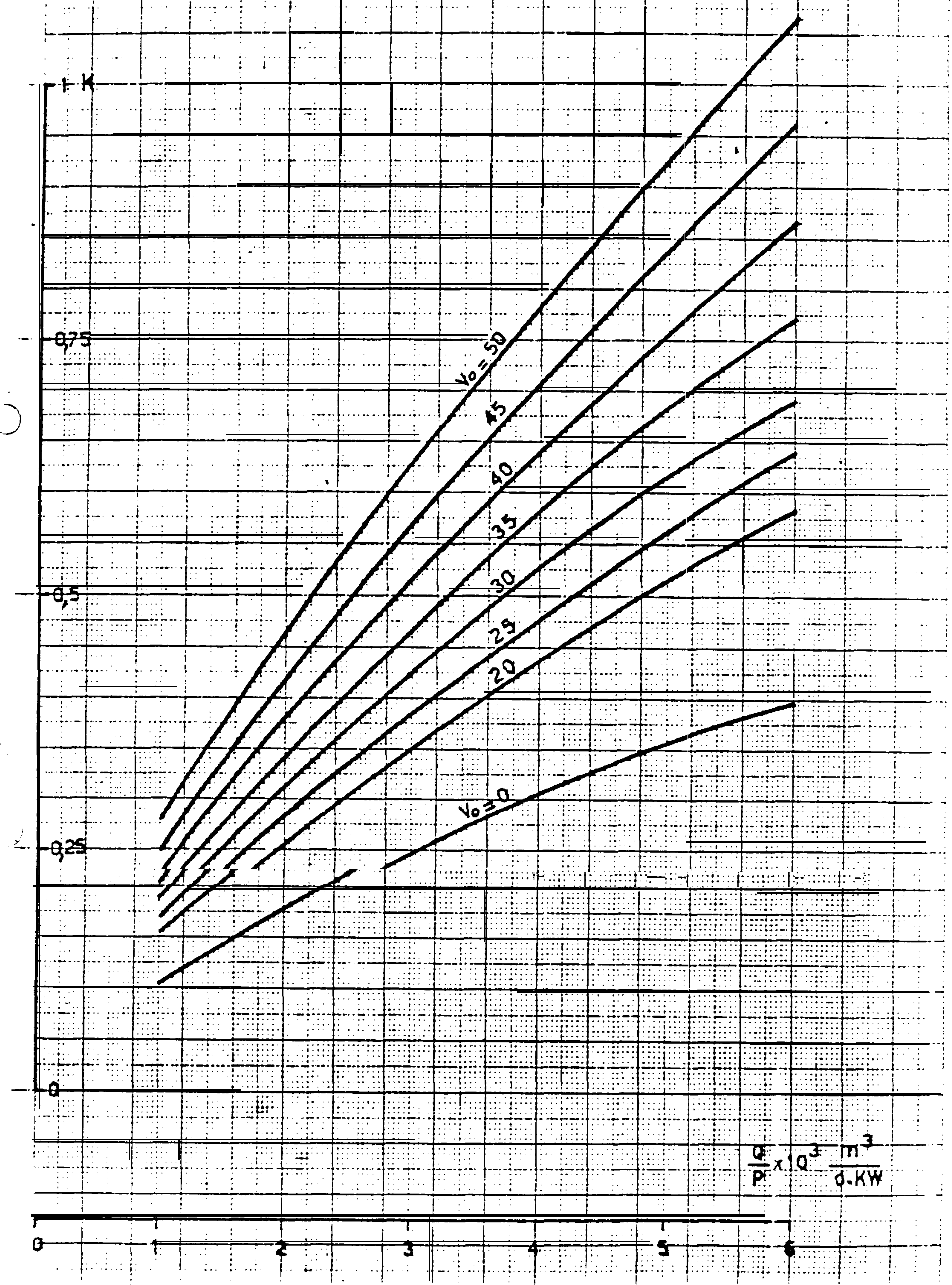
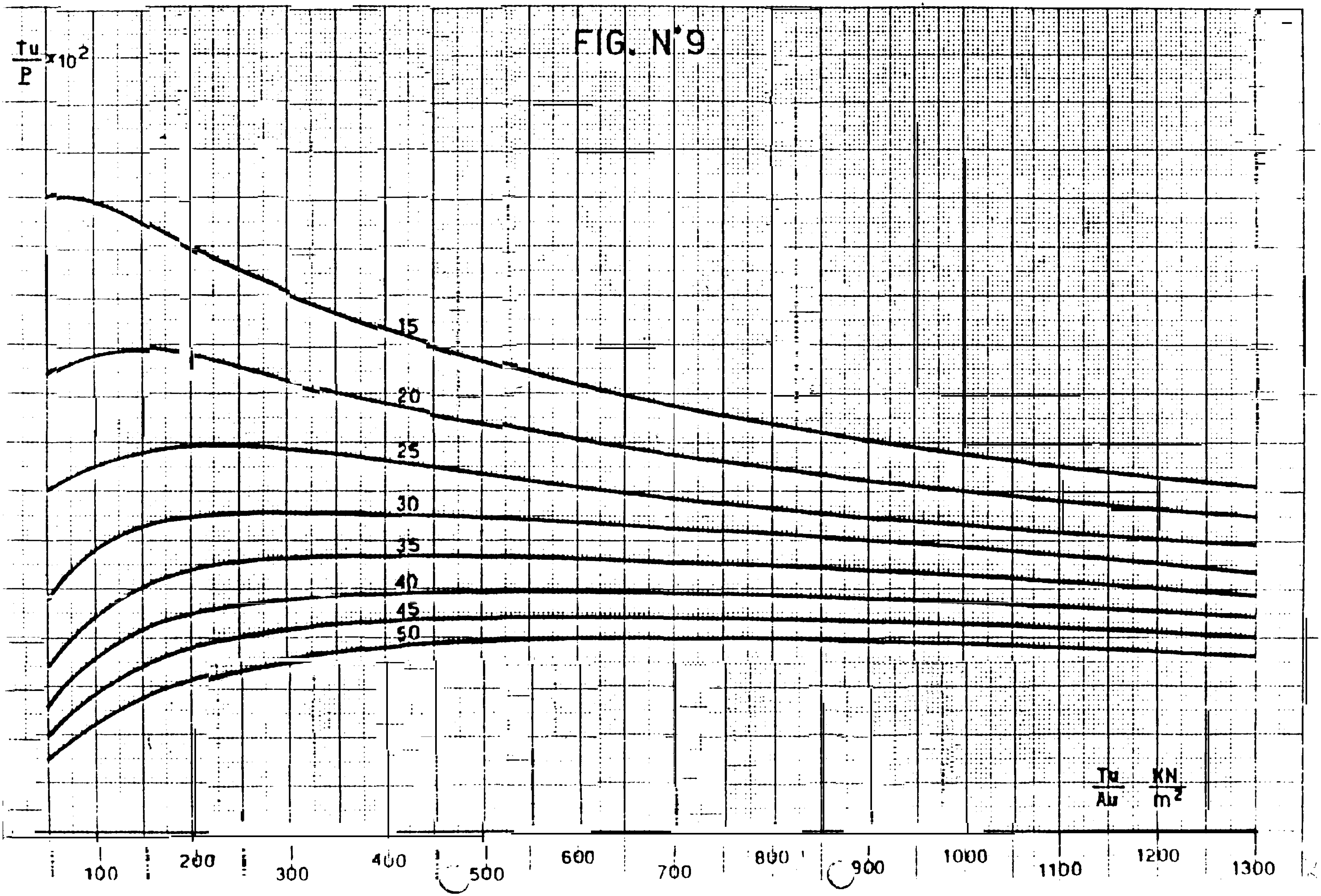
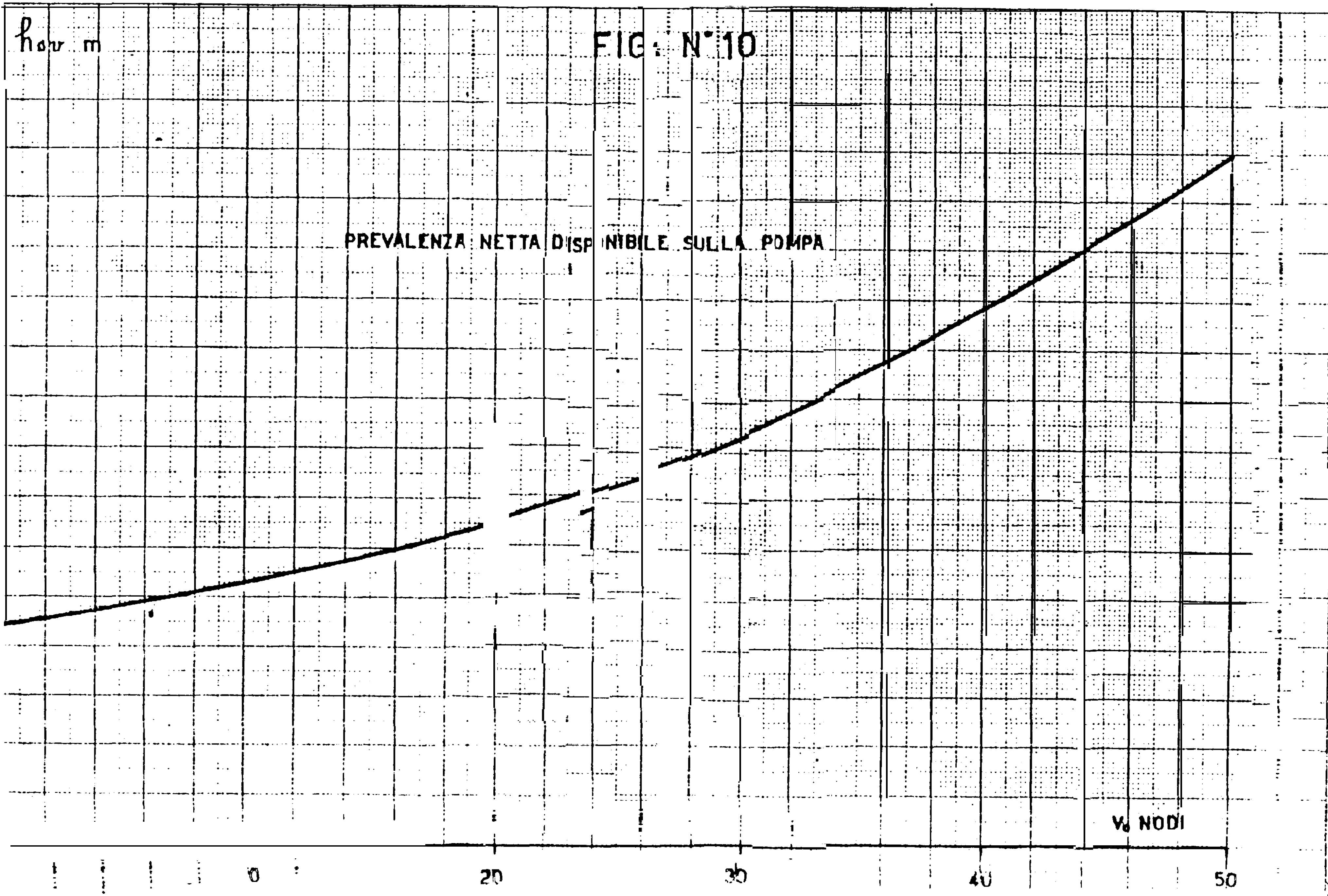

1 K
0,75
0,5
0,25
0
$V_o = 50$
45
40
35
30
25
20
$V_o = 0$
$\frac{Q}{P} \times 10^3 \ \frac{m^3}{d.KW}$
0
1
2
3
4
5
6

FIG. N°9



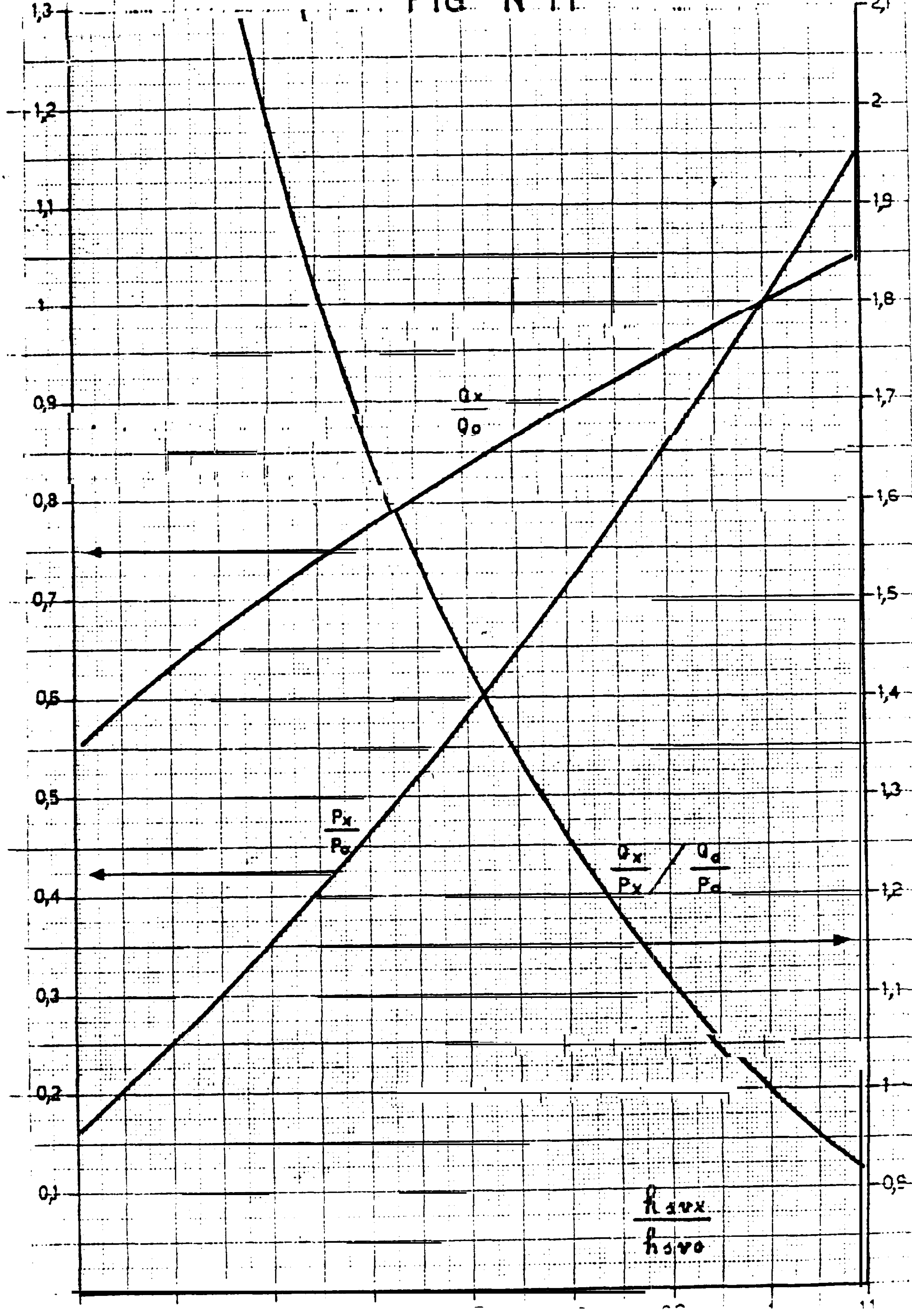
$\frac{Tu}{P}$ x10²
15
20
25
30
35
40
45
50
$\frac{Tu}{Au}$
$\frac{KN}{m^2}$
100
200
300
400
500
600
700
800
900
1000
1100
1200
1300

FIG. N° 10



h_{ov} m
PREVALENZA NETTA DISPONIBILE SULLA POMPA
V_0 NODI
10
20
30
40
50

FIG. N° 11


1,3
1,2
1,1
1
0,9
0,8
0,7
0,6
0,5
0,4
0,3
0,2
0,1
2,1
2
1,9
1,8
1,7
1,6
1,5
1,4
1,3
1,2
1,1
1
0,9
$\frac{Q_x}{Q_o}$
$\frac{P_x}{P_o}$
$\frac{Q_x}{P_x} / \frac{Q_o}{P_o}$
$\frac{h_{svx}}{h_{svo}}$