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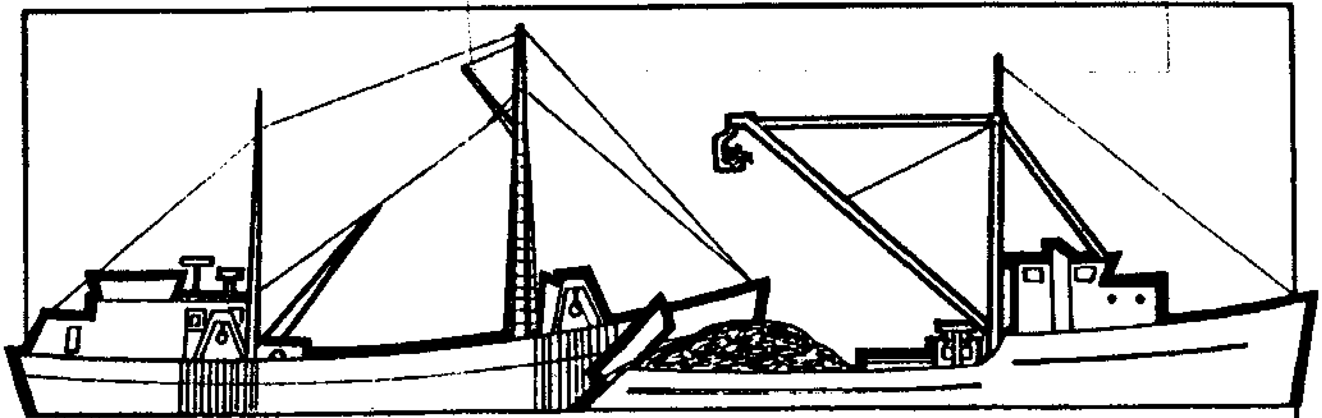

International Conference on Design Construction and Operation of Commercial Fishing Vessels--- Proceedings

John C. Sainsbury and Thomas M. Leahy

FLORIDA SEA GRANT COLLEGE MAY 1985

Copies available from:

Sea Grant Extension Program
G022 McCarty Hall
University of Florida
Gainesville, FL 32611

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FLSGP-W-85-001

INTERNATIONAL CONFERENCE
ON
DESIGN, CONSTRUCTION, AND OPERATION OF
COMMERCIAL FISHING VESSELS

Sponsored By:

Florida Sea Grant College Program
Society of Naval Architects and Marine Engineers,
Southeast Section
Florida Institute of Technology
College of Science and Engineering: Ocean Engineering Program

Held At:

Florida Institute of Technology

Edited By:
John C. Sainsbury

Thomas M. Leahy

Sea Grant Project No. IR/84-6
Grant No. NA 80AA-D-00038

Report Number 67
Florida Sea Grant College
May 1985

Price: $10.00

ACKNOWLEDGEMENTS

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Society of Naval Architects and Marine Engineers, Southeast Section
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FOREWORD

This publication is a collection of papers accepted for presentation at the International Conference on Design, Construction and Operation of Commercial Fishing Vessels. Forty papers are included; One of these "Development of a Sail Assisted FRP Fishing Canoe for the Comoros" by Tokuzo Fukamachi did not arrive in time for presentation at the Conference, but is considered to be of sufficient importance to be included in this collection.

There were 112 registrants from eighteen countries at the Conference. That and the generally high quality of the papers given confirmed that the time was right for a technical meeting of naval architects, shipbuilders, fishermen, economists and other social scientists, government and international officials, in order to update developments in the field.

This had not occurred in such a generalized manner since the third and last World Fishing Boat Conference organized by the U.N. Food and Agriculture Organization in 1967. That series of conferences led to steady and marked development in the technology of fishing vessel design, construction and operation, and it was fitting that Jan-Olof Traung, organizer of those Conferences presented a paper here which traces fishing vessel improvements since 1950 and looks to the year 2000.

During the past ten years, rapid changes have occurred worldwide in the economic climate surrounding fisheries and fishing operations due to factors such as Extended Jurisdiction and the energy crisis. As a result fishing vessels are presently in a phase of rapid development and change, especially in terms of energy consumption and fishing systems to meet the changing situation. This collection of papers is illustrative of the changes occurring, e.g., the return to sails as a means of reducing fuel consumption, the increasing extension of oil production into fishing areas, and the effect on fishing industry operations of changing demands for waterfront use.

Conference participants were asked to vote for the best paper each day, and for the best student paper. The three best papers in the overall category were: "Fishermen and Oil Men Working together in the North Sea" by Alan Wilson, "A Review of Some Recent Stability Casualties Involving Pacific Northwest Fishing Vessels" and "Model Tests of Dynamic Response of a Fishing Vessel" both by Bruce Adee, and "Particular Needs of Fishing Vessels for Use in The Developing World" by Alejandro Acosta, Theophilus Brainerd, Bambang Priyono, Norbert Simmons, and Rajapaksa Don Warnadasa. Best Student Paper award went to Douglas McGuffey who presented the paper "Effect of Loading on Fishing Vessel Stability - A Case Study".

In order to bring within bounds the immense amount of material contained in the original papers, it has been necessary to undertake some editing. In undertaking this task, I attempted to maintain the authors' text and remove only supporting material and illustrations which did not appear to be essential. However, I am sure that some authors will be disappointed; to them I apologize, and accept full responsibility, emphasizing only the necessity of reducing the material to a publishable amount.

The Conference itself would not have been possible without the support of the sponsoring organizations, and the dedication and hard work of the coordinating committee and conference staff, all of whom are listed elsewhere. We owe them a special thank you.

The authors, session chairmen, and the participants were the flesh and backbone of the whole affair, and were the ingredients of the success which they achieved so well. My thanks to them all.

I would especially like to acknowledge the work of Tom Leahy, Director of Communications and Publications for the Florida Sea Grant College Program. It is mainly through his efforts that publication of these papers has materialized.

John C. Sainsbury
Conference General Chairman
Florida Institute of Technology
Melbourne, Florida
May 1985

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*Statements contained herein are the private opinions and assertions of the writers and should therefore not be construed as reflecting the view of the sponsors, nor of any other organization with which the writers are affiliated.

The following resolution was approved unanimously by delegates to the Conference:

> "THIS CONFERENCE WISHES TO GO ON RECORD AS RECOGNIZING THE URGENT NEED TO REDUCE FISHING VESSEL CASUALTIES RESULTING FROM PROBLEMS DURING OPERATION.
>
> THERE IS AN URGENT NEED FOR BETTER UNDERSTANDING OF VESSEL BEHAVIOR UNDER REALISTIC SEA CONDITIONS, AND APPLICATION OF KNOWLEDGE TO THE OPERATING PROFILES OF INDIVIDUAL FISHERIES AND VESSELS.
>
> IT IS RECOMMENDED THAT PRIVATE, PUBLIC, AND COMMERCIAL SECTOR GROUPS, TOGETHER WITH THE UNIVERSITIES, RESEARCH AND EXTENSION AGENCIES INCREASE MARKEDLY THE LEVELS OF EFFORT AND INTERACTION IN THIS AREA. IN PARTICULAR THIS WILL REQUIRE INCREASED LEVELS OF FUNDING THROUGH INDUSTRY AND SUCH AGENCIES AS SEA GRANT, COAST GUARD, AND N.M.F.S."

PART I

FISHING VESSEL DEVELOPMENT

CHANGES IN THE WORLD FISHERIES SITUATION AND VESSEL NEEDS DURING THE FISHING REVOLUTION AND AFTER

THE PAST TWENTY YEARS

Peter Hjul
Editor
Fishing News International
London

Abstract

Awareness of new and previously underused resources, together with several technical innovations caused a "revolution" in fisheries during the years 1950 to 1970. Catches rose by five to six percent a year and estimates of end-of-century production suggested a harvest of over 120 million metric tons. However, rising catches put pressure on many stocks, and this, together with the widespread claims to 200-mile limits and soaring fuel and other costs, halted the rapid growth in production. There remains a need to find around 100 million tons of food fish a year by 2000, double the present supply. This will require further efforts to fish under-utilized species, to assist Third World countries make the best use of their resources, and to get more from fish and shellfish already being caught.

The fishing revolution

The original intent of this paper had been to review the momentous developments in the world's commercial sea fisheries over a period roughly spanning the Third FAO Fishing Boat Congress in Gothenburg in 1965 and this conference here today. For the author, it was a convenient period as he went to Gothenburg soon after taking up an appointment in London as editor of Fishing News Internationl. On reflection, however that conference, important though it was, did not mark a beginning nor an end in fishing boat design construction, or use. It did, in fact, take place when some of the most vital changes in the way we seek and catch fish were still in progress, or were just beginning.

Jan-Olof Traung, the organizer of the Series of Water Fishing Vessel Conferences, sensed this when he planned a Fourth Boats Congress for 1971; but by that time the Food and Agriculture Organization itself was changing and those in charge of fisheries were looking to other types of conferences. The major one to assess the state of the world's fish resources and the pressures on them took place in Vancouver in 1973, and a repetition appears planned through a monster FAO Meeting on Fisheries Management and Development to be held in Rome at the end of June, 1984. However, there is unlikely to be much practical talk there of vessels, methods, processes or products. For an opportunity to examine essential aspects of the fishing industry and its future, we have to thank people and those who they persuade to come and share their technical knowledge.

In hindsight, the world-wide fishing industry appears to have experienced a revolution, which, like that of France in the 18th Century or if Iran today, was no sudden turnaround but rather a protracted sequence of drastic and traumatic changes.

The early years

An example of these to come was to be found in 1948 in a village called Laaiplek on the West Coast of the Cape Province in South Africa. Fishing in the Cape had been transformed by the discovery that a small pelagic fish called the pilchard could be caught in great quantities for canning or for reduction into readily saleable fish meal and oil.

The industry there was at the beginning of a boom that would for a while put South Africa among the top fishing nations. Laaiplek, once a bleak community of impoverished fishermen, was the site of one of the new factories. It was a time when people still believed the sea to be "inexhaustible": with its fish and shellfish just waiting for the enterprising exploiter. To help it do this, the company at Laaiplek had called in expert help from the sardine industry of California. Three Neptune mechanical packing lines for A1 talls and a new Enterprise fish meal plant revealed the American experience in this kind of fishing.

It also revealed two other things that presaged the coming revolution in fisheries.

First, the American involvement reflected the concern in California over a decline in the sardine (pilchard) catches - the first of many dramatic collapses that would be blamed (perhaps not entirely correctly) on over-fishing. California was to be one of the early post-war signs that the sea was not inexhaustible and that care had to be exercised in management of precious resources.

Second, with all the evidence of abundant stocks of pilchards and jack mackerel waiting to be exploited, South Africans were eager to try anthing that would get them the biggest catches in the shortest possible time.

The top boat then feeding the factory was a wooden hull lampara seiner, newly built for about $11,000. Her net was worked by hand, with the help of a crude mechanical winch and pulley. The net was made of natural fibre, and a haul in it of 37 tons was talked about up and down the coast. There was no radio nor any other instrument in the wheelhouse other than a magnetic compass. Navigation was usually by sight of landmarks on the shore. But the boat did boast a new Caterpillar 80 hp engine, one of the first of many to be imported to power the growing fleet of Southern Africa.

From Laaiplek and nearby Velddrif, the pilchard boom was reaching 600 miles north to Walvis Bay in South West Africa (now Namibia). Boats grew quickly to 50, to 60 and 70 ft.

All the time in those and many other emerging fisheries, the status of the fishermen was changing and so was the industry's attitude to its equipment and its boats.

Britain, France, Japan and other fishing nations whose fleets had been devastated by war or forced to stand still for years were doing more than just replacing vessels. They were also experimenting with new types, new gear and new ideas.

In addition to some remarkable recoveries, and a few steep declines, the 1950's saw another aspect of the revolution - the emergence of fisheries and fishing countries starting almost from nothing.

Peru was a boom industry more on the Southern African pattern. There was very little there on which to base any kind of fishing activity. The Peru catch in 1938 was 23,000 tons. At the beginning of the anchoveta surge in 1958 it was 235,000 tons. By 1970, it had soared past ten million tons a year and was to peak soon after at 12.5 million tons. There were then some 1500 purse seiners supplying more than 100 meal plants. Aboard the boats were fishermen and even skippers who, until the boom, had never even seen the sea, let alone move out onto it. Peru was the world's largest catcher by the end of the 1960's, followed by Japan, and then by another newcomer to the fishing top league.

Like Poland and other Eastern European neighbors, Russia does not have a large coastal sea fishery resource. Her development resulted from a deliberate decision in the mid-1950s to obtain protein from fish by sending ships out to work distant stocks that were then still readily avaiable.

Russia, Peru and the expanding industries of Iceland, Norway, Canada and Japan, were leading the fishing revolution because they were making use of resource opportunities partly by going out and finding them, but also because significant advances in the techniques of fish finding, navigation, vessel design and construction (including use of materials such as GRP), in gear materials and handling, and in processing and preservation enormously increased their hunting capacity.

Further, the revolution was a response to an increasing demand for fish, and to an effort to assist developing countries make use of it as a nourishing food and an export earner. But it was the advances in technology that made it all possible.

The new technology

It is possible to mention only the most obvious of these profoundly significant technical developments of the 1950s and 1960s:

1. The British experiment with the stern trawler and with freezing at sea. From the Fairfree trials of the late 1940s, the Salvesen Company in Scotland planned and built its three Fairtry factory trawlers.

Not only did these ships demonstrate the advantages of trawling from the stern- they also pointed the way to hull arrangements that would accommodate processing machines and freezers in a factory deck.

Since the 1920s, Rudolf Baader of Luebeck had been devising machines, compact and proved by tests in shore factories, that were becoming available just at the time of the stern trawler experiments. Together with the horizontal and later the vertical plate freezers, they made it possible to take a factory and a freezing plant and cold store out to sea.

Although the Fairtrys enjoyed only a brief period of profitability, the idea quickly caught on in countries seeking ways of getting their fishermen to distant resources. Russia was one of the first, with an order placed in West Germany for 24 ships very similar to the Fairtrys. By the time of the Gothenburg conference, Britain and Russia had been joined by Japan, Poland, East Germany, Norway and Spain as operators of stern trawlers.

The heydays of the ocean-roaming fish factories have been compared to the age of the dinosaurs in the sense that they marked the upper extreme in size and capacity in the industry. If the high-performance processor trawler or big purse seiner was the Tyrannosaurus rex of this brief era, the meal factory ship was the gluttonous Brontosaurus.

Two such ships were allowed by South Africa to work for about three years at the end of the 1960s off the coast of Namibia. They had daily capacities of 2000 and 2500 tons (exceeded later by the 3000 tons of a Norwegian ship, the Norglobal). In 1968, the factory ship Suiderkruis was being fed by 18 catcher purse seiners, each about 75 ft. long and and worked by eight crew. The top boats were catching 30,000 tons of pilchards a year and their crews were said to be among the world's richest fishermen. In 1968 and again in 1969, the owners of the Suiderkruis reported profits of more than 1 millions pounds.

By the early 1970s protests from the shore forced the factories out. They went north to West Africa where they were joined by the Norglobal and the Astra, also from Norway. But the fishery of Namibia carries the scars of their pillage and the yearly haul of pelagic schoal fish is still less than a tenth of what it was in the years just before the factory ships arrived.

2. Floating factories such as these meal ships were able to get consistent large supplies of fish partly due to invention of the Power Block and because of the introduction of synthetic fibres to netting.

Mario Puretic's power block for purse seines in 1953 and its application to fishing by the Marco company of Seattle ranks among the most significant of all technical innovations in fisheries. When Peter Schmidt of Marco visited Southern Africa in the 1950s not long after his company had introduced Puretic's block to salmon seiners working out of Puget Sound, even bigger applications were being made in Iceland. Soon the power block in conjunction with the lighter and stronger synthetic fibre nets was to turn the ailing pole and line tuna clipper of California into the highly prosperous purse seiner.

The excesses in fishing that ruined the stocks off Namibia, that still threaten the tunas, and which have made the purse seine the most feared of all man-made predators of fish should not detract from the inventive genius of Puretic and the development flair of Schmidt and Marco. Their combination provides a classic study of R&D in the fishing industry.

Along with the power block and the nylon nets, should also be linked the work of applying hydraulic power to hauling machinery. In particular, this development extended high-capacity fishing to even the smaller boat and is today an important feature of the compact, multi-purpose vessels extensively used by industries operating within their 200-mile limits.

3. The novel idea that the electrically generated sound pulses and echoes of the early depth meters might be used to locate fish was being tested in the 1930s. In Britain herring drifer skipper Ronnie Balls began getting interesting results from a set in his boat the Violet & Rose. The war interrupted this work but Balls and others took it up again in the later 1940s. By the 1950s the fish finding echo sounder was quickly becoming an essential aid to catching.

   A logical offshoot from this development was the application of side and forward probing asdic to fishing in the form of sonar. A number of countries participated in this but the early commercial drive was by Simrad of Norway.

   With sonar even more than up-and-down echo sounding, interpretation of the echoes was for years dependent on the skills of the skipper in reading the signals. Now, with microprocessor technology, the modern fish finding sonars and sounders "think" their way through all that their high-performance transducers pick up from the sea. Fish spotting has become a precise operation with the boat able to use its instruments to garget right onto its quarry.

   Other instruments have been part of the fishing revolution - position fixing Decca Navigator, Loran, Omega and now satnav; radar; weather facsimile recorders; gyro compasses and autopilots; and all the advances in radio communications. The wheelhouse of a modern fishing boat is as instrument packed as that of a warship, and more than that of the average merchant ship. The fishing skipper has become a skilled and enterprising user of electronic aids.

4. The fishing echo sounder and sonar were to play an important role in the 1960s in the trials leading to the one-boat mid-water or pelagic

trawl. This work was undertaken from West Germany by a team led by Dr. Joachim Scharfe. Between June 1959 and October 1968, 43 trial trips in commercial and research ships showed that big nets towed by ships with sufficient power between the sea bottom and the surface could take huge hauls of shoaling species such as the herring. One early success was an increase in the West German herring catch from 18,000 tons in 1964 to 93,000 tons, 93 percent caught in pelagic trawls.

One of the most recent applications of mid-water trawling is to fish the concentrated shoals of small blue whiting as they migrate in the winter and early spring to the west of the British Isles. Norwegian trawler/purse seiners now take around 180,000 tons of blue whiting a year fishing down to 600-800 fathoms.

Another application has been by German, Polish and other trawlers test fishing for the small crustacean krill in the Southern Ocean.

With the stern trawler design, synthetic fibre nets and improved gear handling machinery, the attractions of distant waters encouraged the building of larger and larger ships.

Biggest of all was probably the Russian prototype class Gorizont, 364 ft. long and 7000 hp. In regular service all over the world are the East German built Super-Atlantiks, 335 ft. long with a claimed processing capacity of 125 tons.

From Italy came a 352 ft. super-trawler of 4000 hp with four Baader lines for filleting her catch. She was bought last year by a Faroe Island company and is now being employed fishing for blue whiting which is processed aboard into minced product for use in fish sticks.

5. Improvements and innovations in handling the catches and in processing and preserving them contributed to the fishing revolution. Apart from the processing machines and the plate freezers, they included plastic boxes which have encouraged the trend to boxing catches at sea; compact and economic ice-making machines; advances in refrigeration to increase opportunities for deep freezing and chilling-smoking plant such as the Torry kiln; shipborne meal plants; shore meal plants with stickwater evaporators and other devices to extract the maximum yield from industrial fish; and pelleted and powdered bulk meal transport and storage.

Rapid increase in catches

One very noticeable effect of this revolution was the rise year by year in catches, reported to FAO and compiled in its Yearbooks of Fishery Statistics. From the early 1950s and on to the beginning of the 1970s, the growth in the harvest of fish and shellfish was a remarkable five to six percent a year.

By the late 1940s the recovering fishing industries had restored the pre-war total of around 20 million tons a year. Some 15 years later, at the time of the Gothenburg conference, the total had reached 53 million tons with the marine sector contributing over 45 million tons.

A few years later the total was 60.5 million tons and by 1971 it was 70 million tons.

At that stage the industry had accepted what appeared to be inexorable expansion accepted confident forecasts for the future were accepted without looking too closely at some ominous clouds looming just over the horizon.

Over some 20 years, agencies such as FAO working in the Third World had been striving to introduce the benefits of modern fishing technology to boost fish supplies. There were the inevitable failure. Workers in the field learned by hard experience that not all machines or improved techniques are suitable for improving traditional small-scale fisheries.

But many developing countries were participating in the general rise of fisheries, not only Chile or Peru.

At about the end of the 1960s FAO attempted to relate its experiences in fisheries to what it estimated to be the fish protein need of the world at the end of the 20th century. It calculated that this would probably rise to over 120 million tons and suggested it could be met by an increase in the marine catch of known species by known methods to around 100 million tons. To this might be added another 20 million tons or more from inland fisheries with the main contribution coming from aquaculture.

The plateau of the 1970s

During the early 1970s the yearly catch settled onto a plateau at around 68 to 70 million tons. This was caused partly by a slump in the Peru fishery due to the effects of an El Nino; however, several other countries were reporting severe declines, and pressures began to build for more coastal state protection of stocks heavily fished by distant water fleets.

When the United Nations Law of the Sea Conference assembled in Venezuela for what was to be the first of many meetings, high on the agenda was the question of fishing limits.

Iceland could not wait. First she claimed 50 miles and, after contesting this with British and other ships, got it accepted. Iceland's trawler fleet was expanded to take the increased share of the resource. Soon she was claiming the full 200-miles. Eventually this was accepted and by 1977 Britain's huge distant water trawler fleet had been forced out of its most valuable grounds. As the Icelandic deepsea fleet grew, the British fleet faded away. In 1975, it totalled 168 distant water ships, with 45 of them freezer stern trawlers. This has now all but disappeared.

Within the EEC, Britain has negotiated a 36 percent share of the most popular species. But the fleet taking it consists mainly of compact coastal-type trawlers and seine netters, and smaller boats.

For a while Iceland revelled in her hard-won new limits as the cod catch rose past 400,000 tons a year, all of it for her boats and factories. But the stern trawler fleet has now grown to 104 ships and poor year-classes have meant less cod. This year her industry has had to go onto vessel quotas to eke out a permitted cod haul of only 220,000 tons.

This is just one example of how wider limits have failed to produce the fishing bonanzas expected by their protagonists. In North America there are the problems of the Pacific Northwest and Alaska fisheries, and the troubled fishery of the Canadian East Coast.

These cases bear out the warning given by several authorities when the agitation for wider limits was at its height. One of them, Dr. John Gulland of FAO, showed that in 1970 non-local fleets took 7.1 million tons in a world marine total of 54.6 million tons. Other figures indicated an even higher proportion - up to 16.4 million in 58 million tons in the mid-1970s. No less than 11 million tons of this was from waters that would be enclosed by the wider limits.

One early effect, therefore, of the exclusive economic zones was a switch in catching effort from the high-performance distant water fleets to often smaller and usually less efficient coastal craft from the littoral state. In some cases this led to an improvement in fishing and in management of the stocks. In others catches fell or, as in the case of Iceland, over-capacity soon built up in the national fleet.

Soaring costs

With the spread of limits, the fear for stocks and the fishing collapses, the 1970s also brought the oil crisis.

In Britain it was noted that from 8 million pounds in 1972, the fuel bill of the fishing industry increased to 27.5 million pounds in 1974 and was expected to rise to over 33 million pounds in 1976. In France fuel prices went up 350 percent between 1970 and 1976, while fish prices rose 65 percent.

If limit extensions had not impeded the progress of distant water fishing, it might well have been stopped for many countries by the soaring price of fuel linked to the much slower rise in the price of fish.

As it is, some countries and fisheries could be reaching the unhappy position where it may no longer be economic to go out for species that do not command attractive prices.

More and more, as species and area controls put ceilings on how much can be fished, industries will need to consider allocating the catches permitted among skippers, owners or vessels. Such individual quotas are anathema to fishermen steeped in the idea of their craft as one involving great risks alleviated by great opportunities. But in this age of the Exclusive Economic Zone (EEZ), quotas and high costs, the fisherman is no longer gambling with a fair chance.

Canada is already applying a form of vessel quotas on her East Coast. Iceland is introducing quotas this year, and they are sure to spread to other fisheries.

The future

Given these and other curbs on free fishing of a common resource, what is the future of the industry and its boats?

First, the idea of replacing the millions of tons of hunted fish with the increasing crops of farms must be rejected.

Aquaculture appears sure of a strong future. Already it is contributing an estimated seven to eight million tons to the world supply of fish and shellfish. But much of this still comes from non-intensive, small-scale pond farming long established in countries such as China, the Philippines, Indonesia and India.

Intensive farming while holding out much promise is still mainly in a development stage, despite the successes round the world in trout farming, in the USA in freshwater catfish and in Norway and Scotland in farming Atlantic salmon. There is great promise in the non-intensive pond growing of penaeid shrimp. The mussel crops of Spain, France and Holland total hundreds of thousands of tons. However, it is likely to be necessary to wait well past the year 2000 before aquaculture produces the 20 million tons a year forecast for it 15 years ago.

In 1983, the world total harvest by hunting and farming was probably about the same as the 75 million tons of 1982. Of this, about 25 million tons went to fish meal leaving 50 million tons for direct food use.

In Rome during 1983, FAO Director-General Edouard Saouma said the food fish supply would have to be more than doubled by the year 2000 if the industry was to meet world consumption needs. But world production is presently rising by only about one percent a year and the total at this rate would barely reach 90 million tons.

Immediate challenges therefore are to get the best possible use from fish already being caught or farmed. This means reducing waste (for example, it is estimated that five to six million tons of so-called "trash" fish is being discarded each year by shrimp boats); developing new products such as crab sticks from Alaska pollack or minced fish from blue whiting; working on the restoration of once-great fish runs (in Norway, for example, there is strong evidence that the Atlanto-Scandian herring stock is recovering); and developing stock enhancement and ranching.

FAO, quite rightly, stresses the important role to be played by the developing countries in getting the maximum possible use of resources within the new economic zone. Work in the Third World fisheries will be discussed in a later session of this conference; for the present it may be noted that the needs and the possibilities of these fisheries are vital to any consideration of the future of the industry.

Established large industries that have fallen on hard times could well be revived, given the right injections of investment, the markets and the will to recover.

As an example, Peru has seen her industry plunge from the top of the world to a catch in 1983 of 1.42 million tons. Even that may seem a good enough haul. But it has to support a fleet that still exceeds 300 vessels and too many meal plants and factories ashore. One urgent need is to cut down the number of meal plants to eight or ten (from over 100 in 1971). Another is to modernize an ageing and inadequate fleet so that Peru's fishermen can go after food species in the deeper waters, such as the mackerel and horse mackerel.

Looking to underutilized species, there does not appear to be untried options left.

The small ocean mesopelagic fish are mentioned in the more optimistic forecasts which estimate the resource at anything from a few million up to 100 million tons. But these fish, like so many others, are underused because they are small and difficult to process, are in remote fishing areas and even with the help of modern electronics may be hard to find.

The cephalopod resource offers a better prospect with estimates ranging from 10 million to 50 million tons and more. As with the mesopelagics, exploitation will require mainly long-range fleets and the cost of ships and fuel may curb enthusiasm.

The much-publicized krill resource in the Southern Ocean is estimated to be capable of yielding catches from a few million to hundreds of millions of tons a year. Krill has now been investigated, caught and test processed for well over ten years. The catch in 1980 was 480,000 tons with Russia, Japan and Poland among the main catchers. In 1981 it dropped to 450,000 tons. It seems that krill fishing using a big pelagic trawl will have to be done by large processing ships, which are costly to build and run.

Just how expensive is indicated by the latest freezer stern trawlers built in Holland in 1983 and this year. The biggest so far is the 312 ft. long Dirk Dirk, a ship of 3019 gross tons and capable of freezing 180 tons of fish a day. She is powered by a 4300 hp MaK engine and is reported to have cost about $8.5 million. Dozens of ships of this type and size might be needed even to take the most modest estimate of the possible krill catch.

Perhaps it might be better to follow the advice of some American Pacific Coast researchers and try to seed the Southern Ocean with salmon. They would at least convert the abundant krill into a readily acceptable fish protein!

50 YEARS OF IMPROVEMENTS IN FISHING VESSELS - FROM 1950 TO 2000

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Abstract

Acoustic fish detection instruments and fishing gear made up of man-made strong fibres increased the fishing power considerably. The main machinery went from heavy-duty types like diesels and steam engines to light-weight automotive type diesels and sterling engines coupled through reduction gears with large ratios to ensure high propeller efficiency. Propellers were fitted with controllable blades and installed in nozzles. Hydraulics were introduced for the flexible drive of fishing winches. Special powered rollers and blocks were introduced as well as drums for the haul of purse seines and trawls. Full scale measurement and results from model tests gave designers the possibility of designing hulls with minimum power requirements and at the same time having sufficient stability and good seakindliness. Toward the end of the period a new optimism went through the fishing industry which then again invested considerably in medium-sized fishing vessels. A number of major International organizations like FAO, Unesco's IOC, EEC and OECD joined forces and built a most unorthodox prototype fishing vessel to celebrate the 21st century.

The 50s

In 1950 the world had started to recover after World War 2. In great secrecy during the occupation, France had planned a new fishing fleet for which, after the peace, they at once had placed orders also in the UK and the USA. That fleet had now begun to fish and those vessels influenced the design of similar vessels particularly in Germany, the Netherlands and the UK. The Scandinavian countries had got together in 1947 to compare notes of their fishing vessels and they found great differences between vessels fishing on the same waters. The learned naval architecture societies like RINA and SNAME published a few papers describing fishing vessel development. Hardy (1947) had described fishing vessel developments in many places in the world and thus also emphasized how little, if any, cooperation there had been between nations to develop efficient and economic fishing vessels.

Small vessels were built of wood and, when longer than about 80 ft, of steel. Most engines were heavy-duty, if not semi-diesels and steam. Some echo sounders had come into use for depth indication - but not for fish detection. Winches were powered from the main engines by belt or chain not hydraulically or electrically. Trawls were handled from the side. Purse seines were handled either from two smaller boats or by the derrick of the main mast and a turntable. Netting materials were natural fibres which were very much subject to deterioration by rot or sun. Fish was preserved by ice. Crews accommodations were ascetic.

The Pacific coast fishing vessels of USA fished from the stern, and they had the wheel house forward. They created much interest in Europe. Also whale factory ships which hauled up the whales on a slip aft stimulated designers to consider stern fishing. The first fish factory trawler, Fairfree, working from the stern appeared in 1949 and she was soon followed by the Fairtry class built for the UK and USSR.

The United Nations Food and Agriculture Organization (FAO) organized in 1953 a World Fishing Boat Congress with sessions in Paris and Miami. The papers again referred to the many different fishing vessel types being used around the world. They discussed the important aspects of powering, whether high-speed or low-speed engines, whether fixed blade or controllable pitch propellers. They took up the problems of hull shape with regard to resistance in calm water and increased resistance in waves and in one important study measurements of the behaviour in a high sea of actual fishing vessels were given and discussed.

The second half of the 50s was then characterized by intense development work on all aspects of fishing vessel design and construction such as model testing to develop more economic and seakindlier hull shapes, computer techniques to optimize results from such tests, rational formulation of construction rules for wooden ships, multiple reduction gears for main engines, nozzles and diesel-electric drives.

In 1957 FAO organized a Congress on Fishing Gear which again presented basic information on all the different fishing methods used and also for the first time reviewed the progress on fish finding, gear testing and new materials like man-made fibres. Those fibres permitted completely new designs of more efficient fishing gears which then also could be kept on board much easier than before, not being subject to rot or deterioration by the sun. These stronger nets could also be handled mechanically, thus a powered block for the hauling of large purse seines was developed by the American fisherman Puretic. A number of reports were given to the Congress on the use of echo sounders for fish detection, also the use of horizontal echo ranging = asdic = sonar. Similarly the use of light and electricity to attract fish was covered in papers.

The FAO Second Fishing Boat Congress in Rome in 1959 demonstrated now the large steps taken in fishing vessel development during the 50s. One paper on the use of glass reinforced plastics was presented. Again engineering was dealt with at large. There was a long list of papers dealing with resistance,seakindliness, and the use of computer techniques to develop optimum hull shapes. The use of bulbs and the choice of the optimum prismatic coefficient was suggested. The importance of sufficient stability was dealt with by several authors. Very small craft were covered in detail.

The time had come to predict the future, 15 years hence or 1975, and among the many ideas presented, this author stated:

> " There are many technical developments which have been successfully applied on a laboratory or pilot scale, but which are not in common use by the industry. Here are a few:
>
> - Echo-ranging (asdic)
> - Fish attraction (light, electricity, vibration)
> - Fish collection (pumps)
> - Net design (using synthetic fibres and engineering principles)
> - Underwater television (for record of gear behaviour and of fish entering the gear)
> - Mechanized handling of trawl gear (by stern trawling or winding the gear directly on the winch)
> - Extension of storage time (chilled seawater, anti-biotics and radiation)
> - Transfer of crews and cargoes by airplane
> - Fishing under ice (submarine)
> - Artificial upwellings (by nuclear heating)
> - New materials (plastic, aluminium, rubber)
> - New power plants (gas turbines, nuclear, Wankel principle)
>
> The fishing boat designer must keep such possibilities in mind, and he must also follow the development of his own profession, and in this he cannot avoid seeing how new boat types are being developed and how knowledge of theoretical naval architecture is being acquired at an accelerated pace."

The 60s

This decade also was one full of developments. The Second FAO Fishing Gear Congress in 1963 was the venue for a number of important papers describing further developments of man-made fibres and their applications in more effective fishing gear. Similarly progress in the use of acoustics for fish detection was reviewed. Several papers covered the powering of winches, particularly the use of hydraulic power to ensure a high degree of flexibility. The powered block had come to stay.

The fishing vessel designers and builders were quick to utilize the development in the fishing methods field. Much of this was reported to the Third FAO Fishing Boat Congress in 1965. Now one also started to consider techno-socio-economic problems and the very small fishing boats.

As before, FAO gave high priority to questions of hull shape in connection with economy of powering and seakindliness because they felt that this was a common field where one country could learn from the other even if the fishing methods were different and boat types and boat sizes were not the same. A first report was given on FAO computer statistical analyses of a great number of results from model tests of fishing vessels which FAO had succeeded in collecting from various model testing establishments around the world - and from individual owners of such tests. With the aid of the study four vessels were designed, 40, 55, 70 and 85 ft. Models were tested in two different establishments and, as Figure 7 shows, they proved to be better than anything else earlier tested, e.g., better than any of the models which had been used to make up the regression analyses. The final analysis, together with the obtained coefficients, was published in a separate document by FAO (Hayes and Engvall 1969).

FAO had organized a meeting on stability utilizing the Rahola suggestion for minimum dynamic stability in Gdansk in The International Maritime Organization (IMO) organized, together with FAO, a fishing vessel stability working group which after many deliberations came up with essentially the same recommendations which were presented to a meeting in Torremolinos in

Plastics and aluminium as boat building materials were covered in several papers. The Italian architect and engineer, Nervi, had, during World War 2, developed a concrete-mix with which to build boats, and his paper describing this had been rediscovered by boat builders in the UK and New Zealand, who had started to use this material for fishing boats as well. FAO started collaboration with Nervi.

Actually at the end of the 60s it had become quite popular to arrange International meetings to discuss advances in fishing vessel design. There were meetings in Triesto, Italy; Montreal, Canada; Copenhagen, Denmark, and the Gold Coast, Australia, to mention a few. Also, some text books on fishing vessel design were published.

The 70s

FAO's planned Fourth Fishing Boat Congress in 1971 was not organized because FAO's governing body in the field of fisheries, the so-called Committee of Fisheries (COFI) felt that so many other organizations would gladly organize one. The Canadians continued with a number of important meetings on several aspects such as materials and winches. FAO organized smaller meetings on subjects like beach fishing.

The 70s was characterized by the implementation of National fishing zones, the so-called EEZ's. Also great increases in fuel prices scared people off from investing in fishing craft. The result was that many long distance fishing fleets disappeared. In some countries a few vessels were built for medium distance fishing.

The 80s

This decade was characterized by the energy problems and a general lack of initiative and imagination for fishing vessel improvements. Fishing vessel yards were closed down.

There was some positive thinking about the possibilities of using wind power to economize on liquid fuels. An important outcome of this was the reactivation of Flettner's proposal to utilize the Magnus effect for propulsion by wind. Also a suggestion to use the Magnus effect for rudders and propeller blades.

Aluminium caught on, ferro cement fell into oblivion. Working decks were covered in, so that the crews could work more independently of the weather. Cranes and mechanized rollers came more and more into use.

With advanced catch technology and sophisticated electronic fish detection apparatus, great quantities of fish were now caught so that many species suffered from overfishing.

Echo sounders and sonars had gone through extensive improvements in performance. A whole range of fish finders had ridden in on the waves of micro-processor technology. Electronic circuitry and signal processing had undergone great changes. The information was processed and presented in an easily understandable way for the operators, the fishermen.

With an understanding of the acoustic principles and the physical limitations such as beam-width, frequencies, source level, noise level, target strength, etc., a new generation of fish detection apparatus had developed: echo sounders, sonars and trawl mounted equipment.

The echo sounder now had functions such as scale expanders, dual frequency, trawl watch and operational memory. It became possible to determine whether the size of the fish was large enough to be worth fishing or not.

The use of color became common and it permitted a wider range than the paper recorder. The colors made it easier to interpret echo strength. Fish of a certain size could be recorded by the same color at all depths by the use of receivers with accurate time varied gain (TVG).

With the trawl instruments it was possible to monitor the sinking or raising of the trawl relative to bottom and surface due to the speed of the trawler, to observe the gap of the trawl and whether fish were entering or not. With a special catch indicator one obtained quantitative information of the catch.

To the original searchlight sonars were added omni-sonars and multi-beam sonars. They covered a wider sector in shorter time and had a very high source level and long detection range. Some such sonars were equipped with automatic target tracing, even several fish schools could be registered at the same time. The sonar's computer calculated the course and the speed of the schools which was displayed in true motion relative to the vessel itself.

At mid-80s a review was made of the 1959-predictions, e.g., what had happened after 25 rather than 15 years. The following developments had then taken place:

- Asdic - or sonar as it had then been renamed
- Use of light in fish attraction
- Pumps to transfer heavy catches into the hold
- Advanced net design
- Underwater television
- Winding the gear directly on the winches
- Storage by chilled sea water
- New materials such as plastic and aluminium

But many things had not yet materialized, and were perhaps not to come:

- Use of electricity and vibrations for attraction
- Use of anti-biotics and radiation for storate
- Use of rubber for construction
- Transfer of crews and cargoes by air craft
- Fishing under the ice by submarines
- Artificial upwelling by nuclear heating
- New power plants(such as gas turbines,nuclear,Wankel principle)

As far as the general recommendation that fishing boat designers should follow developments in their field, there is a great improvement here thanks to organizations like FAO and the technical journals which do much to disseminate information. Also, makers of engines, acoustic instruments and net makers do much to help the designers improve their technical knowledge. A major FAO book entitled: Fishing Boat Design made it possible for trained engineers without specialized knowledge of fishing vessel design to design efficient ones for the future.

The predicted use of hydrofoils, hovercraft and catamarans had not come about. The power requirements of hydrofoils and hovercraft seemed to have appeared too excessive for such a low priced commodity as fish.

The possibility of using aircraft, especially helicopters, was not utilized in spite of their popularity in connection with oil exploration. Aircraft could also have been used for acoustic detection - and perhaps carrying a light net for imprisoning catches until catcher craft could come.

The idea of locating fish under the ice cap had not,as far as known,been tested.

The ideas of using containers and other recent cargo handling equipment such as fork lift trucks came slowly into use by some progressive fishermen who also had started to use pallets.

Anti-rolling tanks were coming into use. However little of the research published on how to design less resistant and more seakindly fishing vessels had been utilized by practising fishing vessel designers; they had apparently hoped for some prototype to confirm the validity of the suggested possibilities.

In 1959 one talked about nuclear propulsion of aircraft - so something similar was suggested for fishing vessels. This was 'out' because of the widespread resistance against nuclear power production - and also because no such plants were sufficiently light. There were several suggestions for the revival of steam. And the advocates of the sterling principle of external combustion engines worked hard to get this principle adopted by the automobile industry.

The ideas of using inflatable rubber catcher vessels might still materialize.

There was no awareness in 1959 that the price of fuel would increase so much. Other factors that influenced the situation in the mid-80s were:-

- Cost of labor (requiring further automation and mechanization)
- Shorter trips to improve quality of catches
- Demand by the crews for more comfort (both while fishing and while not fishing) and shorter trips

At the end of the 80s owners and fishermen were still very reluctant to obtain technical advice from institutions and consultants. They did not trust the competence of those offering to improve their operations. In some cases, they were actually buying considerable amounts of advice indirectly: when they bought acoustic instruments and other electronic equipment for navigation,they did not quite realize that the main part of the cost was for soft ware and development and that only a small part was for the hardware itself.

The 90s

After the 'quiet 80s' a wave of optimism gave the fishing industry a strong lift at the beginning of the 90s. Population pressure, scarcity of meat and a general understanding that fish was a health food had increased fish prices so much that investments into equipment like vessels was not such a marginal investment as in the 70s and 80s. Also, Governments were now assisting the fishing industry as much as they did agriculture. In order to protect loans and subsidies, they required fishermen to pay openly for soft ware, for the design of vessels and equipment. The Governments also had their own ship research institutes devoting considerable time to fishing vessel developments.

Computer programs were developed with which one could determine the best shape of a vessel both with regard to minimum resistance at the required working speeds and at the same time having sufficient stability and the most agreeable working motions.

With the help of such programs it was now possible to consider new materials, like aluminium and light-weight high tensile fibre reinforced plastics without having (as in the 70s and 80s) to compensate for the lighter construction with ballast, which in many cases introduced impossible ship's movements.

All components became lighter: Sterling-engines did reduce the engine-weights by % without the propeller efficiency suffering because new light-weight reduction gears permitted high reduction ratios, thus low propeller r.p.m.

Remote sensing had become a reality: daily reports were now issued about the catchable fish concentrations, like weather maps. The acoustic instruments carried on board had increased in range and cover at the same time as prices had become comparatively less, so that even a small boat could use the most sophisticated equipment. A single instrument with a small transducer could act as both a high-speed sonar and an echo sounder with the whole range of practical frequencies. Actually, the sounder decided in relation to the target what would be the most effective frequency and it would then report to the operator in easily readable form, such as by print-out, what catches to expect at the fishing power of his specific vessel.

The whole unit would become much more compact than today. Similarly, other electronic devices like the radio would be miniaturized and able to cover all wave lengths,eliminating the need to carry a whole range of receiver/transmitters.

Also some special highly efficient fish attraction devices had come into use which worked with both acoustic signals,light of various colors and frequencies, electricity and vibrations. At the beginning of their use, serious problems arose when competing vessels were 'fighting' for the same schools. However,at the end of the decade, with the scientific management of the resources as a whole,solutions were also found to have vessels share the available schools in a just matter.

The work onboard was made easier with the aids of cranes and,more particularly, with the help of pumps to extract the fish from the nets and place them in the hold and then after landing to move them from the holds to the shore plants.

Individual boxing of fish with ice and the heavy and difficult internal moves of those boxes which had been found to be the best indication of good fish handling in the 80s, became outmoded by these more rational handling methods which also resulted in higher standards of hygiene. The fishing industry did realize that other industries had solved their handling problems more efficiently, such as the potato - and fruit - handling systems, and they took their experiences into account.

Crew accommodation, their feeding and entertainment when off work, became very much improved so that the crews were as well off as those on oil rigs. Several crews to one vessel became standard, no owner could afford keeping vessels idle when a crew was off duty.

Vessels were again built with better proportions between length and beam to ensure best possible behaviour in seaways, especially in head seas.

The Year 2000

To celebrate the new century, a number of the major International organizations with fisheries and oceanographic departments, decided to join forces and to build an unorthodox prototype fishing vessel. These were organizations like FAO's Committee on Fisheries (COFI), Unesco's IOC, EEC, OECD, etc. When ready the prototype will visit all major fishing grounds and fishing ports. Highliner fishermen from those ports will be invited to take command of the vessel to carry out trials to satisfy their curiosity. All operations will be carefully monitored by instruments and visual observations, and progress reports issued at frequent intervals. The trials will be recorded by TV-crews so that the interest of all fishermen can be kept high.

The outline specification of the prototype will roughly be:-

Length	80 feet	24 meter
Beam	21 feet	7 meter
Hull weight, light ship		130 tons
Continuous power		500 horse power
Cruising speed		10 knots
Trawling speed		3-3½ knots
Fuel consumption at 10 knots cruising		
Average fuel consumption		

The hull will have optimum prismatic coefficient, transom stern, bow bulb to reduce the bow wave and thus the resistance, stern bulb to equalize the wake and thus to improve the propeller efficiency, large propeller aperture to permit slow running propeller. The hull will have two stabilizing systems: fins for sailing to and from the fishing grounds and flume tanks to be used primarily when operating at low speed such as when fishing.

Main machinery will be by Sterling engines working through a reduction gear with multiple gear ratios to permit the use of most economic propeller r.p.m., propeller with controllable pitch blades or Flettner rotors,Flettner rotor rudder.

Fish attraction system:

Fish detection system utilizing remote sensing from satellites and sensors from the vessels placed in radio-directed unmanned helicopters, long range combined echo sounder and sonar with multiple frequencies and print-out facilities.

All-wave length radio transmitter/receiver

Multi-frequency radar, 3 - 10 cm

Navigation system utilizing radio-beacons, Loran, Decca and the global positioning system NAVSTAR.

WRITTEN CONTRIBUTION

Future Development in Fishing Vessels

Bruce Culver

Fishery utilization will tend to increase over the rest of the century. Nations with large stocks of fish within their own contiguous zones will see their industry enhanced, others will decline.

Specific types of vessels to be built in the future will continue to be heavily dependent on local conditions, and will vary substantially from one part of the world to another.

Plastic will see increased use in small vessels. Aluminum is useful for small boats, but its disproportionate increase in price is already discouraging its use in larger vessels, even as a deckhouse material. Boats of more than 30 meters length or so will continue to be built of steel.

The diesel engine will remain the prime power source, tending to light weight geared designs. Use of low grade fuels will increase dramatically. Fuel cost optimization will be important.

Methods of processing on board and preservation of catch will be important areas of development and will probably produce the most radical changes.

Marketing will be important, particularly to American producers. New products such as imitation shellfish produced from traditional Japanese surimi will open new and potentially lucrative markets.

Electronics continue to improve communication, controls, and navigation as well as fish finding.

PART II

FISHING VESSEL OPERATIONS

WATERFRONT USE CONFLICTS AND THEIR EFFECTS ON COMMERCIAL FISHING OPERATIONS IN FLORIDA

Leigh Taylor Johnson
Florida Sea Grant Extension Program

Abstract

Commercial fishing vessel operators increasingly face challenges to their use of the waterfront. These problems can become acute when changes are planned for waterfront traditionally used by commercial fishermen and seafood dealers. Five major trends contributing to competition for limited waterfront access points are population growth, waterfront residential development, pleasure boat proliferation, environmental constraints on development of waterfront facilities and growth of deep draft shipping in some areas. Efforts are underway and have succeeded in some places to find feasible alternatives where commercial fishing vessels have lost traditional waterfront facilities. Commercial fishermen and seafood dealers who buy from them should become aware of local trends which may affect future waterfront access, establish good communication with government agencies, and begin work where necessary to ensure that adequate facilities will be available for future docking and off-loading operations.

Introduction

Commercial fishing vessel operators in Florida increasingly face challenges to their use of the waterfront. Although problems of waterfront access seem remote from daily operations, they can become acute when changes are planned for waterfront traditionally used by commercial fishermen and seafood dealers. This report will review sources and examples of waterfront use conflicts affecting commercial fishing operations in Florida and describe ways in which some of them are being resolved. It should help to alert the seafood industry to trends which may affect their operations and to means of seeking solutions to problems which may arise.

Sources and Examples of Waterfront Competition

Five major trends are causing increased competition for limited waterfront access points in Florida. They are population growth, residential development in coastal areas, pleasure boat proliferation, environmental constraints on development of waterfront facilities, and growth in deep draft shipping in some areas. Population growth is the driving factor for much of this change and is predicted to continue well into the next century.

Population Growth

Florida's population has doubled during the past twenty years, rising from 4,951,560 in 1960 to 9,746,324 in 1980, according to the United States Bureau of the Census (U. Fla., 1982) and is projected to reach 14 to 25 million by 2020 (Terhune, 1982). The University of Florida's Bureau of Economic and Business Research estimates an additional 1000 people per day were added to Florida's population in the year from April 1, 1980 to April 1, 1981. The 35 coastal counties contain 7,664,458 people, or 79% of the state's population. (U. Fla. 1982) Thus, the majority of the people are squeezed into the coastal zone (Figure 1).

Residential Development in Waterfront Areas

The rate of growth in waterfront areas of a coastal county may exceed its overall growth rate. For example, the number of electrical meters connected in the barrier island cities of Brevard County in east central Florida increased from 11,058 in 1970 to 17,383 in 1979. However, the total number of meters in the county grew from 69,020 to 98,889 in the same period. (Wentworth, 1982) Thus, barrier island development occurred at a rate of 57% compared to only 43% for the county as a whole.

Florida Department of Commerce statistics for 1970-1979 show that net migration accounted for 91% of the state's population growth (FDOC, 1980). In fact, for the recent year 1980 - 1981, net migration was responsible for 93% of Florida's growth (Terhune, 1982). This trend suggests that many new residents may not be familiar with traditional commercial fishing industries. Also, the atmosphere of urban and suburban neighborhoods is very different from that of rural communities, which are directly dependent on land or sea for prosperity. These factors may account for some of the conflicts which have occurred between commercial fishing operations and coastal residents.

The April, 1984 issue of National Fisherman magazine discusses the effect of special dock and commercial fishing license ordinances in Pinellas County, an urban, peninsular county which lies between northwest Tampa Bay and the Gulf of Mexico. The dock ordinance prohibits the netting of fish, except by cast net, within 100 feet of a public or private dock. The article explains that docks of residential homes line 100 miles of the Pinellas County bayshore, so that 80% of it is closed to commercial gill net fishermen. This restricts the mullet gill net fishery, because mullet prefer shallow waters close to shore. The 1983 Florida legislative session established a $300 commercial fishing license for Pinellas County, whereas the state's saltwater product license is only $25 for state residents, or $100 for non-residents.

The Florida Keys are a 100 mile long traditional fishing community which is changing in character as land is developed for retirement homes, weekend retreats, and resorts. Commercial fishermen in many areas of the Keys have found it convenient to live along a canal, moor at home, and work on gear in the backyard. However, they now face competition for waterfront property, as well as pressure from other landowners who prefer a suburban or resort atmosphere.

Monroe County includes the Florida Keys, has 3768 registered commercial boats, and leads the state in fishery landings (NMFS, 1982). In 1980, the county enacted an ordinance regulating work on fish nets and fish, lobster, and crab traps in some residential and business

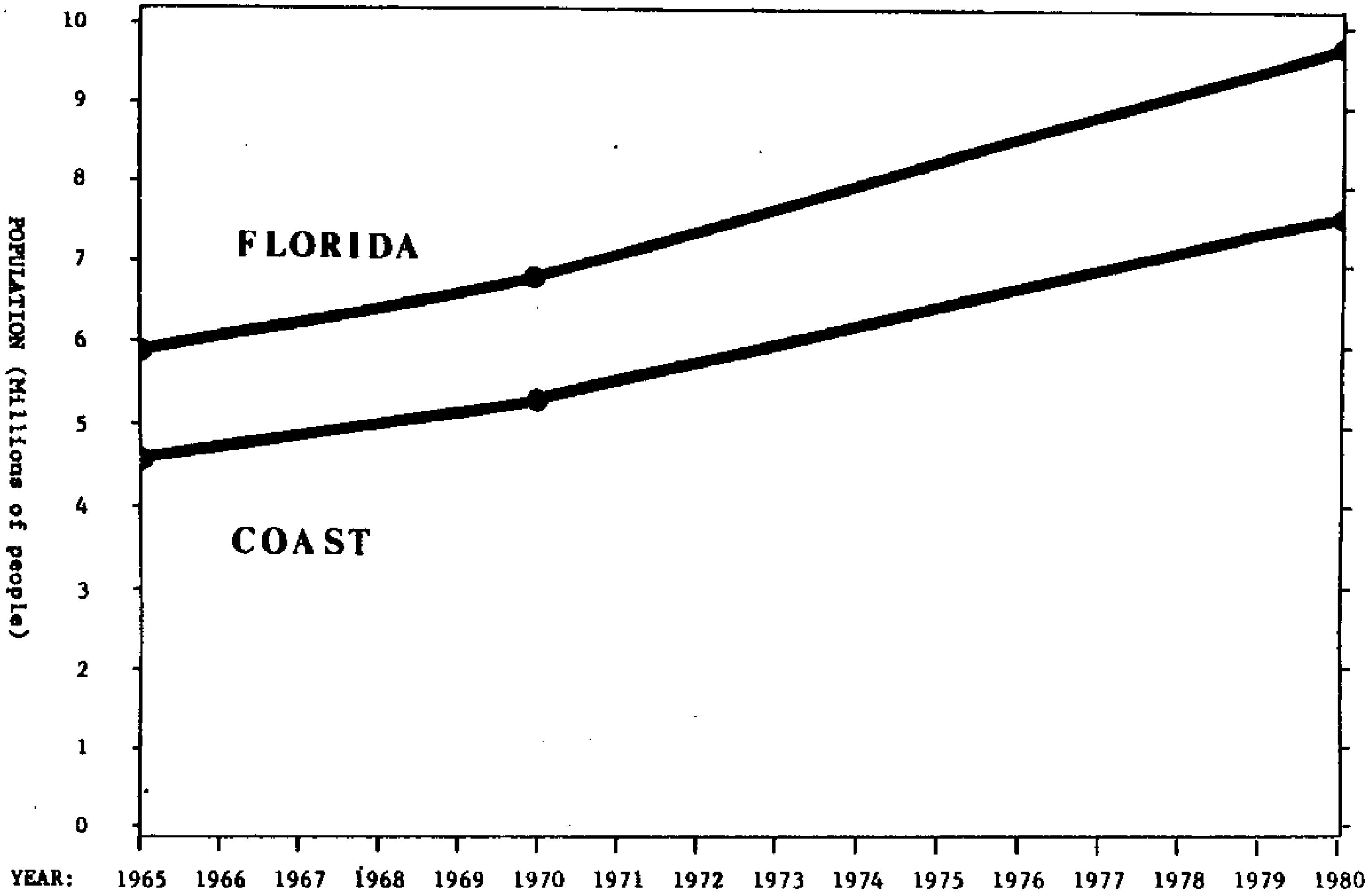

Figure 1. Population growth in Florida and in coastal counties of Florida, 1965 - 1980.

zoning districts. New fishermen and those unable to prove they qualify for exception under a grandfather clause may not conduct some or all fishing gear related activities at home. Because land is scarce, expensive, and in demand for residential and tourist development, finding reasonable alternatives can be difficult.

Pleasure Boat Proliferation

The number of pleasure boats registered in Florida grew from 128,723 in 1965 to 466,775 in 1980, whereas the number of commercially registered boats grew from 27,608 to 31,116 in the same period (FDNR, 1965 - 1980). Figure 2 illustrates this difference in growth rates and Table 1 lists actual numbers of boats registered from 1965 to 1980. The anomaly between 1974 and 1975 pleasure boat registration statistics occurred because boats with an engine of less than 10 horsepower were not required to be registered before 1975 (Cato and Mathis, 1979).

Extrapolating population statistics for 1960 and 1970 produces estimated 1965 populations of 5,871,489 for Florida and 4,607,203 for the coastal counties (Figure 1). Comparing boat registration and population data for the fifteen year period 1965 to 1980, it is evident that the state's population increased by 66%, the number of commercially registered boats increased by 13%, but the number of registered pleasure boats increased by 263%!

The increase in pleasure boats has created a demand for slips and launching sites. For example, in southern Brevard County a marina located on the shore of the prime hard shell clam harvest area has been converted recently from a quiet facility serving both commercial and pleasure boats to a plush anchorage specializing in fishing tournaments. The loss of some mooring facilities and the boom in the hard shell clam fishery in south Brevard County have combined to force many commercial clammers to launch directly from the banks of the Indian River. This practice has drawn complaints from communities which fear it will erode the shoreline.

The 12 square mile area designated Body "F" by the Florida DNR supports an estimated 300 to 400 hard clam fishermen, so the competition for waterfront access is intense. A fishing camp which has allowed commercial clam fishermen to launch has found their unloading activities sometimes tie up facilities needed for sport fishermen. The county has built a new launching and docking facility in the area with funds from the Florida Motorboat Revolving Trust Fund. However, parking spaces are limited, so commercial and pleasure boats must compete for them.

Another result of pleasure boat proliferation has been conversion of marinas to private facilities associated with condominium developments. Of 12 marinas in the Melbourne-Palm Bay area of south Brevard County, three have been recently converted to "dockaminiums". Boating Magazine reported a shortfall of 2000 to 3000 slips in Dade County (Miami area) in a 1983 article, which suggested that buying a condominium might be the only reliable way to get a good spot for a new

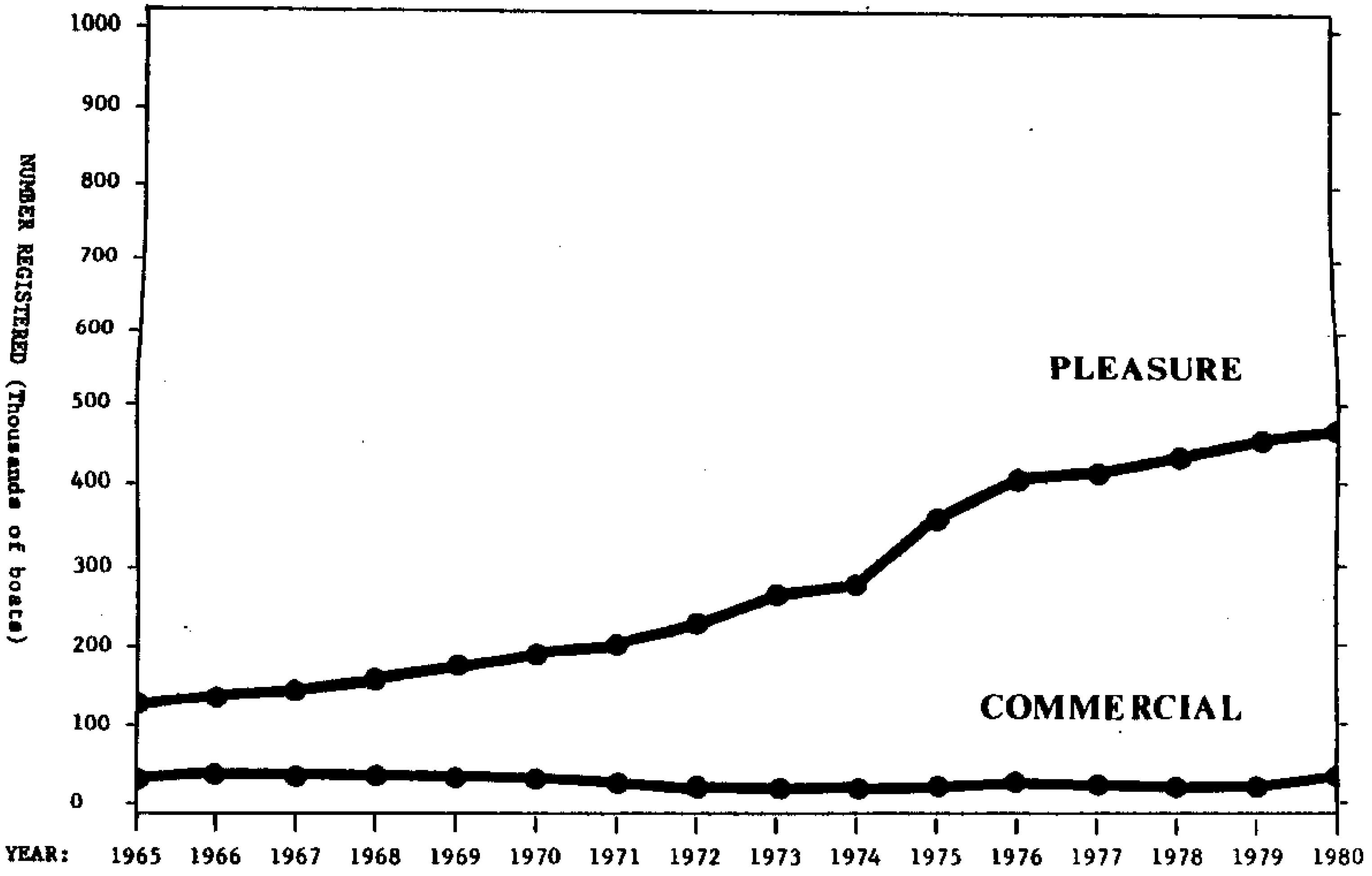

Figure 2. Growth in number of commercial and pleasure boats registered in Florida, 1965 - 1980.

Table 1. Number of pleasure and commercial boats registered in Florida, 1965 - 1980

Year	Number of Boats Registered	
	Pleasure Boats	Commercial Boats
1965	128,723	27,608
1966	136,706	32,927
1967	149,663	31,858
1968	164,875	30,490
1969	177,212	28,183
1970	192,554	29,065
1971	208,096	27,197
1972	229,426	24,962
1973	249,219	23,813
1974	276,112	21,782
1975	360,390	22,566
1976	409,564	26,784
1977	422,398	24,636
1978	434,818	24,805
1979	456,038	24,918
1980	466,775	31,116

boat.

Construction of marinas is expensive, requires a lengthy and complex permitting process, and is dependent on location of suitable land which is increasingly scarce. Because the return on total assets to the average marina in Florida in 1981 ranged from 7.4% to a negative 7.5% with the median less than 1.0% (Milon, et.al., 1983), there is not a strong incentive to develop waterfront for affordable, public marinas.

When pleasure boat marinas are adjacent to commercial fishing vessel facilities and seafood processing plants, conflicts can arise. At Port Canaveral, pleasure boat marinas mingle along the waterfront with scallop, shrimp, and fish processing plants, where fishing vessels unload along the bulkhead. These activities generate noise, odors, and a hectic appearance which are at odds with the resort ambience preferred by marina clientele.

Environmental Constraints

Increased attention to protection of the Florida marine environment is another factor constraining development of marinas which might replace lost commercial fishing vessel anchorages. The Florida Department of Environmental Regulation and the U.S. Army Corps of Engineers have strict policies regulating dredge and fill operations, which are necessary to construct and maintain basins of sufficient depth. Finding appropriate sites for disposal of wet, silty, salty spoil material is difficult, because coastal land is expensive and intensively used.

Bulkheading eliminates shallow water habitats which support many marine species and has a negative impact on fishery productivity. Approved shellfish harvesting areas are few, and marina developers proposing to build in or near these waters face opposition from shellfishermen and review by the Florida DNR's Shellfish Environmental Assessment Section. Would be marina developers must also consider whether the proposed basin would be in a state Aquatic Preserve, a Manatee Sanctuary, or an Outstanding Florida Water. Appropriate stormwater drainage or retention facilities must be constructed, and hazardous waste disposal guidelines and local zoning ordinances must be observed.

Much of the Indian River in south Brevard County is in the Body "F" shellfish harvesting area. A preliminary economic survey by the Florida DNR recently reported to the Florida Marine Fisheries Commission (Berrigan, 1984), estimated the ex-vessel value of Body "F"'s booming hard clam fishery to be between five and fifteen million dollars a year. Although docking, off-loading, and maintenance facilities are needed by the 300 to 400 fishermen participating in this fishery, their construction and operation could degrade the water quality in Body "F" to the point where it would lose its "Conditionally Approved" status.

In central Brevard County, a developer has received approval from the Canaveral Port Authority to lease waterfront access to the Barge Canal crossing Merritt Island and providing access for boats from Port Canaveral to the Intra-Coastal Waterway in the Indian River. The developer faces a lengthy permit application process before he can build a commercial boat repair yard serving vessels with a draft less than 12 feet. Other shallow draft facilities may be needed along the Barge Canal in the near future as the Port Authority implements its plan to increase the percentage of waterfront committed to deep draft shipping.

Deep Draft Shipping

It is hard for ports to justify retaining deep draft docking areas for the relatively shallow draft vessels used in commercial fishing operations. When a deep draft vessel must use a shallow water port, it cannot carry a full load, which considerably increases cargo transportation costs. Maintaining bulkheads and other facilities is expensive, so ports must maximize revenues in order to keep up with the trend toward deeper draft cargo ships. Revenue from commercial fishing vessels and commercial seafood companies is generally lower than that from industrial shippers.

At Port Canaveral, much waterfront is owned by the military, which limits space for civilian use. The Port is a deep draft (35 feet) facility and the Port Authority plans to make more waterfront available to deep draft shipping, such as cargo ships and cruise liners. This will eventually displace a seafood company, a recreational marina, and some charter and party boats. It will also utilize much of the mooring area used by the commercial scallop, shrimp, and finfish fleets between trips and during storms. Seafood dealers estimate 100 to 150 boats may be in port during stormy weather at peak fishing periods.

Like Port Canaveral, Port Tampa is a deep draft facility. Its channel is being dredged from 34 feet to 43 feet deep, in order to accommodate larger, modern vessels. Phosphate ships currently must carry light loads, but should be able to ship full when the dredging is complete. Tampa has been home port for a commercial shrimping fleet of 70 to 80 vessels for 30 to 35 years. Four major companies purchase shrimp from these boats and processors handle both domestic and imported products. However, in 1977 the shrimp fleet was displaced when a shipyard needed space for expansion. The story of the new Tampa Shrimp Dock is an example of a successful effort to find an alternative facility for commercial fishing vessel operations.

Examples of Conflict Resolution

Port Tampa

When the Tampa shrimp fleet faced displacement from their docking area in 1977, the Administrative Services Director of the Port, Mr. Thomas O'Connor, undertook a relocation project. The U.S. Economic Development Administration granted three million dollars and Hillsborough County matched those funds with another two million dollars for construction of a new shrimp facility.

The new Tampa Shrimp Dock was begun in 1979 and opened in 1981. It features four finger piers, four seafood companies, two marine suppliers, a boat lift, repair yards, and some open bulkhead for maintenance activities requiring the boat to be tied alongside a broad bulkhead area. The four seafood companies are responsible for managing the piers and the open bulkhead, saving the Port Authority the expense of supervision.

Successful planning and implementation of this new facility required two years of planning, two years for construction, and careful coordination among funding agencies, the Port Authority, and the seafood companies who were the prospective tenants. The effort has preserved access to the Port of Tampa for commercial fishing vessels, and thus will maintain the possibility for fishery expansion to harvest the many underutilized species found in the Gulf of Mexico.

Port Canaveral

The portion of the commercial fishing fleet which will lose docking space with the expansion of deep draft facilities at Port Canaveral will have access to a commercial fishing dock and boat repair yard at Daytona Beach. The Ponce de Leon Inlet Port Authority is developing a shallow draft facility and would welcome additional vessels. Ponce de Leon Inlet is a reasonable distance from the calico scallop beds located off Cape Canaveral, and could present scallop trawlers an alternative to Port Canaveral without too much extra fuel cost.

Commercial seafood companies at Port Canaveral are working to improve the appearance of their facilities in order to establish an atmosphere more in harmony with neighboring marinas, restaurants, and cruise ships. Modern plants and offices have been built and work areas have been enclosed by attractive fences. The first annual Port Canaveral Seafood Festival successfully served 50,000 people in April, 1984 with the assistance of food, supplies, and labor donated by the seafood companies (Shealy, 1984). Such attention to good public relations will help the seafood industry face future challenges to use of the waterfront.

Fishermen's Pointe

Fishermen's Pointe is a limited cooperative established in Marathon in the Florida Keys to provide commercial fishermen with an alternative to the use of residential lots for construction, storage and maintenance of commercial fishing gear. The Organized Fishermen of Florida, the Florida Department of Community Affairs, the Florida Department of Environmental Regulation, and Monroe County staff worked with local fishermen and attorneys to develop suitable permitting, review standards, and ordinances for this development. One of the fishermen who played a key role in planning Fishermen's Pointe reported that all lots were quickly taken. Clearly, the project has met a need for onshore facilities for commercial fishing operations.

Conclusion

Trends affecting waterfront use for commercial fishing operations continue to pose problems for commercial fishermen and seafood dealers in Florida. Successful conflict resolution requires adequate planning time, funding sources, and good coordination among the seafood industry, government agencies, and developers of marine facilities. New facilities are expensive, appropriate land on which to build them may be hard to find, and permitting procedures are complex and time consuming.

Commercial fishing vessel operators and the seafood dealers who buy from them should evaluate local trends which may affect their operations. They should also establish good communication with government agencies and begin work where necessary to ensure adequate facilities will be available for future docking, maintenance, and off-loading operations.

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A SIMPLE METHOD TO DETERMINE OPTIMUM VESSEL SPEED

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Abstract

Slower speeds save fuel. Do they save money? That depends upon the cost of fuel compared to the value of time. The paper describes a simple method of analyzing individual vessel fuel consumption characteristics from which a rational optimum operating speed may be chosen.

For any incremental reduction in speed, a corresponding increment of extra time required is the price which must be paid to save that incremental amount of fuel.

By measuring or estimating fuel consumption vs. vessel speed, the monetary value of each increment can be calculated and plotted. Optimum speed is then given directly for any value of the operator's time.

Examples are: A 100 ft SNAME trawler, 85 ft and 40 ft optimized fishing vessels from Traung, Doust, and Hayes (FBW 3), and measured data from a 58 ft Alaska seiner, and a 33 ft deep-vee planing hull.

Introduction

Every boat has fuel consumption characteristics which are unique and distinct from all other vessels. Even sister ships will exhibit slightly different characteristics because of differences in displacement, trim, engine condition, etc.

Higher speeds save time but require more fuel. What is the best trade-off between time and fuel costs? The answer can be found using a simple analysis of a boat's individual fuel consumption "fingerprint."

The method is as applicable to a naval architect doing conceptual design as it is to a fisherman trying to decide what RPM to run. It requires knowing only: (1) gallons per hour consumed vs. speed, (2) the cost of fuel per gallon, and (3) the value of one's time.

Fuel consumption must be either measured or predicted over a range of hull speeds. Measured values automatically include complex effects of variations in hull, engine, and propeller efficiencies. Fuel consumption can also be predicted indirectly from horsepower, using either traditional EHP prediction methods or measured RPM values and propeller law assumptions. By assuming values of propulsive efficiency and engine specific fuel consumption, fuel rate is determined.

Optimum speed is inversely related to the price of fuel. The higher the price of fuel, the more it pays to slow down.

Although it's difficult for some fishermen to come up with monetary values for their time, the term "optimum speed" is meaningless otherwise. A good approach is to ask: would I be willing to get where I'm going one hour later if somebody paid me five thousand dollars? How about fifty cents? Then just zero in between those values until it feels right.

Since the value of a person's time (or vessel's time) is different under different conditions, optimum speed is also different under different conditions.

The Method

The key to the analysis is to look at the difference in fuel needed to travel a fixed distance at two different speeds and compare that difference to the difference in travel time. To illustrate, consider the following example for an arbitrary 100 mile trip.

RPM:	1840	1808
Knots:	9.94	9.87
Hours for 100 mile trip:	10.06	10.13
Extra hours needed:		.07 hours
Gallons per hour:	19.0	18.0
Gallons used:	191	182
Gallons saved:		9.0 gallons

Reducing engine speed 32 RPM adds a little over four minutes to a 100 mile trip, but saves nine gallons of fuel. That is equivalent to saving 128 gallons of fuel for every extra hour taken. The results are the same no matter what distance is chosen. The VALUE of traveling at the lower speed is 128 gallons per hour. Assuming fuel at $1.00/gallon, the vessel's time would have to be worth at least $128/hour to justify traveling at the higher speed.

If VALUES are calculated for increments at successively lower speeds, eventually the point will be reached where it is no longer worth one's time to go any slower. That speed is the "optimum" or most cost effective speed. Going faster uses too much fuel, and going slower takes too much time.

In this paper, effects of incremental <u>reductions</u> in speed are analyzed. Marginal speed increases can just as well be used.

A Typical Example

The previous illustration used measured data from a 335 HP, 58 ft Alaska seiner, displacing 130 tons, towing a seine skiff. VALUES for increments at other speeds are shown graphically in Figure 1 along with curves showing gal/hour and miles/gal vs. speed. Looking at the value curve gives some interesting insight into the economical operation of this boat:

If the fisherman's time is worth about $20/hour, the optimimum speed to run would be 1500 RPM (8.7 knots). Note that nothing in the shape of either the GPH or MPG curves could produce the same conclusion. One might have suspected that 9.2 knots (1600 RPM) was the "best" speed, since fuel consumption rises sharply at higher speeds. In reality, it would only be the best (optimum) speed if time is worth $65/hr (assuming $1.00/gal).

Note that it doesn't make sense to run anywhere between 7 and 8 1/2 knots. When time is worth more than about 16 gal/hour, speed should be above 8.5 knots. When time is worth less than about 16 gal/hour, speed should be less than 7 knots.

As previously shown, time must be very valuable to justify operating at the higher speeds. As much as 128 gallons of fuel can be saved for each extra hour running, simply by slowing slightly from full throttle. That may seem extraordinary since the engine's maximum fuel consumption is 19.0 GPH, but is typical of vessels analyzed.

A Towed Model Example

In addition to being able to determine optimum operating speeds for existing vessels, the technique is useful during design and evaluation stages. Model tests of SNAME Trawler Model W-8 (sheet # 169) are used to demonstrate another example. Waterline length is 103 feet, beam is 22 feet, and displacement is 300 tons.

Some assumptions are first necessary to convert predicted EHP to expected fuel consumption. It is expedient and reasonably accurate to assume constant values of .5 for overall propulsive efficiency, and 18 hp per gph for thermodynamic efficiency of a typical four-cycle diesel engine (BSFC=.39 lb/hp-hr). In other words, for displacement boats, dividing EHP by 9.0 gives reasonable estimates of fuel consumption in gal/hr. The increase in partial load specific fuel consumption at part throttle is somewhat offset by an increase in propeller efficiency.

In Figure 2 are plotted the original EHP data along with VALUE and MPG vs. speed curves. Again, several conclusions may be drawn which would not be obvious from either the EHP or miles/gal curves: Time would have to be worth 900 gal/hr to justify running free at 14 knots. A big change in efficiency occurs just below 12 knots. Optimum speed is 9.2 knots when time is worth 50 gal/hr and 11.7 knots at 100 gal/hr. Obviously, many other statements could be made regarding economical powering or operational decisions.

Two FAO Optimized Hulls

As a result of a regression analysis of resistance characteristics of many fishing vessels, Traung, Doust, and Hayes developed four optimized low-resistance hulls. The results were published in "Fishing Boats of the World: 3" and presented at the Third FAO Fishing Boat Congress in 1965. To further demonstrate this method of analysis, the largest (85 ft) and smallest (40 ft) were chosen.

As before, fuel consumption in gal/hr was assumed to be equal to EHP divided by 9.0. The results for the 85 footer are shown in Figure 3. Some observations are: from 8.3 to 10.3 knots, the relationship between EHP and speed is nearly linear. If time is worth 20 gal/hr, it pays to run at 9.8 knots instead of any slower. Time value has to double to 40 gal/hr to justify the .4 knot increase from 9.8 to 10.2 knots. If time is only worth 10 gal/hr, 8 knots is optimum rather than 9.2 knots (since VALUE is greater than 10 gal/hr between 8 and 9.2 knots).

The results for the 40 footer are similarly plotted in Figure 4. Several comments regarding the 40 footer are: resistance is virtually constant from 5.7 to 7.3 knots giving constant miles/gal over the same range. The effect makes the VALUE zero over the range, meaning no matter how little time is worth, it never pays to run between 5.7 and 6.8 knots.

Since the boat is so easily powered, it is seen that speeds up to about 7 knots are extremely economical. Even at a time VALUE of 1 gal/hr, it doesn't pay to slow below 7 knots.

A Planing Boat

The previous examples utilized computer predicted resistance, model test resistance, and measured fuel consumption for several displacement vessels. The analysis is general in nature and is equally applicable to any mode of transportation such as automobile or aircraft.

The final example uses measured fuel consumption on a 33 ft deep-vee sport fisherman with twin turbocharged 270 hp, V-8 diesels, capable of 28 knots. Beam is 12.7 ft, deadrise is 17 1/2 degrees, and displacement is 19,900 pounds.

In this case, a propulsive efficiency of .55 and an engine efficiency of 18 hp/gph were assumed so EHP and resistance could be estimated. This was done only for discussion's sake since only GPH vs. speed is required to calculate VALUE. The resistance, GPH, MPG, and VALUE curves are shown in Figure 5.

The slope of the resistance curve is seen to be moderate from hump speed at about 10 knots to about 25 knots. This corresponds to moderate VALUES over that range, meaning it doesn't save much fuel to slow down in that range.

Above 25 knots resistance increases sharply, making VALUE increase to 50 gal/hr. This means that it is very worthwhile to run at 25 rather than 27 knots. Good places to run this boat would be 8, 17, or 25 knots. Poor places would be 10, 20 and 27 knots.

If this operator's time were worth 10 gal/hr, 23 knots would be the best trade-off of fuel for his time (even though miles per gallon are better at 17 knots).

Propeller Law Estimates

For existing vessels, if a fuel flow meter is not available, reasonably accurate estimates of fuel consumption can be made using tachometer readings.

Maximum fuel rate can be obtained either from manufacturer's data or estimated from maximum horsepower, using one GPH for each 18 hp. A more accurate value can often be derived from engine sales literature.

For displacement vessels, fuel consumption at lower RPM can be estimated by assuming a 3.0 (cubic) propeller law:

$$\text{GPH} = \left[\frac{\text{RPM}}{\text{max RPM}}\right]^{3} \times \text{max GPH}$$

For planing hulls, a 1.9 power relationship can be used:

$$\text{GPH} = \left[\frac{\text{RPM}}{\text{max RPM}}\right]^{1.9} \times \text{max GPH}$$

Figure 6 compares measured to estimated fuel consumption for the 58 ft seiner and the 33 ft planing boat examples. Differences in the two methods are seen to be small.

Summary

An analysis of the relationship between speed and fuel consumption is essential to both responsible new design and intelligent operation of existing vessels.

The proposed method is simple and uses either: (1) theoretically or empirically predicted EHP or resistance data, (2) measured fuel consumption, or (3) estimated fuel consumption using measured tachometer data.

VALUE is an indicator of the slope of a vessel's resistance curve in terms of the value of time. A negative VALUE means resistance is increasing as speed decreases, so a lower speed is pointless. A VALUE near zero means resistance and MPG are approximately constant.

If VALUE keeps rising as speed decreases, it pays to look at even lower speeds. Peaks in the VALUE curve are good spots to run (or design to) when speed is important.

If there is more than one possible speed for a given VALUE, use the lower speed if the curve has a maximum between the two possible speeds. Use the higher speed if the curve has a minimum between the two speeds.

The current value of one's time (or vessel's time) in equivalent gallons per hour determines the associated optimum speed directly from the VALUE curve. Other speeds simply do not economically balance time against money.

Figure 1

58 Ft Alaska Seiner

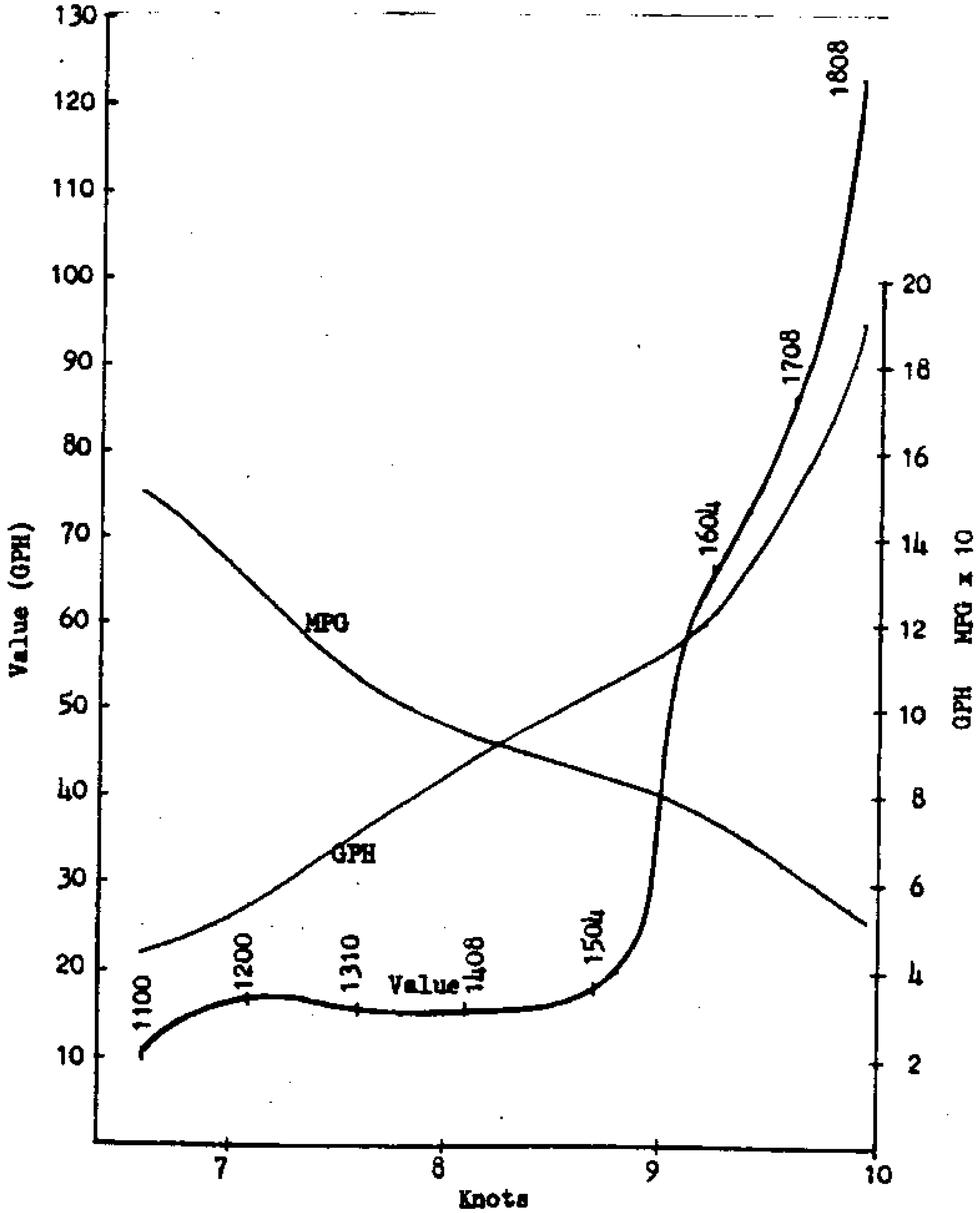


Figure 2

100 Ft SNAME
Trawler

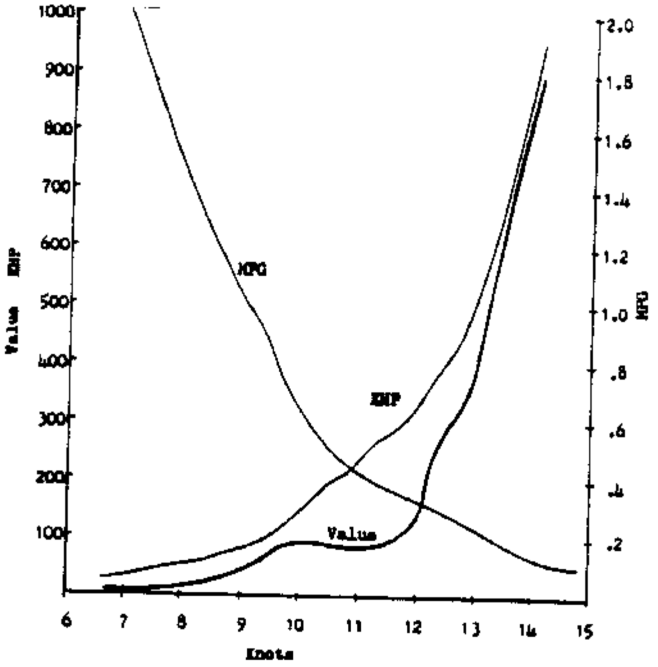


Figure 3

85 Ft FAO
Optimised Hull

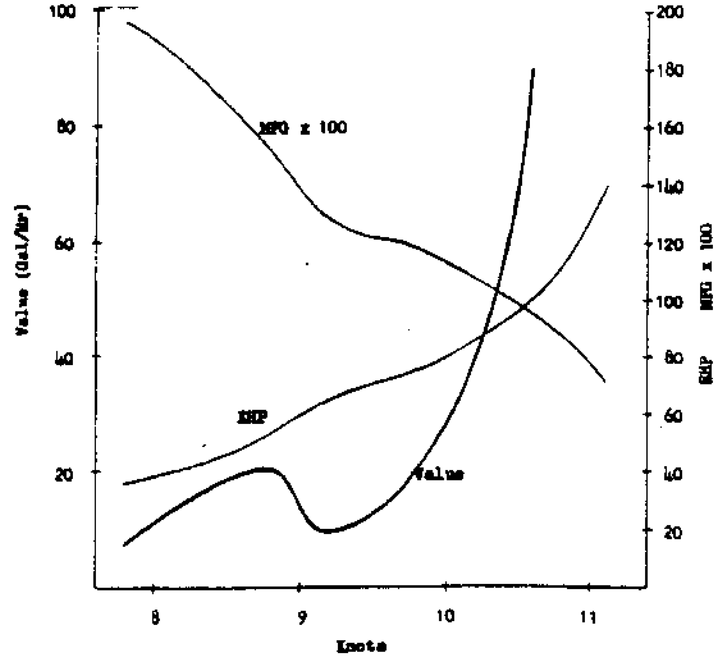


Figure 4
40 Ft PAO
Optimized Hull

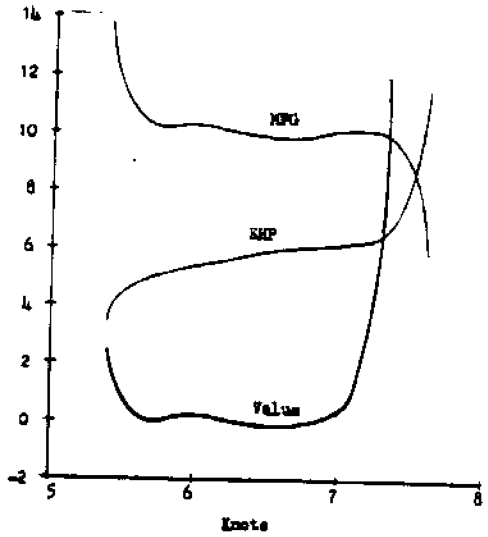


Figure 5
33 Ft Planing
Deep-Vee

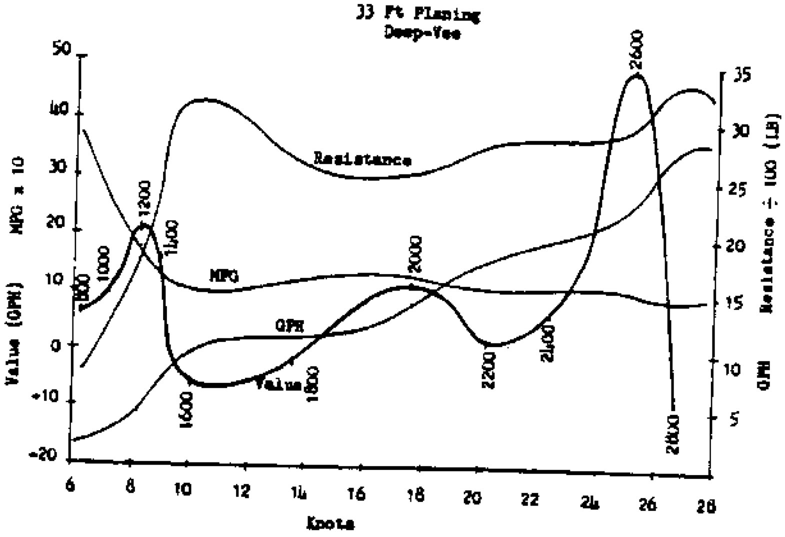
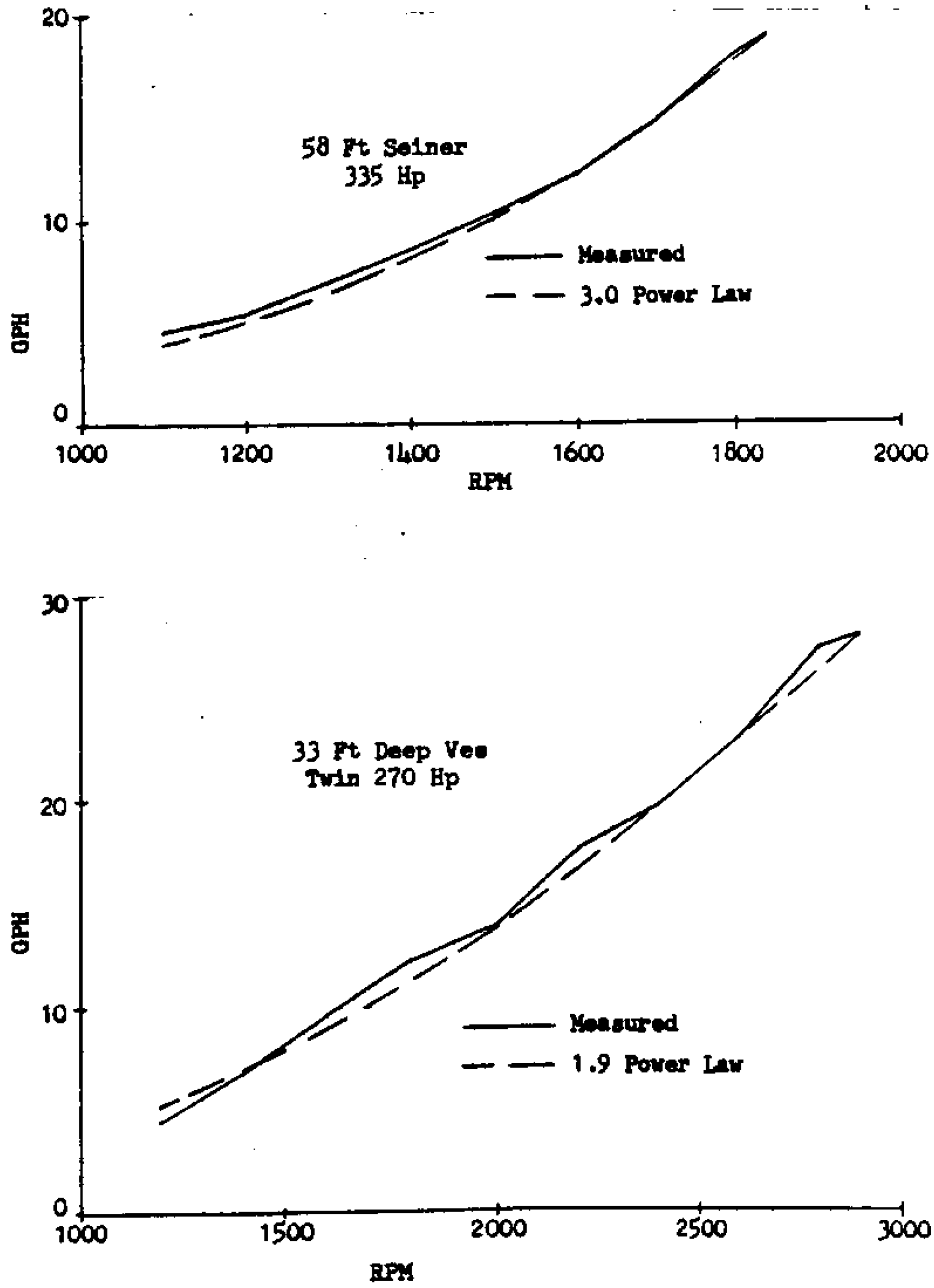


Figure 6


58 Ft Seiner
335 Hp
Measured
3.0 Power Law
GPH
0
10
20
1000
1200
1400
1600
1800
2000
RPM
33 Ft Deep Vee
Twin 270 Hp
Measured
1.9 Power Law
GPH
0
10
20
30
1000
1500
2000
2500
3000
RPM

THE DEVELOPMENT OF AN OFF-BOTTOM SHRIMP TRAWL

PRELIMINARY REPORT

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ABSTRACT:

This paper presents the preliminary results of an effort to develop an off-bottom trawl for Gulf of Maine shrimp, Pandalus borealis. This fishery is seasonal, typically from December through April, and is an important near-water activity for nearly 200 inshore vessels from Maine to Massachusetts. Present techniques of bottom trawling are plagued by damage to the groundfish resources and ineffectiveness at night.

These shrimp move up at night, beyond the reach of even the highest opening trawls. An off-bottom trawl has been constructed which could potentially solve both the by-catch problem and have far greater effectiveness with more hours of potential operation than bottom trawls.

A 26' by 13' rigid frame was used to support a four seam net of 2"- #15 nylon webbing. The frame is towed by twelve wire-rope bridles coming together for attachment to one or two towing warps.

Preliminary tests have shown the rig is easy to handle and has a drag far less than conventional gear. Results of those engineering trials are presented along with the details of the trawl design.

The concept has shown potential and fishing trials are planned for the 1984-85 season.

INTRODUCTION:

Gulf of Maine shrimp, Pendalus borealis, are an important seasonal fishery for coastal Maine, New Hampshire, and the north shores of Massachusetts. Annual landings vary from a high of 5,300 metric tons (mt) in 1975 to 1,000 mt in 1977 to a present level of over 3,000 mt.

This catch is landed by approximately 200 vessels during the winter shrimping season, usually December through April. The rest of the year most of these vessels return to groundfishing. The gear currently being used by these small draggers is small mesh bottom trawls with mesh sizes of 1 3/4" to 2", stretched. This gear is plagued by two serious problems. The first is that the small mesh catches juvenile groundfish, typically first and second year cod, haddock, and flounder, in amounts that concern both the fishermen and resource biologists.

Estimates of this by-catch vary from one third of the haul to well over half. Since most of these fish don't survive, the effect on the groundfish stock is significant. The task of on-deck sorting is also a burden. Often, a brine settling tank is used to float the finfish. This reportedly has a deleterious effect on shrimp quality.

The second problem with the trawls is the fact that the shrimp can be caught only during daylight (nine hours per day during December) since only then are they found on the seabed. During the night and even on heavily overcast days they remain off-bottom, well beyond the reach of the trawls now used.

There is some conjecture about whether this species disperses when up in the water column or whether it remains in schools suitable for midwater trawling. As more skippers use color sounders, many are finding the stripes of color they seek during bottom trawling tend to rise 5 to 15 fathoms at dusk and remain in discrete bands. No serious sampling has been done to determine the composition of these bands; however, many fishermen think it must be shrimp.

The size of vessel which enters winter shrimp fishing is typically between 35 and 55 feet long with under 200 horsepower. This small size, combined with the irregular coastal water and strong tidal currents, prevents the use of conventional single or pair midwater rigs. In addition, most midwater gear is designed for schooling fish which can be effectively herded by large mesh netting. Since shrimp must essentially be filtered from the water, small meshes must be used throughout.

OFF-BOTTOM SHRIMP TRAWL DESIGN:

In cooperation with Portsmouth, New Hampshire fisherman Lyle Chamberlain, an innovative rig was designed which would be easy to handle while offering a reliable mouth opening necessary to evaluate the feasibility of an off-bottom fishery. Through the use of a rigid rectangular frame, the variabilities of horizontal spread and vertical height would be eliminated. Also, the resistance associated with trawl doors and headrope floats would not exist.

The size of the frame was based on what seemed reasonable to handle from Capt. Chamberlain's vessel, the availability of salvaged sailboat mast extrusions, and funding limits. A horizontal width of 26 feet and a vertical height of 13 feet was selected and the frame sections were cut from two 39' extrusions of crossection shown in Figure 1.

Ends were cut at 45 degrees and pre-drilled flanges were welded for assembly. The side and lower frames were provided with 2" diameter vent holes while the upper frame was welded water tight. The buoyancy of the upper section was just sufficient to balance the 200 pound weight of the frame while keeping the frame upright when deployed. A smaller vertical centerline brace was used to add rigidity.

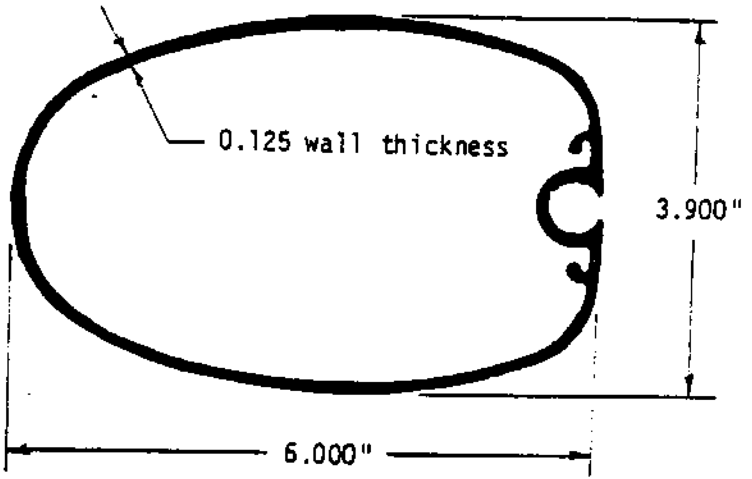

Figure 1. Mast extrusion used for the trawl frame.

The net was designed to be made fast along the trailing edge of the frame. Sail track slugs were placed at one foot intervals along the perimeter rope. A hanging ratio of 0.5 was used with a taper rate of 1B1P in the four-seam portion and 2B1P in the two-seam portion. This produced a net of approximately 50' in length. The net plan is shown in Figure 2 with the panel drawn to shape. The gradual taper of this design may seem unusual compared to conventional shrimp trawls.

It has been reported that the sharp tapers of common shrimp trawls cause the substantial buildup of shrimp and debris against the netting, causing increased resistance.[1] This material often becomes dislodged during a turn and usually gets washed back into the codend during haulback. Since the planned design would not allow such washing back, the long taper was selected.

All gores were made up bunching four meshes from each panel. A 120 mesh codend was used to finish the net construction.

To pull the frame, a bridle arrangement of twelve 3/16" diameter 7x7 galvanized wire rope was used. The number of bridles was based on an attempt to minimize the bending stesses in the frame through close spaced supports. In addition, since one of the objectives of the rig is to eliminate by-catch, the spacing of the wires could effectively herd finfish from the net's path, allowing cleaner catches.

For reasons now unclear, it was decided to tow the gear using both warps. Six bridle wires would be lead to form two apexes. The problem of slack bridles should warps be paid out unequally was remedied by having only four bridles of fixed length with the other eight arranged through sheaves at the apex to allow self adjustment and uniform tension. Computer modeling of the design was used to achieve the configuration presented in Figure 3. The brief program used is included in Appendix I.

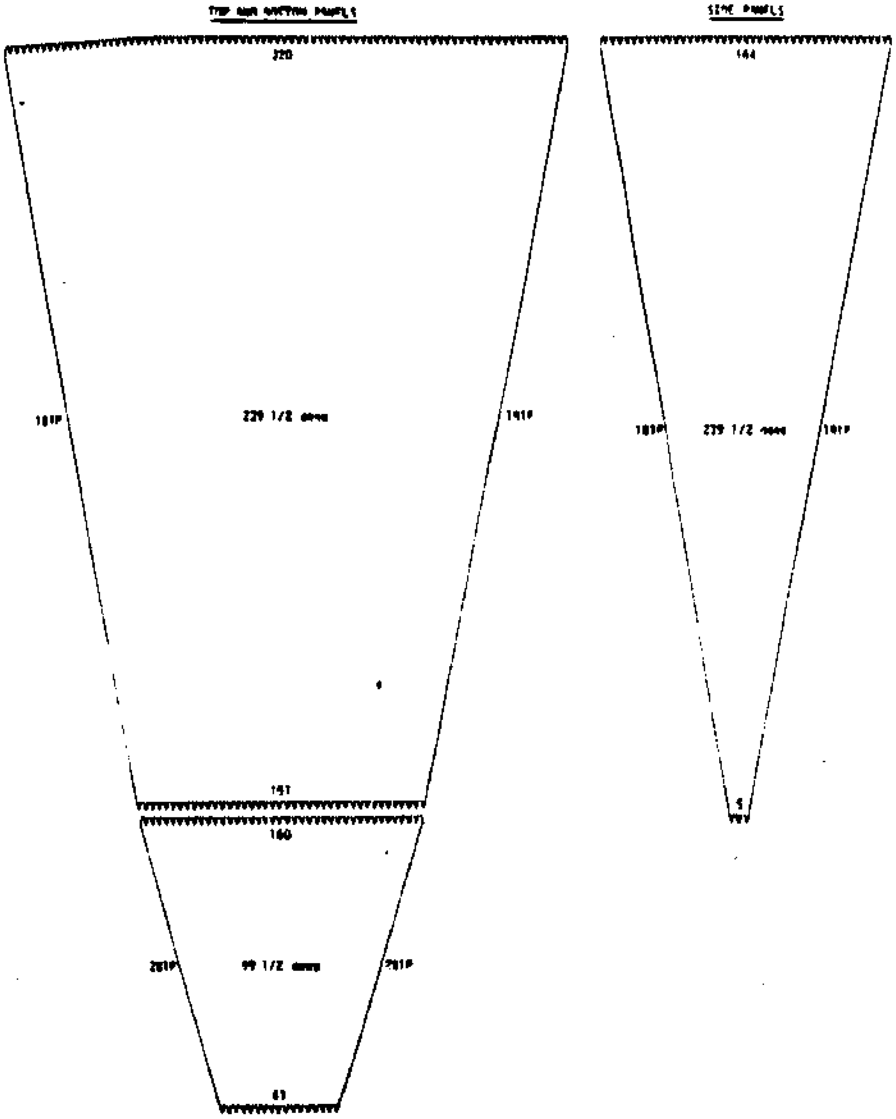

Figure 2. Net plan for off-bottom trawl.

SEA TRIALS:

In October of 1983, the completed rig was assembled aboard the F/V Jayma-Ellen. Preliminary tows off Portsmouth Harbor demonstrated the relative ease of handling of the gear. Initially the rig was launched from along the starboard rail where it was stowed in-port. Subsequent deployments were from the stern where the rig was to be stowed between tows. A bullrope was to be used to pull the codend aboard while maintaning some headway.

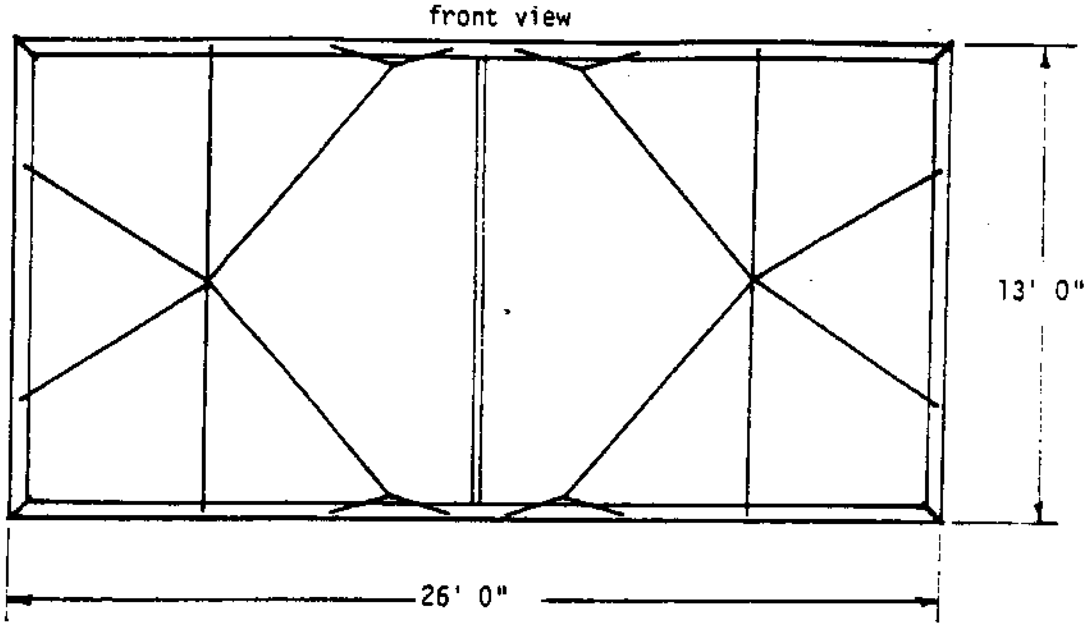

Figure 3. The bridle arrangement to provide uniform loading in spite of any warp length misadjustment.

Due to funding constraints of this project, fishing time could not be supported, therefore a request was submitted to use the gear and sell the shrimp to defray boat expenses. Permission from both New Hampshire and Massachusetts was denied. Commercial trials were therefore postponed until the spring, during the later part of the season when the weather for experimentation would again be acceptable. In early March the F/V Jayma-Ellen was destroyed by fire along with the plans for a commercial demonstration.

To determine the resistance of the trawl, it was transported to Boston for use aboard MIT's research vessel Edgerton. A single warp rigging was used and cable tension was monitored using a dial-indicating dynamometer. Speed through the water was determined with a strut-mounted knotmeter with a yacht-type digital display.

Cable tension measurements were taken with 300' of 5/8" wire paid out. The results are tabulated in Table 1 and presented graphically in Figure 4. The effects of cable resistance have been neglected.

Speed	Declination Angle	Tension	Resistance
1.0	20 degrees	400 pounds	372 pounds
1.9	8	1,200	1,188
2.8	5	2,200	2,191
3.4	4	2,700	2,691

Table 1. Measured data from sea trials.

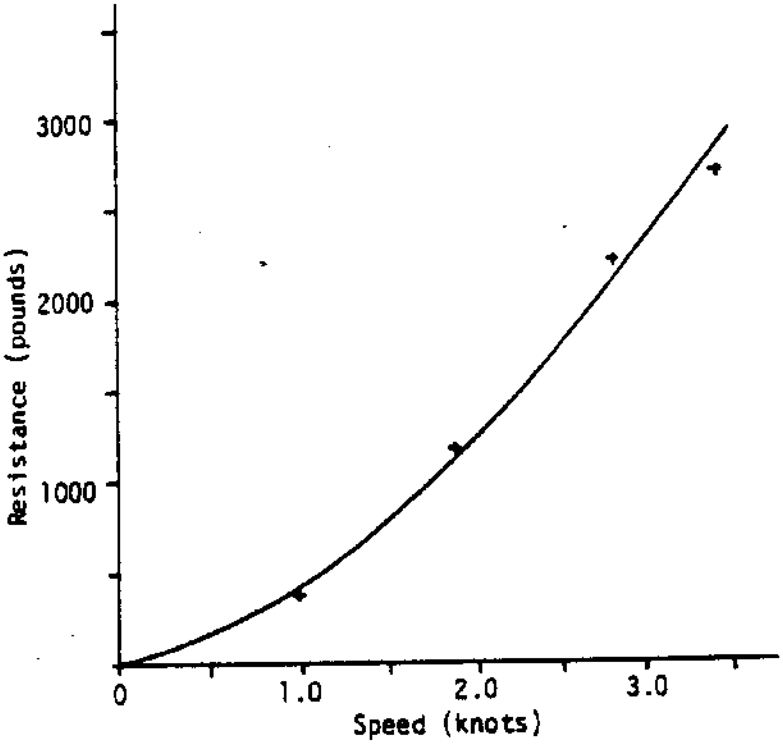

Figure 4. Resistance versus speed of off-bottom shrimp trawl.

As with nearly all data taken during sea trials, these measurements are subject to numerous sources of error. Sea conditions during the test runs caused wide variation in the instantaneous displays of tension and speed. Averages of visual observations were recorded. The proximity of the knotmeter strut to the vessel hull may have affected those readings. The propellor slip stream may have had an effect on the water flow at the trawl. Unlike resistance experiments where these parameters have been within the control of the researcher[2], the above results should be considered approximate.

ANALYSIS:

From the above cable tension data, a relationship for speed, towing depth, and required weight can be established. Again, cable characteristics have been ignored and a straight line configuration assumed.

The weight required to maintain the desired warp angle can be expressed

$$W = R \tan \theta$$

where W = weight, R = resistance, and θ is the vertical warp angle.

Similarly, the relationship of trawl depth to the warp length is

$$d = L \sin \theta$$

where d = depth and L = warp length.

Combining these two equations we can determine the fishing depth based on warp length, weight, and resistance. Guidelines for the operation of the trawl can then be developed for use in intercepting shrimp indicated on the sounder. Tables 2 and 3 have been so developed for weights of 200 and 600 pounds.

Warp length	Depth				
	1 knot	1.5 knots	2 knots	2.5 knots	3 knots
30 fm	14 fm	9 fm	5 fm	3.6 fm	2.6 fm
60	28	18	10	7	5.2
90	42	27	15	11	7.8
120	55	36	20	14	10.4

Table 2. Fishing depth with 200 pounds of weight.

Warp length	Depth				
	1 knot	1.5 knots	2 knots	2.5 knots	3 knots
30 fm	25 fm	20 fm	14 fm	10 fm	7.5 fm
60	50	41	28	20	15
90	76	61	42	31	22.5
120	101	82	56	41	30

Table 3. Fishing depth with 600 pounds of weight.

These tables reveal a trawl which, because of its fixed geometry and predictable resistance, can be directed to a wide range of depths using adjustments of warp length and vessel speed. Incremental changes in resistance due to codend loading could be compensated for based on experience.

COMMERCIAL POTENTIAL:

By eliminating the hydrodynamic resistance associated with conventional trawl doors and the seabed friction of a bottom trawl, this gear could be appropriate for the harvesting of off-bottom shrimp stocks, particulary in the Gulf of Maine. Further development of the concept is needed. The present design is suspected of being overbuilt. Fewer bridles or smaller crossection extrusions could be used.

Since contact with the bottom can be prevented by using the configuration diagrammed in Figure 5, much of the damage that occurs with conventional trawls is prevented. For the same reason, the twine diameter of the netting could be significantly finer, resulting in reduced resistance or larger frame dimensions.

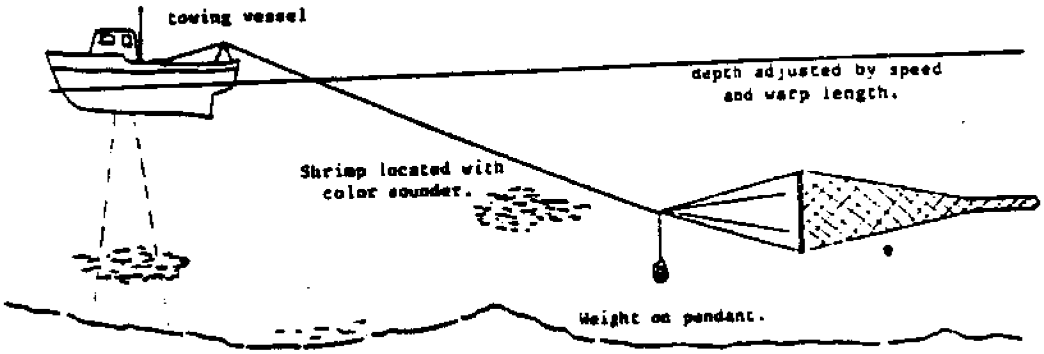

Figure 5. A recommended arrangment to prevent damage from bottom contact.

The material costs of the gear as constructed were as follows.

Aluminum extrustion	$360.00
2"x#15 tarred nylon netting	327.00
Rigging and hardware	244.00
Total	$931.00

This compares favorably with a complete bottom trawl of equivalent proportions.

FUTURE PLANS:

Commercial trials of this rig are being planned for the 1984-85 shrimp season for the determination of the gears effectiveness. The predictions of fishing depth verses speed and warp length will be verified and the results will be reported to the industry.

ACKNOWLEDGEMENTS:

The work described in this paper was supported in part by the MIT Sea Grant College Program, National Oceanic and Atmospheric Administration, U.S. Department of Commerce, through grant number NA29AA-D00101. Funds for the rig materials were provided by the New England Fisheries Development Foundation.

The author wishes to thank Capt. Lyle Chamberlain and his crew for their participation in this development effort. The assistance of Winninghoff Boats of Rowley, MA is gratefully acknowledged for its donation of the excellent welding services in fabricating the frame. The assistance of Paul Shuman in the design and assembly of the net is also appreciated.

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1. Watson, John W.. NMFS Pascagoula. Personal communication.

2. Goudey, C.A.. 1983. Full Scale Resistance Tests of Yankee Trawls of both Nylon and Polyethylene Construction. Center for Fisheries Engineering Research Report No. 1. MIT, Cambridge, Massachusetts.

APPENDIX 1: Bridle Optimization Program

```
READY.

1 REM:BRIDLE OPTIMIZATION PROGRAM, CLIFF GOUDEY, 4 OCTOBER 1983
2 REM:ORIGIN IS THE FRAME END OF THE FIXED LENGTH BRIDLE
3 REM:X AXIS IS TOP FRAME,Z IS PARALLEL TO SIDE FRAME,Y AXIS IS NORMAL TO FRAME
4 DIM ST(15)
5 SX=5.25:SY=0.00:SZ= 3.25:TX=-5.08:TY=1.72:TZ=0.430
10 WIDTH=25.88:HEIGHT=12.94:BRIDLE=29
20 L=BRIDLE:N=0:SUM=0:DF=0:PRINT"(HC)"
30 R=(BRIDLE^2-(HEIGHT/2)^2)^.5
40 PHI=ATN(R/(HEIGHT/2)):DSQ=0:DF=0
45 PRINT"ANGLE"," SIDE"," TOP"," TOTAL"
46 PRINT"EEEEE"," EEEE"," EEE"," EEEEE"
50 FOR THETA=0T+.01 STEP/12
60 PX=L*SIN(PHI)*COS(THETA)
70 PY=L*SIN(PHI)*SIN(THETA)
80 PZ=L*COS(PHI)
90 S=SQR((PX-SX)^2+(PY-SY)^2+(PZ-SZ)^2)
100 T=SQR((PX-TX)^2+(PY-TY)^2+(PZ-TZ)^2)
115 T=INT(T*1000)/1000:N=N+1
116 S=INT(S*1000)/1000:ST(N)=S+T
120 PRINTTAB(1)THETA//180),S,T,ST(N)
130 SUM=SUM+ST(N):NEXT:AVG=SUM/N
140 FOR M=1 TO N
150 DF=DF+ABS(ST(M)-AVG)
160 DSQ=DSQ+(ST(M)-AVG)^2:NEXT:PRINT
170 PRINT"AVG.="INT(AVG*1000)/1000
171 PRINT"DIFS.="INT(DF*1000)/1000
172 PRINT"DIFSQR.="INT(DSQ*1000)/1000
190 PRINT"DO YOU WANT A HARD COPY?(Y/N)"
200 GET A$:IF A$="" THEN 200
210 IF A$="Y" THEN 270
220 IF A$<>"N" THEN 190
230 GOTO500
270 L=BRIDLE:N=0:SUM=0:DF=0:PRINT"(HC)"
280 R=(BRIDLE^2-(HEIGHT/2)^2)^.5
290 PHI=ATN(R/(HEIGHT/2)):DSQ=0:DF=0
295 OPEN4,4,2:CMD4
296 PRINT#4,"FRAME WIDTH="WIDTH"  FRAME HEIGHT="HEIGHT
297 PRINT#4,"  MAIN BRIDLE LENGTH=";BRIDLE
298 PRINT#4,CHR$(27)"D"CHR$(3);CHR$(15);CHR$(30);CHR$(45);CHR$(0)
299 PRINT#4,"SX="SX"  SY="SY"  SZ="SZ"  TX="TX"  TY="TY"  TZ="TZ
300 PRINT#4
310 PRINT#4,CHR$(9)"ANGLE"CHR$(9)"  SIDE"CHR$(9)"  TOP"CHR$(9)" TOTAL"
320 FOR THETA=0T+.01 STEP/12
330 PX=L*SIN(PHI)*COS(THETA)
340 PY=L*SIN(PHI)*SIN(THETA)
350 PZ=L*COS(PHI)
360 S=SQR((PX-SX)^2+(PY-SY)^2+(PZ-SZ)^2)
370 T=SQR((PX-TX)^2+(PY-TY)^2+(PZ-TZ)^2)
380 T=INT(T*1000)/1000:N=N+1
390 S=INT(S*1000)/1000:ST(N)=S+T
400 PRINT#4,CHR$(9)THETA//180)CHR$(9)S;CHR$(9)T;CHR$(9)ST(N)
410 SUM=SUM+ST(N):NEXT:AVG=SUM/N
420 FOR M=1 TO N
430 DF=DF+ABS(ST(M)-AVG)
440 DSQ=DSQ+(ST(M)-AVG)^2:NEXT
445 PRINT#4
450 PRINT#4,CHR$(10)"AVG.="INT(AVG*1000)/1000
460 PRINT#4,"     DIFS.="INT(DF*1000)/1000
470 PRINT#4,"     DIFSQR.="INT(DSQ*1000)/1000
480 PRINT#4:PRINT#4
490 LIST5
500 CLOSE4:END
READY.
```

Sample Output:

```
SX= 5.25   SY= 0   SZ= 3.25   TX=-5.08   TY= 1.72   TZ= .43

  ANGLE        SIDE          TOP           TOTAL
   0           19.17         29.895        49.065
   15          19.394        29.391        48.785
   30          20.037        28.623        48.66
   45          21.018        27.624        48.642
   60          22.232        26.44         48.672
   75          23.567        25.132        48.699
   90          24.921        23.776        48.697
   105         26.204        22.464        48.668
   120         27.346        21.301        48.647
   135         28.29         20.399        48.689
   150         28.994        19.858        48.852
   165         29.427        19.745        49.172
   180         29.574        20.075        49.649

AVG.= 48.838     DIFS.= 2.77     DIFSQR.= 1.048


SX= 5.25   SY= 0   SZ= 3.25   TX=-5.1   TY= 1.75   TZ= .44

  ANGLE        SIDE          TOP           TOTAL
   0           19.17         29.914        49.084
   15          19.394        29.404        48.798
   30          20.037        28.629        48.666
   45          21.018        27.422        48.44
   60          22.232        26.429        48.661
   75          23.567        25.113        48.68
   90          24.921        23.749        48.67
   105         26.204        22.431        48.635
   120         27.346        21.265        48.611
   135         28.29         20.362        48.652
   150         28.994        19.824        48.818
   165         29.427        19.717        49.144
   180         29.574        20.056        49.63

AVG.= 48.822     DIFS.= 2.782     DIFSQR.= 1.06
```

EVALUATION OF KOYAMA'S EQUATION FOR ESTIMATING TRAWL NET RESISTANCE

by

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ABSTRACT

This paper presents a comparison of the trawl net resistance calculated by Koyama's Method /1/ and the full scale trawl net measurements made by Taber /2/,/3/. The comparison indicates the Koyama Method underestimates the measured trawl resistance. A modified Koyama Method is introduced to improve the resistance estimates.

1. Introduction

With the expansion of U.S. fishing fleets /4/ and the concern with energy consumption, it is necessary to have accurate estimates of the fishing vessel power requirements. This includes estimating the power used during trawling. Several studies have been made by Miyamoto /5/, Hamuro and Ishii /6/, Taber /7/, and Amos /8/,/9/ which serve as valuable references.

In the present study a comparison of the trawl net resistance calculated by Koyama's Method /1/ and the full scale trawl net drag measurements made by Taber /2/,/3/ are compared. The limited scope of the paper was necessary in order for it to be completed as a class project /10/ in NA 402 Small Commercial Vessel Design taught by the third author at the University of Michigan.

2. Nomenclature

a ... Maximum circumference of net, m
b ... Maximum length of net, m
C_d... Otter board drag coefficient = 0.3
C'_d .. Warp drag coefficient, Fig. 3
D ... Depth from towing point to trawl, ft
d ... Net twine diameter, m
d_1... Warp diameter, m
HP... Trawling horsepower
l ... Length of net mesh bar, m
l_1... Length of warp, m

L ... Length of tow warp, ft
R ... Total resistance of trawl
R_n... Resistance of trawl net, kgf
R_o... Resistance of an otter board, kgf
R_w... Resistance of warp, kgf
R'... Total resistance of trawl
S ... Otter board area, m^2
T ... Average warp tension, lbs
V ... Towing velocity, m/s
V_k... Towing velocity, knots
v ... Towing velocity, m/s
ρ ... Sea water density, kg s^2/m^4

3. Basics for Study

3.1 Trawl Resistance Components /1/

Following Koyama, the total trawl net resistance R_t is given by:

$$R = R_n + R_o + R_w \tag{1}$$

and the corresponding horsepower is:

$$HP = \frac{R \; V_k}{326} \tag{2}$$

3.2.1 Koyama Trawl Resistance Estimate /1/

Koyama developed an empirical equation for estimating the trawl net resistance /1/. Trawl net resistance measurements were made on ten different trawl nets. These nets were towed by seven different trawlers covering a range of 100 GT-300 HP to 3500 GT-4000 HP. The net measurements were made under typical operational conditions at speeds between 3.0 and 4.7 knots. The results are plotted in Fig. 1.

Koyama fitted the line denoted by "1" to the data in Fig. 1. This is given by:

$$R_n = a \; b \; (d/l) \; V^2 \tag{3}$$

where:

d/l ... average value for net panels 1 through 7 taken at:

Slide Panel for 4-6 seam net

Upper net for 2 seam net

This is illustrated in Fig. 2. Fig. 2 also defines the values of a and b used in the calculation.

3.2.2 Otter Board Resistance Estimate /1/

The trawl nets tested were fitted with upright curved otter boards. Scharfe /11/ has found that an angle of attack of 15 degrees is optimum from the standpoint of resistance. At 15° the corresponding drag coefficient $C_d = 0.3$. In Koyama's approach, the otter boards are assumed to be adjusted so the angle is 15°. The otter board resistance is then estimated by:

$$R_o = 1/2 \quad C_d \; S \; V^2 \tag{4}$$

3.2.3 Warp Resistance Estimate /1/

The submerged weight of the trawl warp (towing wire) is much smaller than the tension. Consequently, it is possible to consider the submerged warp as an equivalent straight line element. Under this assumption the warp resistance R_w is given by:

$$R_w = 1/2 \quad C_d' \; S \; V^2 \tag{5}$$

The warp drag coefficient C_d' varies with the angle of attack α as well as the Reynolds Number. The C_d' values of Diel /12/ are used in Koyama's method. Diel's C_d' values are plotted in Fig. 3.

3.3 Taber's Analysis of Trawl Resistance /3/

In a brochure prepared for the University of Rhode Island Marine Advisory Service /3/ Taber derived the following equation for trawl horsepower:

$$HP = \frac{2Tv}{33000} \sqrt{1 - (\cos(90 - \frac{\phi}{2}))^2 - (\frac{D}{L})^2} \tag{6}$$

For typical operations $\phi/2$ is small, and eq. (6) can be written as:

$$HP = \frac{2Tv}{33000} \sqrt{1 - \quad (\frac{D}{L})^2} \tag{7}$$

It is necessary to convert calculated HP to resistance R' in kgf. This is done by:

$$R' = 33000 \frac{HP}{v} \quad 2.205 \text{ kgf} \tag{8}$$

This is used in the following analysis of Taber's trawl resistance measurements /2/.

4. Comparison of Taber's Measurements and Koyama's Resistance Estimates

In a project sponsored by the University of Rhode Island's Marine Advisory Service, trawl nets were towed in the "torpedo range" approximately 8 miles east of Point Judith, Rhode Island. The runs were made in opposite directions to minimize the effect of wind and tide. Results for 17 conditions for the two bridle trawl and 13 conditions for the three bridle trawl were reported /2/. Taber gave the results in calculated HP which are shown in Fig. 4.

The test measurements were used to calculate the trawl net resistance R using the method of Koyama. The calculated HP was converted to the resistance R' using eq. 8. These values are compared in Table 1. The error in percent is given by:

$$\text{Error} = \frac{R - R'}{R'} \times 100\% \tag{9}$$

The comparisons are plotted in Figs 5 and 6. From Table 1 and Figs. 5 and 6 it is clear that the Koyama Method underestimates the measured trawl resistance. This lead to the development of the modified Koyama Method described in the next section.

5. Development of Modified Koyama Method

The error in Table 1 was analyzed using the MIDAS computer program /13/,/14/. A correlation between the error and the trawl towing speed was determined. This is shown by the line "2" in Fig. 7.

Using the results from Fig. 7, it was possible to introduce a correction factor k in eq. 3:

$$R_n = k \cdot a\ b\ (d/l)\ V^2 \tag{10}$$

where K = $1.0 - V_k < 3.0$ knots
$1.1 - 2.7 < V_k < 3.0$ knots
$1.2 - 2.5 < V_k < 2.7$ knots
$1.4 - 2.0 < V_k < 2.5$ knots

This allows the trawl net resistance to be estimated by the Koyama Method for $2.0 < V < 3.0$ knots.

Table 1 Comparison of Measured and Calculated Trawl Net Resistance

Set	Cal HP /2/	V_k /2/	R' lbs eq. 8	R lbs eq. 1	Error eq. 9
I-1	33.8	3.33	3310	3343	1.0%
I-2	21.2	2.57	2678	1990	-25.7%
I-3	12.1	2.18	1807	1438	-20.4%
I-4	20.7	2.74	2457	2275	- 7.4%
I-5	21.6	2.93	2400	2596	8.2%
I-6	20.2	2.66	2469	2146	-13.1%
I-7	22.1	2.79	2577	2357	- 8.5%
I-8	19.7	2.67	2399	2162	- 9.9%
I-9	19.8	2.70	2385	2210	- 7.3%
I-10	22.5	2.72	2690	2242	-16.6%
I-11	13.5	2.11	2082	1348	-35.2%
I-12	22.1	2.67	2691	2162	-19.7%
I-13	34.6	3.38	3329	3463	4.0%
I-14	21.3	2.73	2537	2258	-10.9%
I-15	21.3	2.69	2575	2194	-14.8%
I-16	21.6	2.66	2518	2146	-14.8%
I-17	26.1	3.00	3833	2720	- 4.0%
II-1	32.6	3.17	3341	3566	6.8%
II-2	21.6	2.60	2700	2398	-11.2%
II-3	18.5	2.49	2423	2185	- 9.8%
II-4	12.8	1.84	2771	1190	-57.0%
II-5	25.4	2.76	2994	2697	- 9.9%
II-6	24.4	2.83	2806	2834	1.0%
II-7	25.5	2.66	3117	2508	-19.5%
II-8	19.8	2.44	2645	2099	-20.6%
II-9	23.8	2.71	2856	2602	- 8.9%
II-10	32.2	3.03	3461	3242	- 6.3%
II-11	22.2	2.64	2734	2471	- 9.6%
II-12	29.8	2.84	3415	2853	-16.4%
II-13	27.7	2.84	3174	2853	-10.1%

Notes: Set I Two Bridle Wing Trawl
Set II Three Bridle Trawl

6. Conclusions

1. The Koyama Method underestimates the trawl net resistance measured in Taber's tests.

2. The error is larger as the velocity is reduced below 3 knots.

3. A correction factor k was introduced to improve the Koyama Method estimates. It is recommended that the original Koyama Method should not be used for speeds below 2.5 knots.

7. Acknowledgements

The authors appreciate the excellent typing of Mrs. Mercedes Latapie.

8. References

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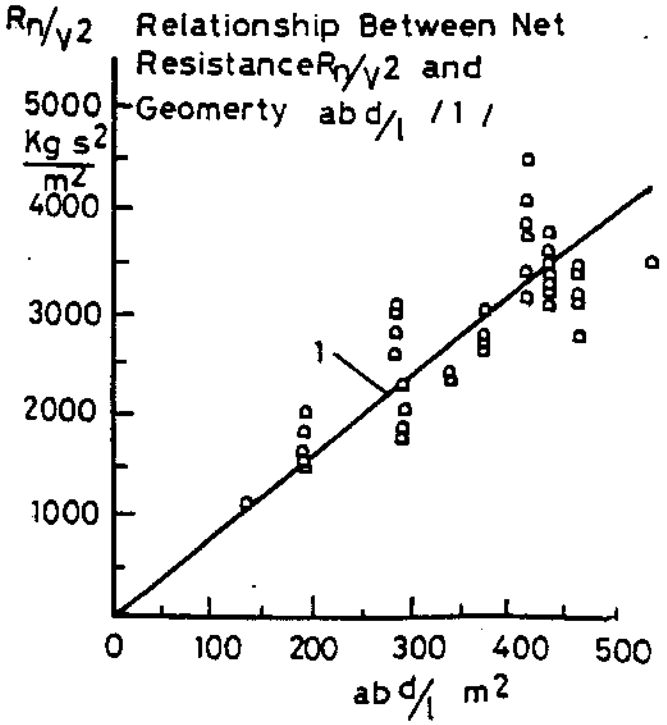


FIG.1

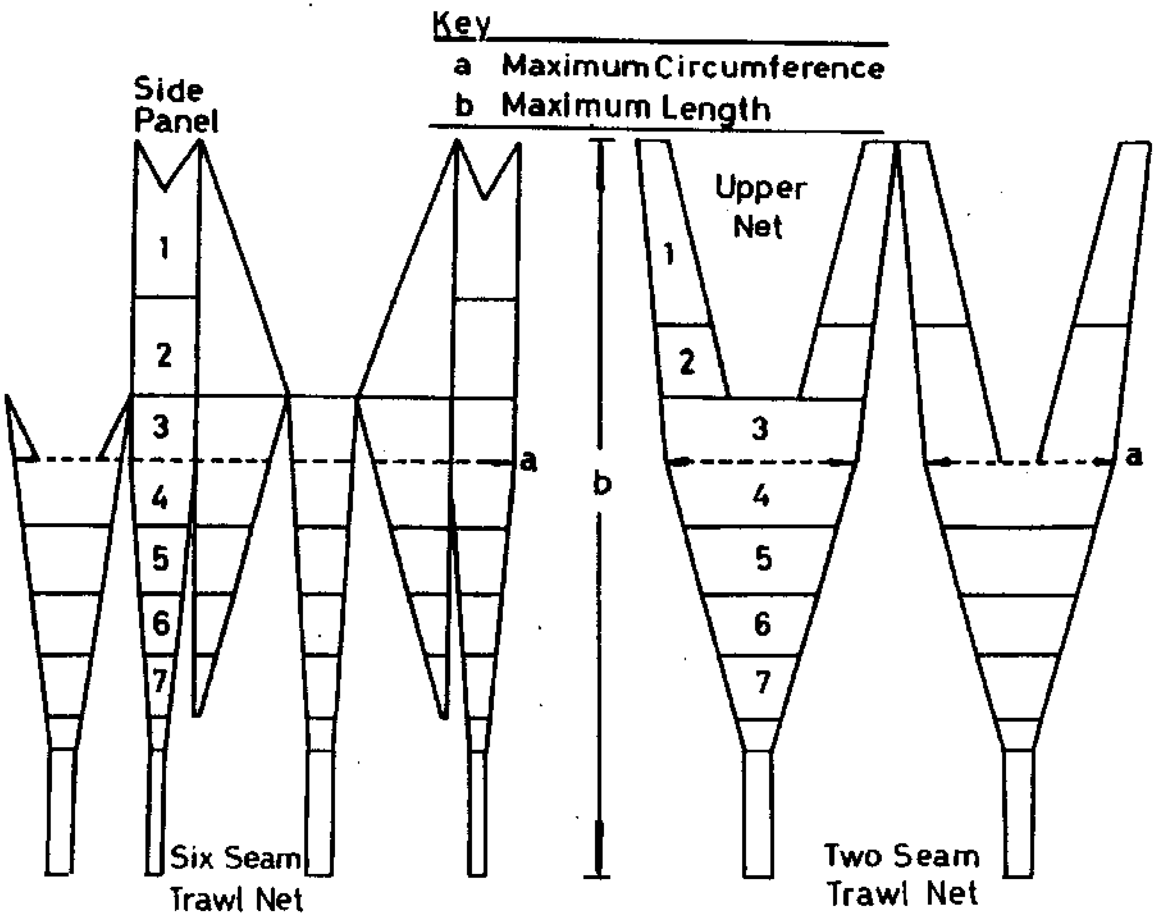


FIG. 2 Trawl Net Geometry For Resistance Calculation /1/

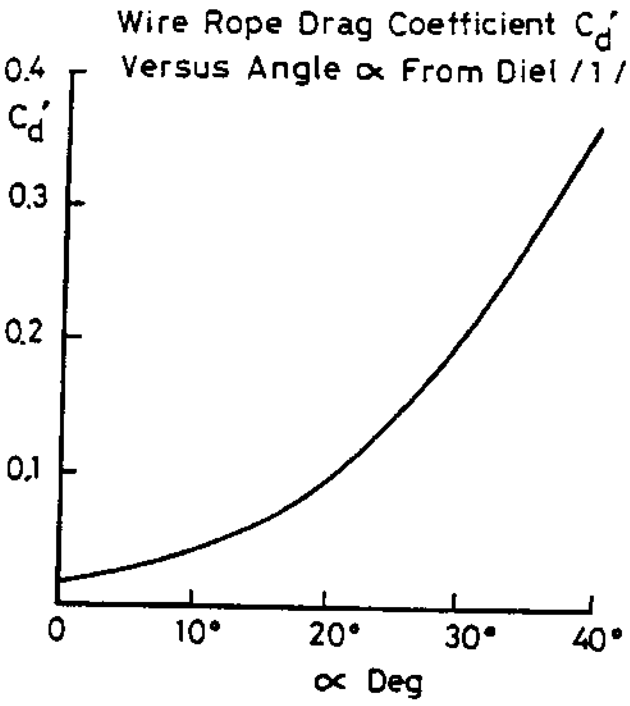

Wire Rope Drag Coefficient C_d'
Versus Angle α From Diel /1/
0.4
C_d'
0.3
0.2
0.1
0
10°
20°
30°
40°
α Deg

FIG. 3

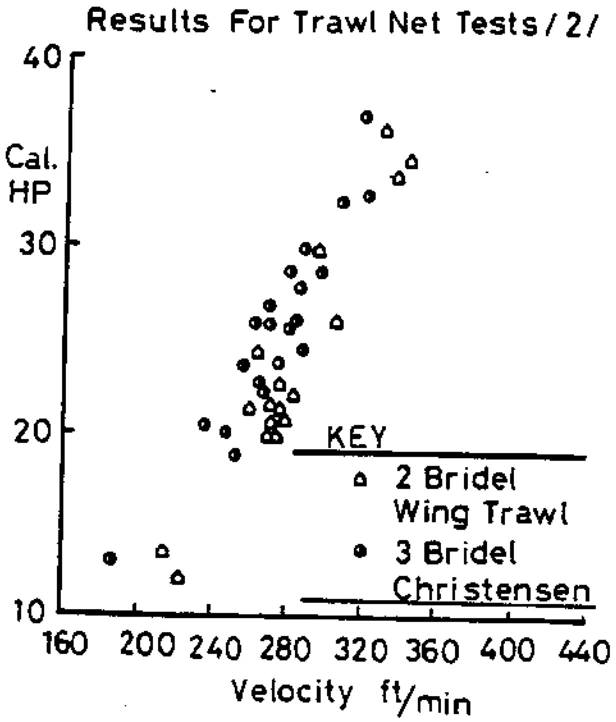

Results For Trawl Net Tests/2/
40
Cal.
HP
30
20
10
160
200
240
280
320
360
400
440
Velocity ft/min
KEY
△ 2 Bridel Wing Trawl
● 3 Bridel Christensen

FIG.4

FIG.5

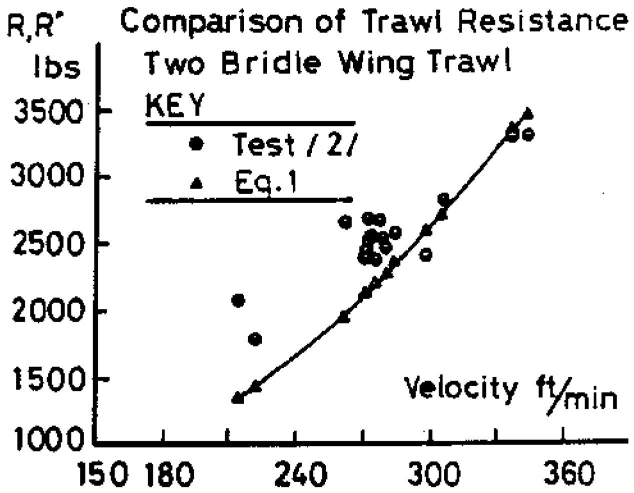


FIG.6

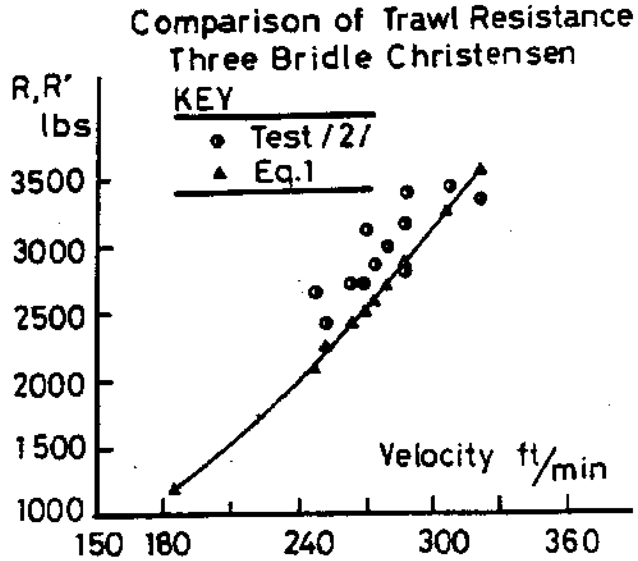
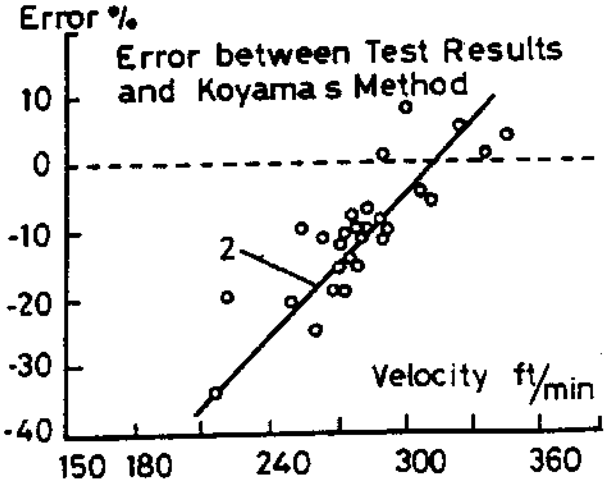


FIG.7



FISHERMEN AND OILMEN
WORKING TOGETHER IN THE NORTH SEA

ALAN WILSON

TOTAL OIL MARINE plc
LONDON WI ENGLAND

Introduction

Today, in many parts of the world the development of offshore mineral rights such as oil is increasing. The impact on the fishing industry of this type of development is constantly under discussion but is it really a serious problem? The matter has been discussed at 'world' level by governments trying to reach agreement under the heading 'Law of the Sea'. It has also been discussed within government between the various ministries or departments who have the responsibility for their country's activities in the fields of economics, energy, fishing, mining, transport and treasury. Finally of course there is the direct contact between the parties and between the people concerned.

The overall question is very vexed but basically it must be accepted that the industries involved are all necessary for people survival. Means must therefore be found to accommodate the needs of each industry. Thus the fisherman can have his gasoline or diesel oil for his boat, car and truck and the oilman can have his protein needs at meal times supplied in the form of crab, oyster, shrimp, or white fish.

This paper will show that in the sea area between Europe and the United Kingdom know as the 'North Sea' co-existence has been achieved. This has been brought about by regular on-going discussions and, in the event that a problem area has been determined, it has been properly addressed and resolved.

By examining the construction of an offshore pipeline some of the problem areas will be revealed. Their resolution can then be discussed.

The Pipeline Route

In the North Sea a typical pipeline to carry oil or gas is likely to be from twelve inches (12") to thirty six inches (36") in diameter and from twenty (20) kilometers to four hundred (400) kilometers long. It may go between two platforms or between platform and shore. At present almost all North Sea field developments are by use of steel or concrete platforms. The drilling of the development well is carried out from deck of the platform and the product produced from the well may receive treatment on the platform before being transported to shore.

Ideally, the pipeline should follow a straight line but having drawn that theoretical line on a map, the route must be examined certainly kilometer by kilometer, and in some areas perhaps even more precisely than that. First, consideration must be given to whether it crosses allocated areas or blocks belonging to other oil companies and in which exploration or development is intended. Next a sea bed survey must be carried out to determine the precise nature of the soil formation down to at least one meter below the bed surface. This may include actual soil sampling to confirm existing data. This survey must also quantify changes in level since abrupt changes, especially if associated with rock outcrops, may be unacceptable for eventual pipelaying.

When the basic route has been determined, it is then discussed with other operators and with the fishermen through their industry organization. If that basic route is acceptable the pipeline can be planned and complex questions such as the effect of sea bed currents along the proposed route studied.

The existence or not of currents will effect whether the pipeline can be safely left on the sea bed or whether it must be trenched. At one time North Sea pipelines were almost always trenched but this produced problems for both fishermen and oilmen. For the former the problem was that trenching seriously disturbs the sea bed along the pipeline route. If the sea bed is sand then the natural movements on the sea floor will soon smooth it over. But, if it is hard sand, boulder clay or rocky outcropping then it may be many years, if ever, before the sea bed returns to its original state. For the latter the problem is that some trenching techniques risk to damage the pipeline or its concrete coating. But regardless of technique it is a high cost item and could impact on the overall economics of a field development. Some authorities quote figures of over $1,000 per meter.

To investigate the question of leaving pipelines on the sea bed, studies were carried out by some of the oil companies operating in the North Sea in conjunction with government departments and universities. These tests showed that, if due precautions are taken, then in more than twenty (20) meters of water, it is better to leave pipelines over sixteen (16) inches in diameter resting on the sea bed and untrenched. (Ref. 1 & 2). Below that figure discussions are still continuing and a new research project to cover pipelines with diameters between six (6) and sixteen (16) inches is presently being developed.

Trenching pipelines in shallow water or shore approach areas is normally needed to avoid the effect of inshore currents and to provide additional protection against, for example, inadvertent anchoring by small vessels.

Pipeline Design

The diameter of the line will be determined by the quantity and nature of the product to be transported. These parameters will in turn also determine the steel thickness and steel grade to be used. For pipelines of thirty (30) inches diameter the wall thickness may well be between three quarters of an inch (0.75") and one inch (1.0"). The grades are likely to be in accordance with the specification API 5 L -grades X.52 to X.65.

The pipe when laid must stay in place on the sea bed and since the steel weight on its own will normally be inadequate it must be given extra weight to ensure that the necessary negative buoyancy or submerged weight is attained. This is normally achieved by the addition of concrete reinforced with steel cages. The concrete is applied over a coating providing corrosion protection and which may well be of coal tar or bitumen reinforced with fibre glass. Epoxy and polyethylene have also been used for this purpose but the former has technical drawbacks whilst the latter may be rather costly if a good thickness of material is used. Corrosion protection is further guaranteed by the addition of self sacrificing anodes made of zinc or complex aluminium alloys. Simple aluminium alloys are not normally suitable for sub-sea pipelines.

Good corrosion protection is essential for the pipeline but it is also essential for the fisherman as it reduces to a minimum the possibility of corrosion resulting in pipeline failure.

The concrete weight coating is also important to the fisherman since it ensures that the pipe stays in the place where it has been laid. It also protects the line against impact from types of fishing gear which are pulled along the sea bed.

Pipe Protection with Concrete Coating

Concrete is added to the steel pipe either by forming or impingement techniques. Its basic purpose is to keep the pipe stationary on the sea bed but the mechanical protection function is equally important for safe operation.

Trawling in the North Sea is done by vessels of from 500 to 2500 H.P. (Table I) and the net is held open by trawl doors weighing from one (1T) to (2T) tonnes. (Figure 1). These doors are metal and reinforced on their edges and hence they have a damaging effect on anything they hit. (Figure 2). Should this be a pipeline then the concrete must protect the line from denting or, in the ultimate, rupture.

Major development programmes have been completed between pipe coating contractors and oil companies to ensure that the concrete as applied is adequate for its protection role. At one time it was applied using wire mesh or 'chicken' wire. This mesh was only really intended to hold the concrete in place during the coating process until the concrete had 'cured'. It did not have a true reinforcing effect. Today such wire has been replaced by either very heavy woven wire or more usually by cages made of 'rebar steel five (5) to eight (8) millimeters in diameter.

The important aspect regarding this use of a greater quantity of steel per meter length of pipe is that is provides a true reinforcement of the concrete, both during the laying and once the pipeline is on the sea bed. In fact, the percentage of steel within the concrete is about the same as that of reinforced concrete for buildings. It therefore provides good resistance to impact from heavy objects such as trawl boards or small boat anchors.

Some oil companies wished to check both the suitability of pipes coated with reinforced concrete for laying and just how resistant it was to impact. Three tests were therefore devised. They were:

- a shear test
- a bending test and
- an impact test.

The shear test is carried out by fixing say a one meter length cut from a coated pipe on a holding block. Hydraulic rams are then used to apply a horizontal force to the steel pipe until such time as slippage occurs between the concrete and the corrosion coating on the steel of the pipe. This force is then related to the shear forces that the pipe will experience during laying and a check made that the pipe will not slip through the concrete.

For the bending test some eight (8) joints of pipe, each of the nominal joint length of twelve (12) meters, are welded together on a flat test site. Using a suitable crane, one end of the pipe is lifted until some six (6) joints are clear of the ground. The concrete is then inspected to see that no major spalling has occurred and that no significant movement of the concrete coating has taken place. This test is repeated, first with the pipe rotated through one hundred and eighty (180) degrees about its longitudinal axis. Next, these tests are repeated lifting from the opposite end of the pipe.

This complete test simulates the pipelaying 'S' bend technique. Thus, it further confirms the suitability of the chosen concrete coating system for actual offhsore laying.

For the impact test a hammer weighing from on (1) to two (2) tonnes is fixed in a jig by wire supports (Figure 3). The exact weight depends upon the trawl board to be simulated. The hammer has an edge on the head of say thirty (30) by two(2) centimeters. A test pipe is placed in front of the hammer head so that the coating may be struck a blow at ninety (90°) or sixty (60°) degrees to the horizontal axis. By allowing the hammer to swing through a distance of from on (1) to two (2) meters, an equivalent speed to from three (3) to five (5) knots can be achieved.

Thus the pipe is struck with a force equal to that of a trawl board being pulled along the sea bed and hitting a pipeline.

Various programs may be devised for this type of test. For example, a coated pipe may be required to withstand at least twenty (20) blows at any one spot without the steel surface of the pipe becoming visible. Alternatively, single blows may be struck at random, at twenty (20) centimeter intervals along the pipe without causing the concrete over the reinforcement to fall away.

Ideally such a test, unlike the two others, should be performed on selected pipes throughout the coating production to check that the quality of the reinforced concrete is being maintained.

Pipeline Field Joints

The pipe when coated is ready for laying and this will be done from a lay barge. On board the barge the pipe is welded together by manual or semi-automatic welding techniques.

After welding and after the weld has been proved by non destructive testing including radiography, a field joint has to be applied. This fills up the space between the two layer of concrete on either side of the weld.

The final exterior shape of the field joint must be such as to provide a contiguous surface and also, as with the concrete, provide a sound mechanical protection barrier. It often consists of an adhesive tape overlaid with mastic material formed from bitumen and aggregate. The tape is applied directly to the steel to complete the anti-corrosion wrapping applied in the coating yard. The mastic replaces the concrete and provides both the weight and the protection against mechanical impact.

The joint itself if properly applied represents no problem to fishermen but the method of application must be controlled. In the past the field joint was formed by using a thin sheet steel cover or former held in place by steel bands. This form was left in place after pipe laying and in some cases the steel holding bands corroded away allowing the thin sheet steel cover to become loose. Cases have been reported of fishing nets being cut when the net has been passed over such loose steel forms.

Following discussion between all the parties involved in assessing such a problem, care is now taken to ensure that the sheet steel form is kept free of sharp edges and the banding is done with either very strong high grade plastic or stainless steel. In either case the bands do not corrode and hence the form stays in place.

This solution may well only be in the short term since three other solutions are now in the last stages of practical proving.

The first is the use of a form made of a magnesium alloy. (Figure 4). The form is made of extruded sections and hence is smooth. Being of magnesium it corrodes away in sea water and thus leaves the field joint in a smooth and acceptable condition. By adjustment of the chemical composition of the alloy the rate of corrosion can be adjusted. Hence adequate life during storage in a marine environment is achieved whilst maintaining the eventual desirability that it corrodes away on the sea bed. (Ref. 3).

The second is a quick curing concrete which will harden within six (6) minutes to a value acceptable for passing over the barge stinger or laying ramp. This is moulded using a machine on the barge and the moulding surface or form does not pass into the sea.

The third is similar to the second but instead of concrete consists of a quick curing plastic containing aggregate or iron ore to provide density. The use of a plastic material for the field joint has been considered for many years but the problems of use have been to achieve adequate weight and good mechanical impact resistance properties. (Ref. 4). Materials possessing these properties are now available and curing in four (4) minutes is possible. Again the form is removable and stays on board the barge. The chemicals involved in producing such plastic must be carefully evaluated for, whilst the final joint is chemically inert and safe, this may not be the case for the curing solution, particularly if subject to heat or fire.

The pipelaying operation is one of passing the pipe over the stern of the lay barge. This is achieved by moving the barge forward using winches pulling on anchored cables. (Figure 5). Naturally the anchors have to be moved as the pipelaying progresses. In certain soils this movement of anchors can leave a disturbed sea bed similar to but on a much smaller scale than that caused by trenching.

Here again a sandy sea bed particularly in areas of known sea floor currents will be self correcting. In areas of boulder clay this may not be the case since if the boulders are pulled to the surface and if the clay is hard they may not be reabsorbed within the sea bed. Anchor mounds can then represent a problem for fishing activities involving trawling or seine netting.

Recently, ploughs have been developed capable of smoothing out such anchor mounds and if pipes do have to be trenched then such ploughs can be used both for the trenching and the grading of the trench sides. Very large ploughs may even be capable of breaking up all except the largest of boulders but can in any case be used to grade the soil around very large boulders to prevent them remaining as a problem to fishermen.

Pipelines in Operation

When a pipeline has been laid and put into operation it must be assumed that fishing activities will take place along and across the pipeline.

As discussed earlier, the direct impact of such items as trawl boards can be guarded against in the design phase.(Ref. 1 & 2). Two other aspects are however important.

The first is the development of free spans during the life of the pipeline and the second is the use of fishing techniques where nets may be dragged across the line.

Freespans

When a pipe is laid on the sea bed it may be supported between two rock outcrops. Alternatively, during operation sea bed currents may wash friable material away from under sections of the pipe leaving a length of say fifty (50) to one hundred and fifty (150) meters unsupported.

In either case the span so formed will need technical appraisal.

The basic questions to be addressed are, can it be:

- left as it is?
- restabilised?

A combination of theoretical and practical studies has shown that certain spans can be left quite safely if their length is correctly related to free linear weight and correctly related to free linear weight and current velocity plus current and tide direction. (Ref. 5).

If the evaluation shows that a span cannot be left safely then means must be used to re-stabilish it. This may mean removing hard zones at either end of the span to let the line again rest on the sea bed, or filling up the space below the span. The latter can be done in the simplest cases by diver positioning of sand bags or grout bags. In more complex areas, use of special vessels to dump heavy aggregate or small sized rocks (below six (6) inches) may be needed to ensure that the problem does not arise a second time. (Figure 6.)

It is worth nothing that on thirty (30) inch diameter pipelines spans of over fifty (50) meters in length have been shown to be perfectly safe under their related sea bed conditions. (Ref. 6).

Seine Net Fishing

This is one fishing technique which may result in net being slowly pulled across a pipeline. To assure both oil companies and fishermen that this was not a problem practical trials were carried out in the North Sea. (Ref. 7).

The gear used for these trials consisted of two towing warps two thousand six hundred (2,600) meters in length connected to a sweep holding open the mouth of the trawl which was approximately thirty five (35) meters across.

The gear was run out in a conventional fashion and then towed. During the tow the net was slowly hauled into the vessel and the action of hauling caused the mouth of the net to be closed trapping the fish inside the nets. Deployment handhaul took about two (2) hours and was carried out so that the net passed over two typical pipelines positioned some thirty (30) meters apart.

A total of seven tows were completed on four different pipeline locations. The vessels used was typical fishing vessel having a gross tonnage of fourty eight (48) tonnes and an overall length of twenty (20) meters.

The trials showed that in the areas tested the pipelines did not cause any damage to the gear. Whilst it cannot be said to be conclusive for all conditions it represents nevertheless an example of co-operation to improve the understanding between fisherman and oilmen.

Other Aspects

Dumping at Sea

Offshore construction is no better but no worse than onshore construction. It produces waste, and what easier place to dump it than in the the sea. This must be guarded against as a matter of ecological good behavior, but additionally some materials if dumped at sea can be hazard for fishing equipment.

Examples of this are, the steel shavings produced when preparing pipe ends for welding, old wires and chains, steel offcuts and any metal detritus which can snag, cut or tear nets. Careful housekeeping, if necessary controlled by legislation as it is in the North Sea can prevent this happening. (Ref. 8). Not that all the blame can be allocated to the oil industry. A study in Norwegian water supported by the Government of Norway showed that when a know sea bed area was trawled for waste some 60% of the material recovered was shown to emanate from the fishing industry, mainly wires and ropes.

Industry Liaison

Within North Sea areas consultative groups have been set up to provide a forum for discussions. An example of this is the Fishing and Offshore Oil Consultative Group (FOOCG) in the United Kingdon. Membership is a mixture of representatives from government departments, oil companies through their association known as the UK Offshore Operators' Association (UKOOA) and fishing industry groups. This consultative group meets on average at least twice per year and has a sub-group which studies in particular pipeline related problems.

Another direct relationship between fishermen and oil men is one which arises when fishermen experience some problem due to fishing gear becoming damaged and where it is suspected that the damage has been caused by some type of oil installation or debris associated with the oil industry. In such a case the location where the damage occurred is recorded using Decca coordinates. On returning to port, the problem is reported to the Government appointed official fisheries officer who should have full details of the position of pipelines and platforms. That officer can therefore advise the fishermen as to which offshore operator owns the installation with which the damage is said to be associated. It is to that operator that the fisherman then addresses his claim for compensation.

Where the incident is close to an oil industry pipeline, well head or platform, it is obviously quite simple for the "offending" oil company to be identified in this way, but on occasions the incident turns out to be nowhere near any identifiable company's installation. In such a case the claim may be directed to a fund which has been set up by the United Kingdon Offshore Operators' Association (UKOOA) in order to pay compensation for an eventuality when an oil industry origin seems to be indicated but cannot be assigned to a particular operator. Naturally, when an oil company has been clearly identified, the claim is addressed to that company. Most operators have found it useful to have an experienced fishing expert available as a consultant to comment upon the merits of the claim and the validity of the amount of money being claimed to cover the repair of whatever fishing gear has been damaged. A recommendation may them be made for either the claim to be settled or rejected. In the case of rejection, the fishermen may address his claim to the UKOOA fund for payment. This fund is administered by a specialist committee consisting solely of representatives from the fishing industry. In the event that that committee recommends payment from the fund then naturally this is made and in effect the claim is distributed between all of the operators who are members of the offshore association.

Despite requests the UKOOA have decided that there should not be an oil man on that committee and payments from the fund are never made in other than well documented and well justified cases. The overall percentage payments are, in fact, lower than those made directly by the operators.

It can be seen that either by direct payment from an oil company or by indirect payment from the fund, fishermen are recompensed for any damage that may be considered to be oil industry related and hence unnecessary tensions between the two industries are reduced to a minimum.

In all North Sea zones a pipeline cannot be laid without prior permission having be obtained from the relevant government department. (Ref. 9). Normally this is the one responsible for Oil and Energy. Whilst the application for permission to lay naturally covers many technical aspects, it also covers sea bed conditions but also is required to include documents showing that the prepared route has been discussed with and accepted by the relevant fishing industry organisations.

Fishermen and Oilmen both have their jobs to do but it is desirable that they work together when common problems arise. Thus an amicable relationship can be generated and they can work together in harmony.

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Ref. 7 Report of Seine Net Fishing Trial on Frigg Pipelines October 1983 - A Total Oil Marine internal report (available on written request).

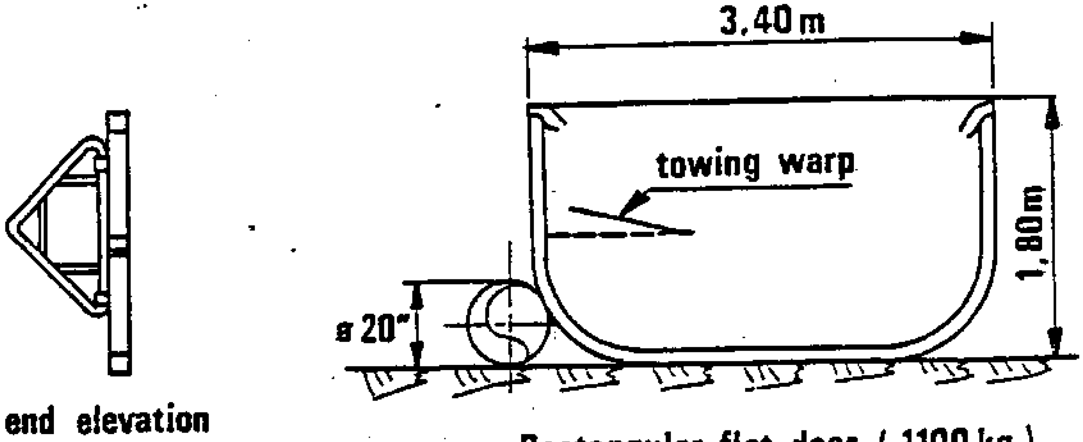
Ref. 8 The U.K. Dumping at Sea Act 1974. Her Majesty's Stationary Office (HMSO), London.

Ref. 9 The U.K. Petroleum and Submarines Pipelines Act 1975 HMSO London.

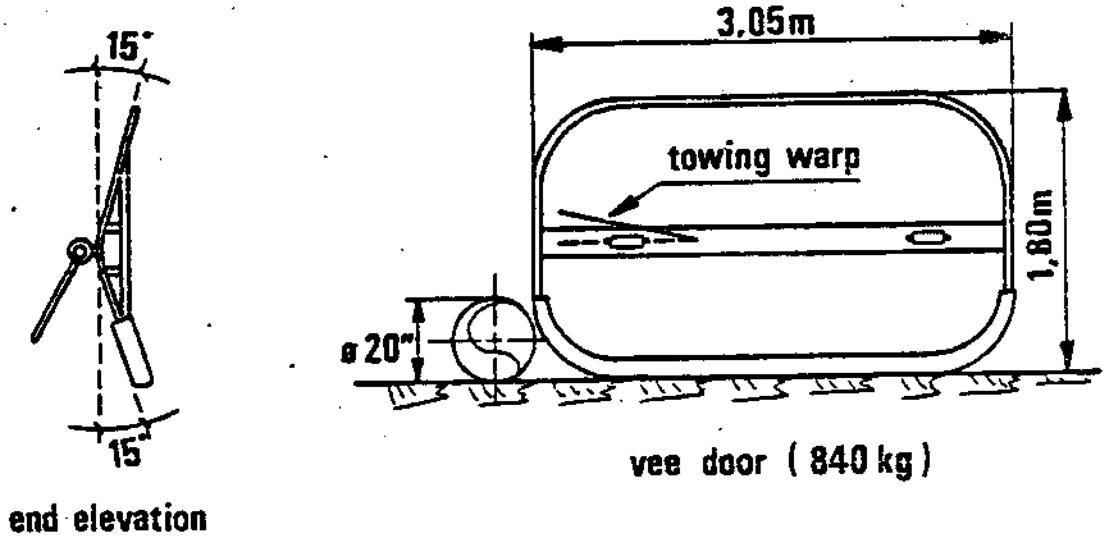
TRAWL DOOR TYPE	WEIGHT IN AIR Kg	OVERALL LENGTH Meters	OVERALL HEIGHT Meters
RECTANGULAR	1100	3.20	1.52
BIG V-SHAPED	1550	3.40	1.84
SMALL V-SHAPED	500	2.35	1.24
OVAL	1200	3.12	1.84

TABLE I TRAWL DOOR DATA

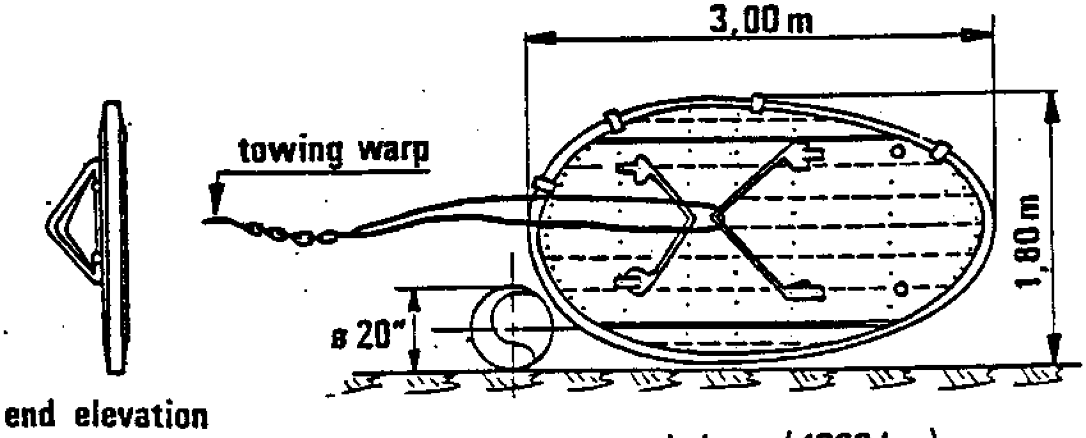
Fig. 1 : Typical Large Trawl Boards



Rectangular flat door (1100 kg)

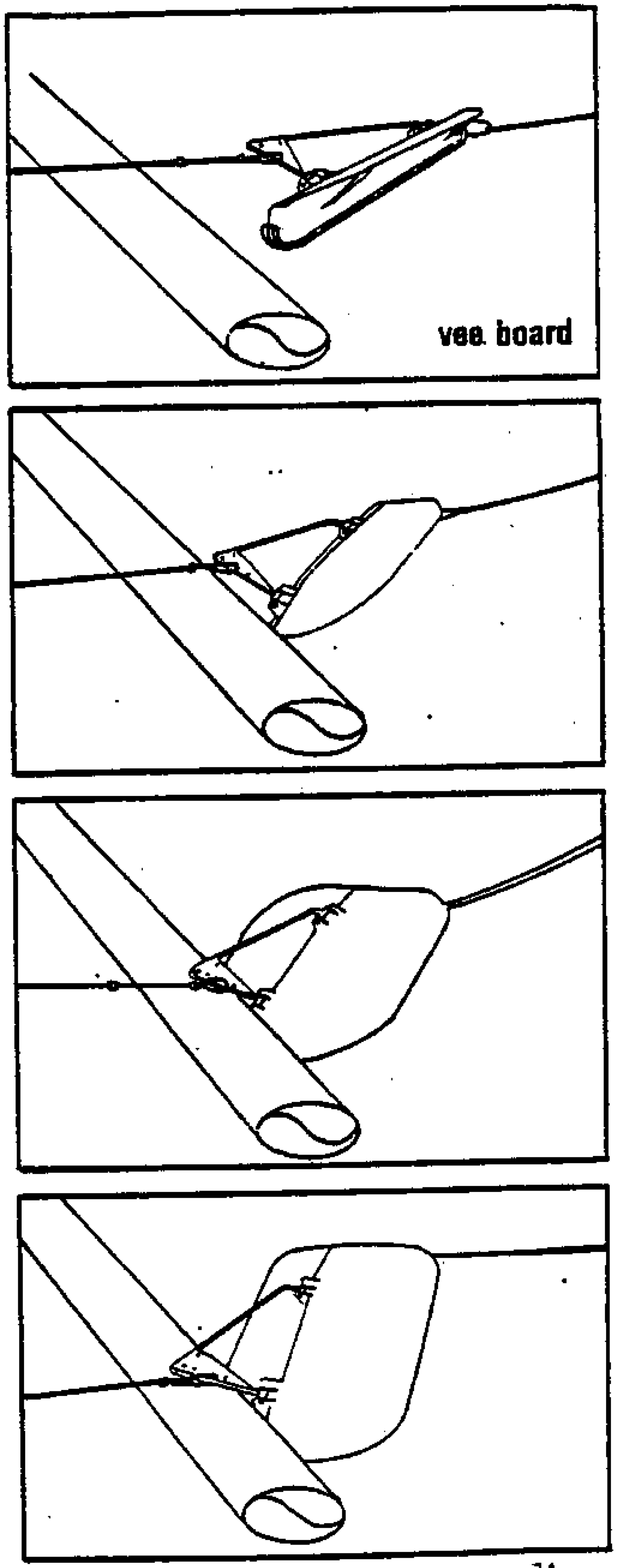


vee door (840 kg)



oval door (1200 kg)

Fig. 2 : Trawl board crossing a pipeline



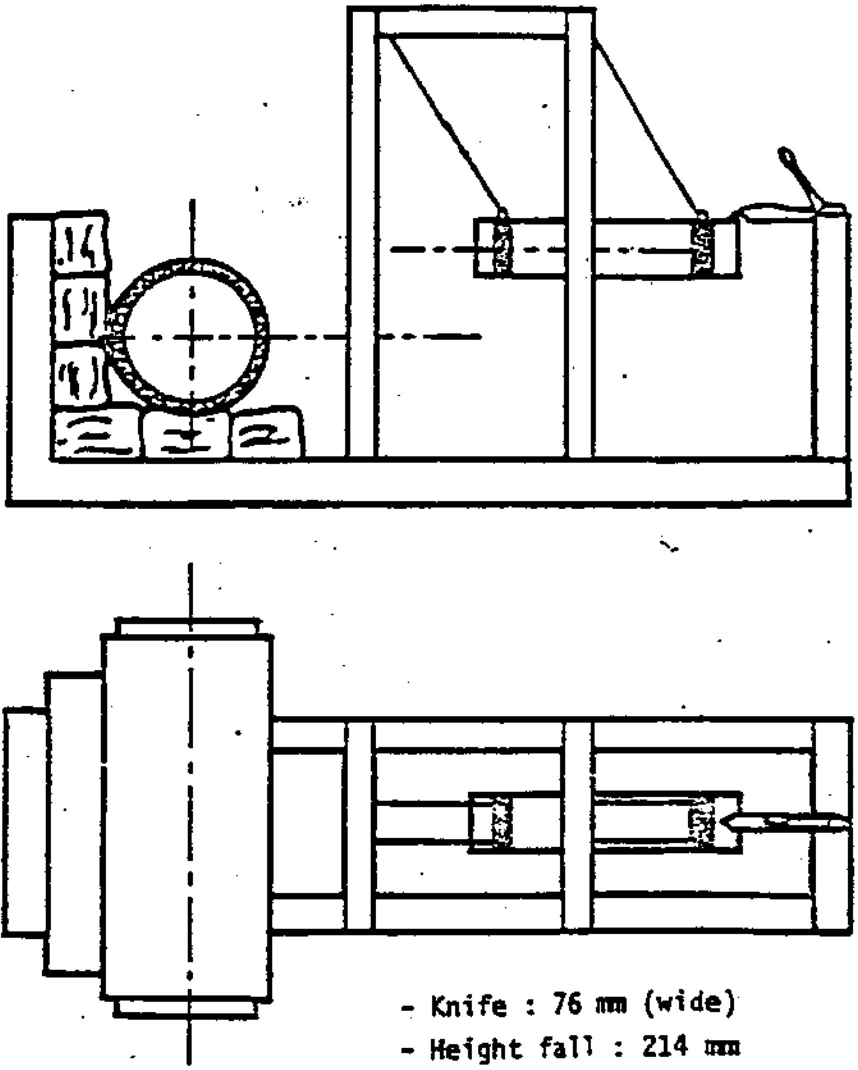
Tow warp contact with pipe

Board topples

Board parallel to bed

Board rolls over pipeline

CONCRETE TESTING DEVICE



- Knife : 76 mm (wide)
- Height fall : 214 mm
- Mass : 1 t

Fig. 3

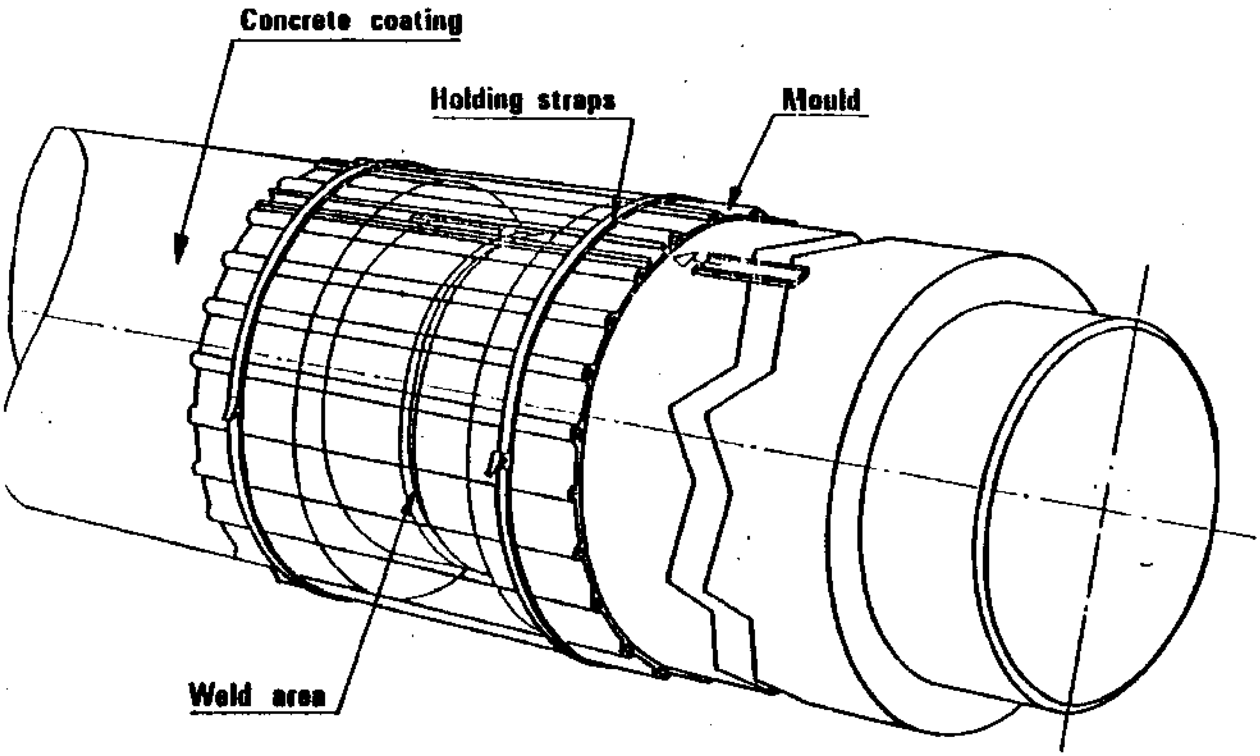


Fig. 4 : Magnesium Mould for Pipeline Field Joints

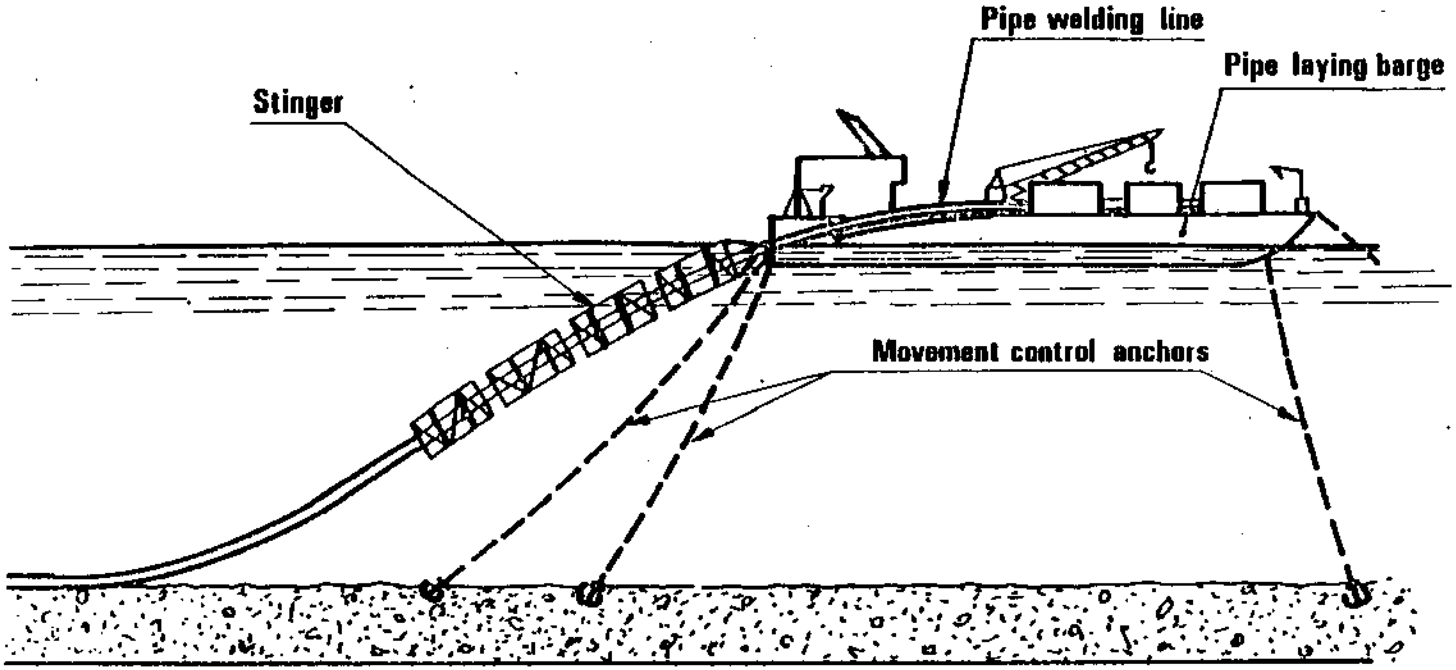


Fig. 5 : Conventional 'S' curve pipe laying

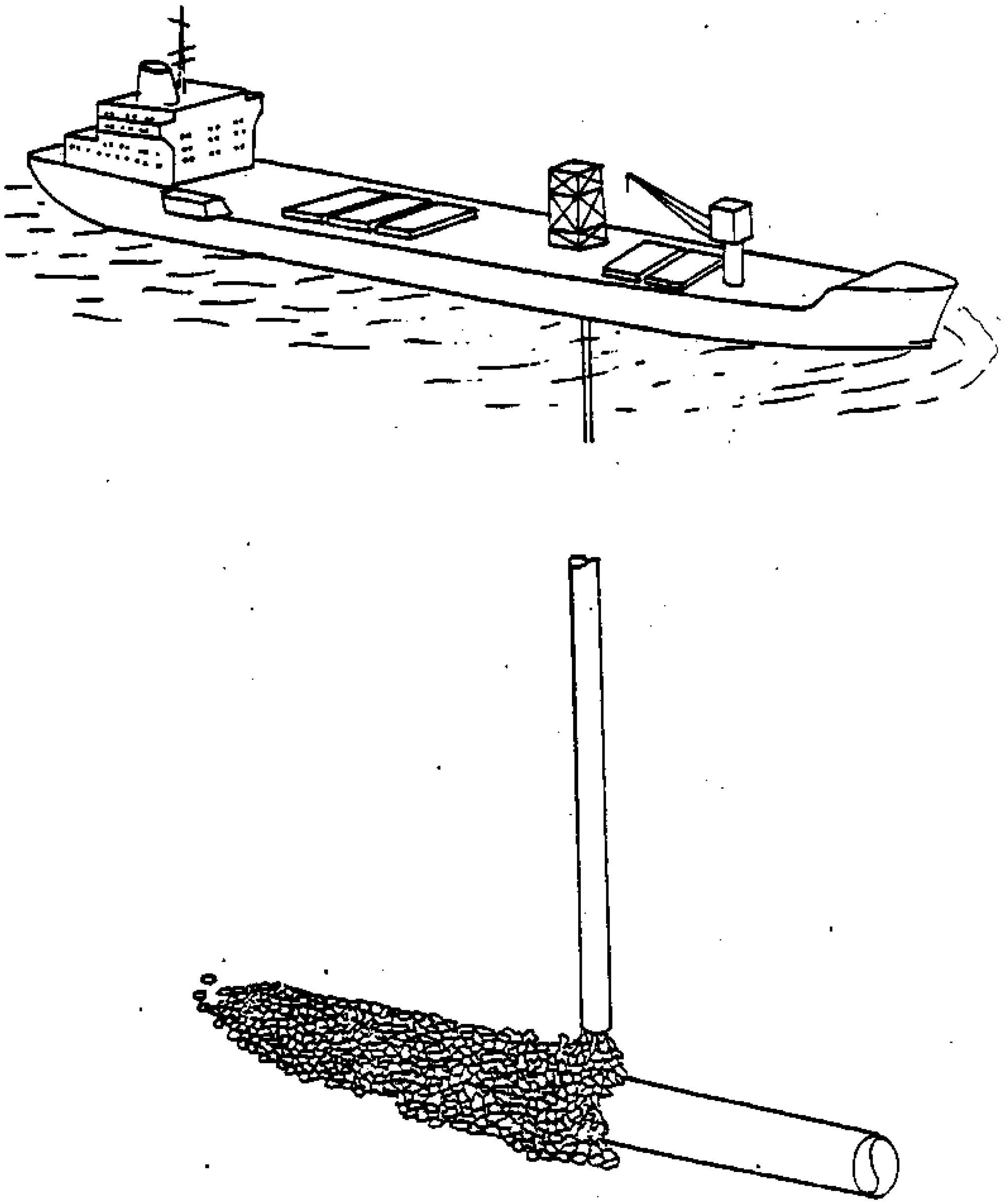


Fig. 6 : Method of Dumping Aggregate on Submarine Pipeline

COMPUTER APPLICATIONS IN
MIDWATER AND BOTTOM TRAWLING

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Abstract

This paper examines the development of automatic controls for trawl winches spurred by recent microprocessor-based technology. Shooting, towing, and hauling of the trawl net are functions that now can be accomplished with minimal user input from a central command station. Maintaining the trawl net in proper configuration and protecting it from inadvertently encountered obstructions using computer controlled constant tension and decision logic is reviewed. Future interface of the automated trawl winch controls with the vertical depth sounding electronics that will maximize fish harvesting capabilities is proposed. Additional link-up of fish-finding sonar and communications electronics that will combine the locating and harvesting of fish into one computer-aided process with shore-based monitoring is outlined.

1.0 INTRODUCTION

Automated trawl winch controls had their start in the early 1960's when large European factory trawlers installed automatic brake release and tension controls to prevent breaking wire on their electrically driven trawl winches. For hydraulic trawl winches, the beginning step was taken in Norway by Hydraulik Brattvaag in the late 1960's. They enabled trawl winches to haul-in and pay-out automatically by balancing and adjusting hydraulic pressure between the split trawl winches. In the 1970's, Norwegian manufacturers Rapp-Hydema, Arctic Heating, Norwinch, Karmoy, and F. K. Smith Electronics all contributed to the development of similar automated controls and the application of microprocessors. The following is a description of IntelliTrawltm, the first microprocessor-based automated controls for hydraulic trawl winches developed domestically in the U.S.A. by Marco Seattle.

2.0 AUTOMATED SHOOTING, TOWING, AND HAULING

The basis for automated winch control is generally characterized by position or tension control. Automated trawling employs both characteristics. Position control can be looked at as a set-up and limitation system; tension control as a balancing or optimizing system. These characteristics are variable from operation to operation, vessel to vessel, and set to set. Operational limits must be established also to protect the machinery and system from self-destruction. These are characteristics of a specific winch system and can be built into a control or programmed into a system capable of operating a variety of winch systems.

2.1 Data Entry

At installation and start-up of an automated system, the control computer must be programmed with the specific parameters of the winch system it will control. This part of data entry is a one-time operation unless, of course, one of the parameters changes. Parameters to be entered are:

1. Drum size: core diameter, distance between flanges, and flange diameter, in combination with wire diameter and length, will allow the system to define the operation point on the drum and calculate the length of wire paid out.

2. Wire diameter and total length: used with drum size in establishing wire position and length paid out.

3. Maximum allowable drum RPM: prevents internal damage to machine.

4. System pressure: prevents system damage or overload.

Each set, or tow, presents a potentially different set of operating requirements. Some of the requirements will change little or not at all from tow to tow, but should be addressed at the start of each tow (see Figure 1).

1. Set length: the amount of cable to pay out in order to place the net in the proper trawling position.
2. Window of operation: maximum permissible plus-or-minus deviation from the set length prior to sounding of an alarm.
3. Guard length: the length from the stern of the vessel within which automatic functions are locked out; manual controls only can be used inside this position to ensure proper setting of the trawl gear and safe handling of the trawl doors.

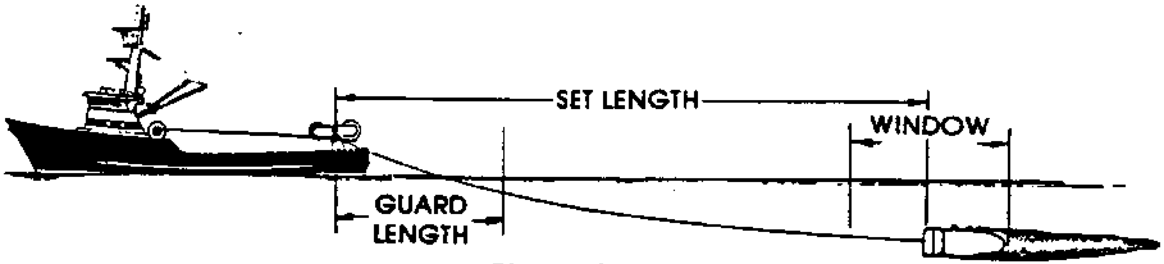


Figure 1

2.2 Video Display

During fishing operations, information such as set length, port and starboard warp length, differential length, guard length, window, port and starboard warp tension, trawl winch hydraulic pressure, net sounder winch hydraulic pressure, drum RPM, warp speed, and time are displayed digitally on the video monitor. Port and starboard trawl warp tension have a bar graph presentation, along with warp length, during shooting and hauling. Port and starboard warp length within the window is deliniated by a cursor which moves as the winches haul in and pay out during towing.

2.3 Manual Controls

During initial pay-out and final haul-in of the trawl gear, it is necessary to handle heavy trawl doors and equipment. Manual controls are provided that are always active, through interlocks, within the guard length. The ability always exists to interrupt the automated control and revert to manual by the push of a button in the event the trawl gear requires a change in position.

2.4 Automated Shooting

Once the trawl gear is properly set past the guard length, automatic shooting can commence. The computer controls the flow of hydraulic oil to the winches to pay out at maximum equal speed, port and starboard. As the trawl gear approaches the set length, an automatic deceleration ensures a smooth stop. An alarm and message warn the operator to slow the vessel to towing speed.

2.5 Automated Towing

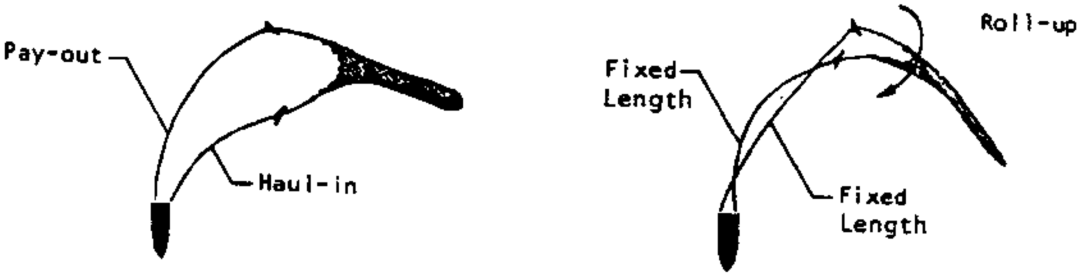
During automated towing, pressure is equalized between the two winches, equalizing warp tensions for optimum spread of the doors. Pressure is automatically adjusted to maintain warp length at the set length by incrementing pressure slightly if the actual length is greater than the set length, and decrementing pressure if the actual length is less than the set length. The microprocessor monitors pressure and position, and gradually increases tension as the net fills with fish, thus maintaining the set length within the window of operation.

In the event of a hang-up, the constant tension allows cable to pay out without damage to the trawl gear. When the cable passes the far end of the window, an alarm sounds and a warning message is displayed. The cable then continues to pay out until the vessel can be stopped and maneuvered to clear the hang-up.

In the case of a broken cable, one winch would haul in due to reduced tension. After hauling half the length of the window, the winch is then stopped automatically. Again, an alarm sounds and the warning message is displayed.

During automated towing, relative tensions are displayed along with the differential length between the port and starboard trawl warps. As the vessel moves through its six degrees of freedom, the continuous pay-out and haul-in tends to maintain the trawl net at constant velocity and height with respect to the ocean bottom. This helps to maintain the trawl net in proper configuration regardless of the motion of the vessel in a seaway.

During turning maneuvers without automated controls, tension increases in the outboard warp and decreases in the inboard warp. With automated towing, the outboard warp will pay out while the inboard warp hauls in, thus ensuring equalized tension and maintaining the mouth of the net in the maximum open position. This also helps prevent crossing of the trawl warps and subsequent roll-up of the trawl gear (see Figure 2).



EQUALIZED TENSION

UNEQUALIZED TENSION

Figure 2

2.6 Automated Hauling

Once the tow period is complete, automatic hauling commences at the push of a button. The computer controls the winches to haul in at maximum equal speed, port and starboard. Warp tension is monitored while hauling, and tension capacity of the winches is maintained only slightly greater than that required for hauling. Thus, in rough weather, the winches may stop, or even pay out, to cushion the shock of violent motions of the vessel. As the trawl gear approaches the guard length, automatic deceleration ensures a smooth stop. Final retrieval and stowage of the trawl gear is accomplished with the manual controls.

3.0 FUTURE AUTOMATED VERTICAL NET POSITIONING

3.1 Net Sounder Interface

The first step in maintaining trawl gear at either a constant height above the bottom or depth from the surface is to determine where the net is. This may be accomplished with a transducer, mounted on the headrope of the net, that sends signals up or down to determine distance to the surface or bottom. These signals can be compared to a pre-set height or depth and used to generate pay-out or haul-in commands to adjust and maintain the net at a pre-set vertical location. One problem that may occur is the net sounder encountering a school of fish so dense that it believes they are the bottom, with resultant positioning of the net over the school.

3.2 Vertical Sounder Interface

In trying to adjust the vertical position of the net to avoid pinnacles and obstructions, it is necessary to elevate the net prior to the signal being received at the net sounder. This would require a tie-in with the vertical sounder transducer on the vessel and a speed monitor, along with a determination of speed and distance to the gear. With this preliminary indication of an obstruction, the signal to haul in could be generated in time to elevate the net after inputting the vessel's speed and distance to the trawl gear. The microprocessor would be programmed to coordinate the haul-in with the signals from the hull and net transducers for the time required to clear the obstruction. Then, the original vertical position would be re-established through automatic pay-out. Using this same method, the microprocessor could be programmed to adjust the net vertically for the various species of fish that have a tendency to either dive or rise as the net approaches (see Figure 3).

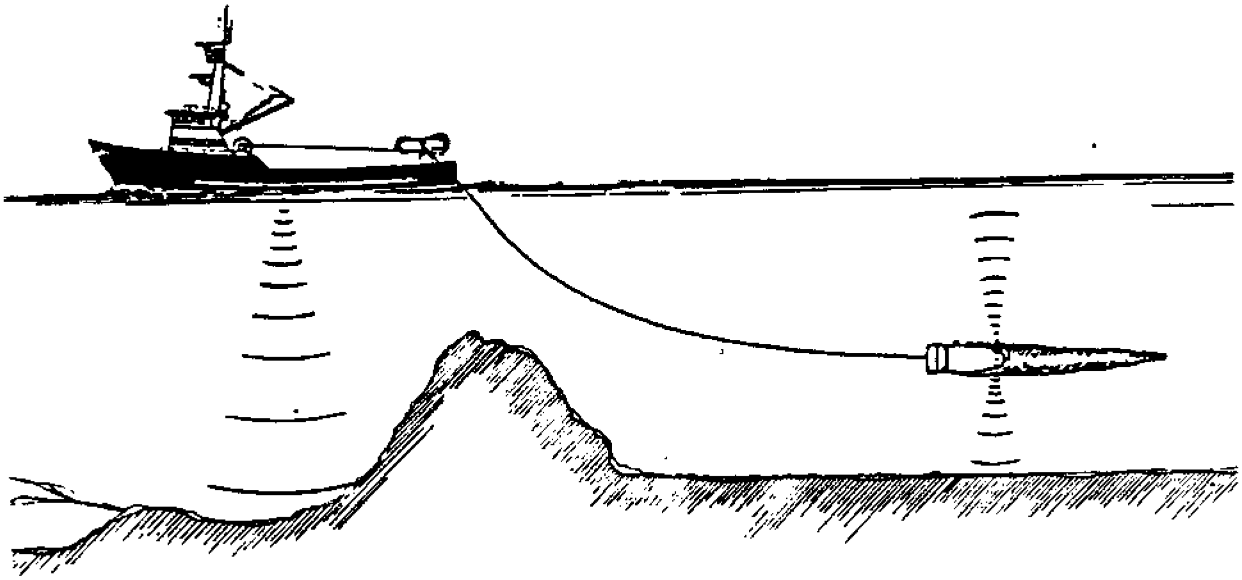
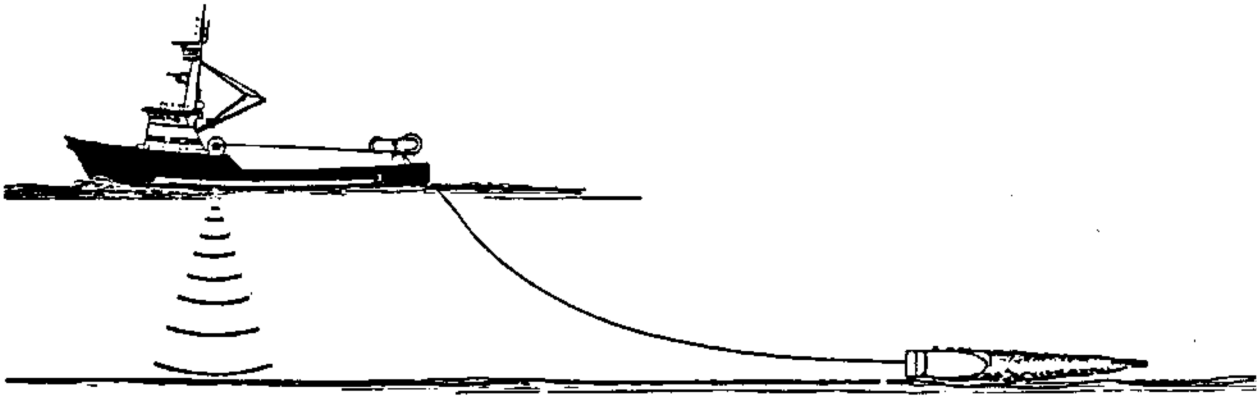


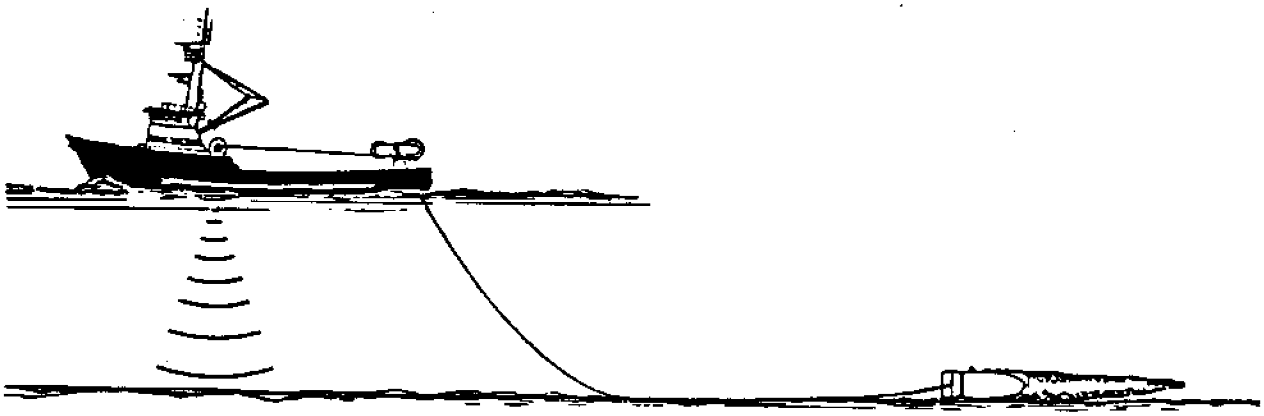
Figure 3

3.3 Bottom Trawling

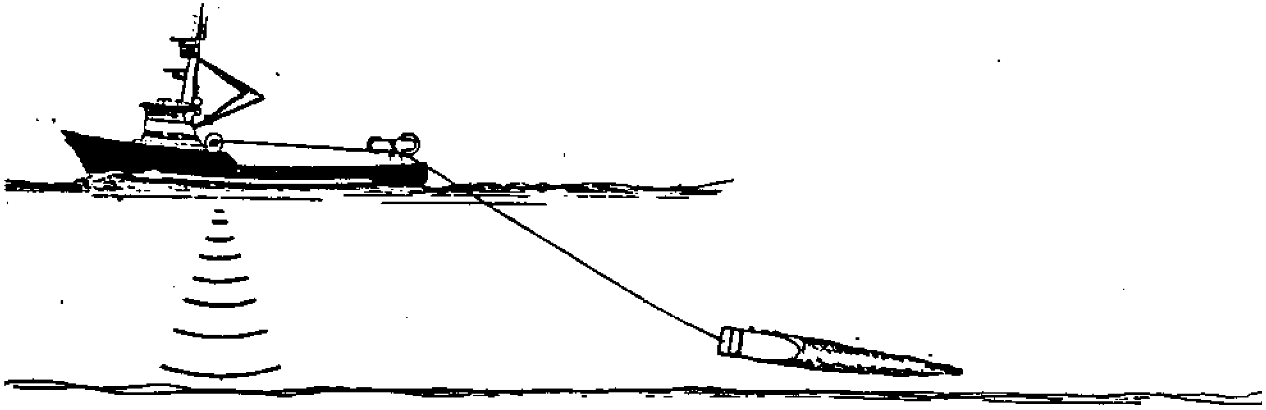
For a given depth and configuration of bottom trawl gear, there is an optimum warp length. Insufficient length tends to lift the gear off the bottom while excessive length allows the warp to drag on the bottom. Automated controls could be programmed to calculate the proper catenary and maintain the optimum length at any depth through a tie-in to the vertical depth sounder (see Figure 4).



Trawl warps proper length - cables horizontal at trawl door



Trawl warps too long - cable drags on bottom



Trawl warps too short - net lifts off bottom

Figure 4

3.4 Main Engine Throttle and Propeller Pitch Control

The case may exist where vertical positioning of the net must be accomplished by vessel speed control in addition to haul-in and pay-out. In this event, computer-generated commands for variation in main engine speed or propeller pitch would be required. A routine could also be developed to maximize the vertical opening of the net through both length and speed adjustments. Developing the program for this control will take experimentation to determine the varied effects of vessel speed and trawl warp length on trawl net vertical position and response time.

4.0 FUTURE AUTOMATED THREE-DIMENSIONAL NET POSITIONING

Locating fish with a sonar and linking their depth and bearing information with the trawl winch microprocessor may be the next step in automated fish harvesting. In conjunction with the sonar bearing readings of the fish mass, the magnetic or gyroscopic autopilot could be interconnected to automatically navigate the vessel on an intercept course. Sonars coupled with current indicators at various depths could be used to calculate drift of the trawl gear and adjust the vessel's heading to compensate. The trawl-gear-locating capabilities of the sonar would facilitate positioning the net with respect to drift due to cross currents.

Problems with sonar recognition that must be dealt with include reflection and refraction through various thermoclines, pressure gradients, and salinity gradients. These tendencies to produce false readings may dictate use of the sonar only as a preliminary net positioning device, with final position adjustments made utilizing the hull and net sounders.

The result of three-dimensional control of the trawl net, coupled with the fish-locating capabilities of sonar, is a reduction of the fisherman's decision-making requirements. This concept may be slow in developing, since experienced fishermen are capable of performing the same tasks with nearly the same reliability and precision as the computer. Perhaps the first stage of this development will not be fully automated control, but rather a display of the options available to the captain to achieve specific results. Becoming totally dependent upon a machine does have its drawbacks, especially when the decision-making process involves input from the unpredictable forces of nature. The failure of any one element in the link of control for automated fishing would require immediate input from the fisherman to avoid either lost catch or damaged gear.

5.0 ANCILLARY INTERFACES TO AUXILIARY MECHANICAL AND ELECTRONIC DEVICES

5.1 Catch Indicators

Normally positioned along the length of the cod end of a trawl net are sensors that indicate the extent of filling of the net. Safeguards to protect against over-filling could be incorporated to automatically activate elevation of the net above the fish mass and commence haul-in.

5.2 Trawl Engine Governor

With vessels that utilize an auxiliary generator to power the trawl hydraulics, either electrically or directly, there is sometimes the possibility of overloading and stalling the engine, depending upon auxiliary electrical loads. Engine loading could be monitored and an interface between the engine governor and trawl winch controls could be set up to limit the delivered hydraulic horsepower.

5.3 Navigation Aids and Catch Histories

Expandable microprocessor memory will allow the input of either Omega, Loran-C, or Satnav coordinates at the beginning and end of each tow. The National Marine Electronics Association (NMEA) has been developing standards (NMEA 0183) for the transmission of data using formats acceptable to manufacturers of all marine electronics. This information, when coupled with an internal clock, will aid in developing catch histories. These catch histories have proven to be a decided advantage to foreign fleets that have enjoyed unrestricted fishing over the past years.

5.4 User Inputs and Shore-Based Communication

A user interface with the microprocessor memory would allow input of comments by the fisherman relative to estimated quantity of various species caught. Satellite communication connections to allow transmission of this data would permit shore-based processors to monitor harvesting and determine the proper timing for return to port for optimum plant operations.

5.5 Miscellaneous

Engine speed, exhaust temperatures, fuel consumption, sea temperatures, vessel speed, trawl warp tensions, and hydraulic pressures are all examples of data that could be recorded during trawling operations for later study and advancement of the developing body of knowledge.

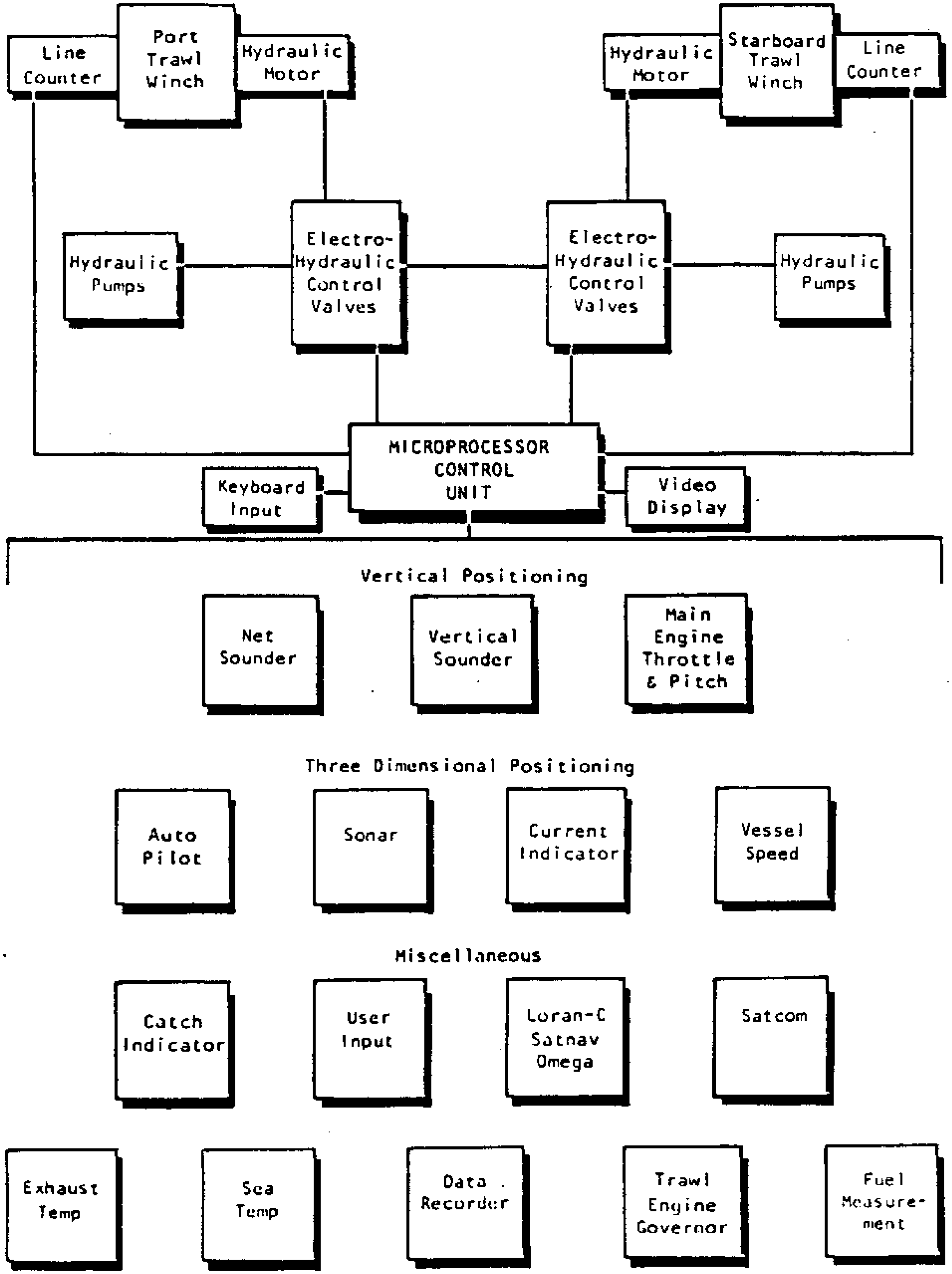

Line Counter
Port Trawl Winch
Hydraulic Motor
Hydraulic Motor
Starboard Trawl Winch
Line Counter
Hydraulic Pumps
Electro-Hydraulic Control Valves
Electro-Hydraulic Control Valves
Hydraulic Pumps
MICROPROCESSOR CONTROL UNIT
Keyboard Input
Video Display
Vertical Positioning
Net Sounder
Vertical Sounder
Main Engine Throttle & Pitch
Three Dimensional Positioning
Auto Pilot
Sonar
Current Indicator
Vessel Speed
Miscellaneous
Catch Indicator
User Input
Loran-C Satnav Omega
Satcom
Exhaust Temp
Sea Temp
Data Recorder
Trawl Engine Governor
Fuel Measurement

Figure 5

6.0 CONCLUSIONS

The captain, who is in full command of the trawling operations, must assemble information from a variety of mechanical, hydraulic, and electronic devices in order to adjust the trawl winch controls, engine speed, and vessel direction to maximize the amount of fish caught. Experienced fishermen, familiar with their vessels and gear, make decisions based on logic gained through previous trial and error. Inexperienced fishermen, however, make decisions based on trial and error only.

The goal of computerization and interface of input devices is to assist fishermen in making command decisions based on clear logic, regardless of their experience or knowledge of gear/vessel interaction. This is accomplished by taking the trial-and-error decision-making process and applying pre-programmed logic from the microprocessor to the input data. Action is then taken by the output devices in the form of trawl winch and vessel control to properly set, position, and haul the net. This automation relieves the captain of the need to worry about details, thus freeing him to concentrate on the business of catching fish.

The current value of automated controls lies in their ability to properly position the net and protect valuable gear. Crew fatigue can be reduced as the computer takes on more of the repetitive tasks of conventional fishing. Computers cannot think, but they are precise and untiring at control tasks, being as efficient at the end of a watch as at the beginning. The end result is the delivery of fresher fish with lower operational costs. Development of interfaces with electronic equipment is the next logical step. However, automated controls can be a valuable tool only when used by technically qualified personnel and combined with good fishing and seamanship skills.

WRITTEN CONTRIBUTION

S. M. Calisal

I would like to comment on the concept "value" presented by Mr. Lundgren. If seems that this parameter could be used by the ship operator to calculate an acceptable speed for a fishing scenario. The determination of vessel speed for departure and return legs seems to be critical in the profitability calculations. To this end Canadian Department of Fisheries and Oceans and the University of British Columbia are developing a computer program to be used by or for the fishermen. This program treats the fishing vessel as a complete system. The objective is not to tell the fisherman what speed they should sail at, but to give them a measure of dollars saved by a change of speed or propulsion hardware. Information on possible savings by the installation of controllable propellers, nozzles, and new propellers on an existing boat are given to the fisherman.

The application of this computer program showed us that the operational profile on the fishing scenario is very important in the calculation of the fuel consumption and in the optimization of the boat parameters. It is also clear that the classical naval architectural concepts based on the separate optimization of the hull, machinery and the propeller is not the proper way to optimize a boat. Finally there is no single "optimum" boat unless an exact operational profile is established. It seems that if the operational profile is not included in the calculation of the parameters for an optimum boat the problem is not a well posed one.

To facilitate the input of the fishing scenario a map of the Pacific Canadian Coast is plotted on the visual display of the computer. The fisherman then inputs the possible fishing route. All the technical calculations are transparent to the user.

It seems that the usage of the concept "value" defined in this conference could very easily establish the proper ship sheep for the operator.

WRITTEN CONTRIBUTION

Christopher Tupper

Regarding the Koyama's equation calculation: The net resistance plots given in the paper seem to be quite a bit flatter in shape than the squared relationship between resistance and velocity as is assumed in Koyama's equation. I have seen this in other recent tank test data for trawl resistance. Is there a reason for this? Should we be questioning our assumptions that net resistance follows this simple quadratic relationship (DRAG V^2)?

WRITTEN CONTRIBUTION

Cliff Goudey

Koyama's equations can be useful for predicting the drag of trawls if the design under consideration is similar to those studied by Koyama and is rigged and operated in a similar way.

Extension of his work to other configurations can prove unreliable particularly when important information on the trawl is neglected or unavailable. A major difference between the Japanese trawls of Koyama and those of Taber is the type of trawl doors used.

The upright curved trawl doors used by Koyama are of aspect ratio 1.0 to 1.5 and are considerably more efficient than those used by the U.S. trawling fleet. While the drag coefficient of these doors is not as low as the midwater door data from Sharfe, it is far lower than flat door performance of $C_D = 0.8$ to 1.0.

Since trawl door resistance can account for 30 to 40 percent of total trawl system drag, the approximately 3 fold discrepancy in C_D used by Dr. Latorre would more than account for the errors revealed by his analysis.

It is tempting to look for a second power relationship between trawl resistance and speed. Test results seldom cooperate.

The explanation for this is several fold. First; gross trawl dimensions change with increased speed reduced headrope height, for example, causes changes in angle of attack of netting panels.

Second; increased speed results in higher twine stresses and elongations effecting not only mesh size but also reducing twine diameters.

Third; netting twine drag coefficients tend to decrease with increased reynolds number, at least in the range of practical importance.

The fourth and possibly most important explanation is the fact that bottom trawl resistance has a component of bottom friction which is somewhat independent of speed. This causes plots of drag versus speed to have a y-intercepts well above the origin.

PART III

FISHING VESSEL DESIGN AND CONSTRUCTION

FISHING VESSEL DESIGN CURVES

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Abstract

Preliminary design curves are presented based on data from Maritime Administration United States Fishing Fleet Improvement Act files for fishing boats built between 1960 and 1970. The curves are developed in both metric and English measurement systems and are arranged in a logical and easily utilized format. Trends shown on the curves are briefly analyzed and a sample design is presented to demonstrate the use of the curves in early stage design. The paper provides the naval architect with the means to quickly determine sizes and weights of technically feasible fishing boats for follow-on use in developing construction cost estimates.

Introduction

During the 1960s, a considerable number of fishing vessels were constructed under the United States Fishing Fleet Improvement Act. To support the Act, technical files for these vessels were created and maintained in the Maritime Administration's Division of Small Ships. Sometime after 1970, these files were sent to Maritime Administration archives for permanent storage.

In the past 15 years, the Maritime Administration continued to participate in occasional fishing vessel designs, in particular serving as the design agent for the National Oceanic and Atmospheric Administration (NOAA). Examples of NOAA designs done by the Maritime Administration are the S1-M-MA124a 120-ft Crabber/Trawler Research Vessel and the PD-226 90-ft Shrimper Research Vessel. Since the old fishing boat files were available as a result of being called out of archives for the NOAA designs, an opportunity existed to collect and reduce the data contained therein. The design curves in this paper are the result of such an effort undertaken during the spring of 1980.

The curves themselves are based on data from 39 fishing boats built in the United States between 1961 and 1972. Of course, this means that the curves are 10 to 20 years out of date, which is a significant deficiency. The major technical shortcoming associated with the outdated curves is the complete absence of glass-reinforced plastic vessels and the inclusion of only one all-aluminum boat. However, the wide variety of vessel types that constitute the data base tends to offset the age of the information and its lack of breadth as far as hull structural materials are concerned. Also, the evolutionary nature of shipbuilding technology development (as is evident in the general consistency of the data over the 1960 to 1970 time frame described in the design graphs) should enable the use of the curves as a good baseline, especially for early design purposes. The format of the design curves is amenable to easy updating by individual designers, consultant organizations, or shipyards.

Nomenclature

B - maximum beam, molded

C_B - block coefficient corresponding to the ready-for-sea displacement $= \nabla/(L \times B \times T)$

Ci - transverse waterplane inertia coefficient $= I_{transverse}/(L \times B^3)$

Ci_L - longitudinal waterplane inertia coefficient $= I_{longitudinal}/(L^3 \times B)$

C_p - longitudinal prismatic coefficient $= C_B/C_x$

Cw - waterplane coefficient $= A_{waterplane}/(L \times B)$

C_x - midship section coefficient $= A_{midships}/(B \times T)$

C - structure/machinery/outfit/ballast - weight coefficients = weight/cubic number

D - molded depth, from baseline at amidships to main deck

GM - metacentric height

KG - height of center of gravity above baseline

L - length, on design waterline

LCG - longitudinal center of gravity relative to the forward perpendicular

T - design draft, molded, at amidships

Standards of Measurement

For the most part, the design curves are presented in the metric, or International System (SI) of units, although secondary axes in English units have been maintained. The standard (SI) units used herein are given below:

Length - Meters

Volume - Cubic Meters

Displacement and Weight - Metric Tons, Equal to 1000 Kilograms. (1 metric ton = 1 m^3 fresh water = 0.976 m^3 sea water)

Power - Kilowatts

Speed - Knots

All ship characteristics are based on the "ready-for-sea" condition, consisting of lightship, crew and effects, stores, fuel, lube oil, fresh water, and ice (if carried).

The characteristic hull length measurement is length on the design waterline in the ready-for-sea condition. (Most of the design data available had 10 stations in length on waterline.) Draft, depth, and hull coefficients are molded values based on the baseline at amidships at the intersection of hull and keel.

The Maritime Administration's weight classification system is employed with the steel category modified to "structure" to account for wooden vessels. The following breakdown is typical:

STRUCTURE
  Hull
  House
  Fastenings (for wooden vessels)
  Welding

OUTFIT
  Auxiliary machinery
  Piping with liquids
  Electrical
  Joiner work
  Furniture
  Hull outfit
  Fishing outfit
  Spars and rigging
  Paint, cement, and caulking

MACHINERY
  Propulsion machinery
  Machinery outfit

BALLAST AND SOAKAGE
  Taken together for wooden vessels

Cubic number equals (LxBxD)/100 in English units, and has been soft-converted to metric units as (LxBxD)/2.834. To obtain the lightship category weight in long tons, the product of the cubic number and the weight coefficient is calculated. For metric tons this product must be multiplied by 1.0163, although for the purposes of early design the 1.0163 factor could be ignored.

Design Curves

The design figures have been arranged to allow an orderly progression from initial vessel requirements to a first-cut set of design characteristics. When using the curves, judgement should be employed to account for particular ship requirements. The individual data points have been plotted with indicators of vessel type to facilitate this process. The symbols given below are reproduced on the first figure, hold volume versus length:

○ - wood side trawler, or scalloper
□ - steel side trawler, or scalloper

▽ - steel stern trawler
+ - steel shrimper
△ - steel seiner
◊ - aluminum multipurpose
× - steel menhaden

Where several points have the same location, a numeral corresponding to the total number of coincident points has been placed adjacent to a single, exactly positioned symbol.

The initial step in using the curves is to enter with a hold volume requirement and select a length and then beam and depth from Figures 1 and 2. With length, beam, and depth, the cubic number can be calculated, and Figures 3 through 7 consulted to estimate the weights and centers of the lightship components. The designer is left to his own judgment to add reasonable weight and KG margins to the sum of the component weights and KGs determined from these curves. It should be noted, however, that the curve of lightship KG/D versus cubic number in Figure 7 includes a KG rise.

Figure 8, propulsion power plotted against ship speed and displacement, should be used for gross estimation only. It assumes a propulsion coefficient on the order of 0.50 to 0.60. Figure 9, machinery weight versus power, is provided as another means, besides cubic number, of estimating machinery weight.

Other hull coefficients and stability parameters are given in Figures 10 through 16 in order to allow trim and stability conditions to be evaluated without hydrostatic curves. Design trends are also presented for bow height, drag of keel, and freeboard in Figures 17, 18, and 19, respectively.

Regression Equations

The following equations mathematically represent the design curves given in the figures. They are linear least-square fits of the plotted data, although logarithmic or quadratic transformations may be involved. In the case of the latter, to avoid the second-order least-squares computations, a location for the curve minima or maxima was selected "by eye".

- Hold Volume versus Length (Figure 1):
  $\text{Vol} = 0.0139\ (L)^{2.77}$, m^3
- L/B versus Length (Figure 2):
  $L/B = L/21.7 + 2.48$
- B/D versus Length (Figure 2):
  $B/D = L/60.2 + 1.40$
- C Structure, Steel, versus Cubic Number (Figure 3):
  $C\ \text{Struct} = \text{Cubic No.}/50{,}000. + 0.326$
- C Structure, Wood, versus Cubic Number (Figure 3):
  $C\ \text{Struct} = \text{Cubic No.}/2{,}500. + 0.30$
- KG/D Structure versus Cubic Number (Figure 3):
  $KG/D = \text{Cubic No.}/14{,}630. + 0.771$

- C Outfit versus Cubic Number (Figure 4):
  C Outfit = Cubic No./17,140. + 0.196
- KG/D Outfit versus Cubic Number (Figure 4):
  KG/D = Cubic No./8,820. + 1.04
- C Machinery versus Cubic Number (Figure 5):
  C Mach = Cubic No./27,300. + 0.0537
- KG/D Machinery versus Cubic Number (Figure 5):
  KG/D = $(\text{Cubic No.}-400)^2/1{,}309{,}000. + 0.494$
- C Ballast and Soakage versus Cubic Number (Figure 6):
  Wood - C Ball = -Cubic No./10,980. + 0.163
  Steel - C Ball = -Cubic No./10,670. + 0.104
- KG/D Ballast and Soakage versus Cubic Number (Figure 6):
  Wood - KG/D = -Cubic No./3,250. + 0.350
  Steel - KG/D = -Cubic No./7,270. + 0.212
- Lightship KG/D and LCG/L versus Cubic Number (Figure 7):
  House Forward
    KG/D = Cubic No./8,000. + 0.790
    LCG/L = -Cubic No./25,000. + 0.510
  House Aft
    KG/D = Cubic No./12,000. + 0.723
    LCG/L = -Cubic No./30,000 + 0.552
- Machinery Weight versus Power (Figure 9):
  Weight = (propulsion power, kw)$^{1.332}$/308.8, metric tons
- B/T versus L (Figure 10):
  B/T = -L/103.4 + 2.85
- C_p versus L (Figure 11):
  $C_p = -L/1175. + 0.662$
- C_B versus L (Figure 11):
  $C_B = L/400. + 0.440$
- GM/B versus L (Figure 12):
  GM/B = -L/400. + 0.185
- KB/T versus C_B (Figure 13):
  $KB/T = -0.467\ C_B + 0.857$
- Cw versus Cp - (Figure 14):
  Cw = 0.65Cp + 0.395
- Bow Height versus L (Figure 17):
  Height = 1.54 $(L)^{0.448}$, meters
- Drag of Keel versus L (Figure 18):
  Drag = $-(L - 32.5)^2/360.9 + 1.176$, meters
- Freeboard versus L (Figure 19):
  Seiners with long upper decks
    F'b'd to main dk. = L/21.95 - 1.625, meters
    F'b'd to upper dk. = -L/55.6 + 3.66, meters
  Other
    F'b'd to main dk. = L/35.2 + 0.270, meters

Analysis of Design Trends

The majority of the trends displayed in the figures are reasonable at first glance; however, the following are considered to warrant special attention:

Figure 1: Hold volume is assumed to be the principal design requirement. Arguments can be made in favor of having alternative principal requirements, such as installed horsepower or working-deck length and area; however, hold volume appears to result in a high correlation of basic hull characteristics for the entire spectrum of fishing boat types.

Figure 3: Wood is measurably heavier than steel as a structural material, and the difference in weight increases in proportion to the length. On the other hand, the KGs for wooden boats are the same or less than those of equivalent steel hulls. The single all-aluminum hull (some of the steel hulls have aluminum or wooden houses) is substantially lighter than steel with a slightly higher KG.

Figure 7: The KG line for house-forward boats is 0.08 D to 0.10 D higher than house-aft vessels across the cubic number range. The corresponding LCG curve for house-forward hulls is about 0.05 L farther forward than that for house-aft boats.

Figure 11: The trend for prismatic coefficient, C_p, is nearly constant versus length, while block coefficient, C_B, increases with length. Therefore, it appears that larger vessels tend to have higher midship section coefficients.

Figure 12: The ratio of metacentric height to beam (GM/B), decreases with increasing length. This trend represents a slight decrease in GM as ship length increases.

Figure 18: The quadratic curve fit for this figure is a result of most hulls in the 20- to 40-m range having a 1.0- to 1.4-m drag. Shorter boats have less absolute drag of keel, but a similar proportion relative to their length. As the hulls approach the size of ships (greater than 40 m), the drag of keel decreases, and for some vessels is zero.

Conclusions

A set of fishing vessel design curves is presented based on data from Maritime Administration technical files generated under the United States Fishing Fleet Improvement Act between 1960 and 1970. The curves are arranged in a logical format that is suitable for early design use. An example concept design using the curves is shown in the Appendix.

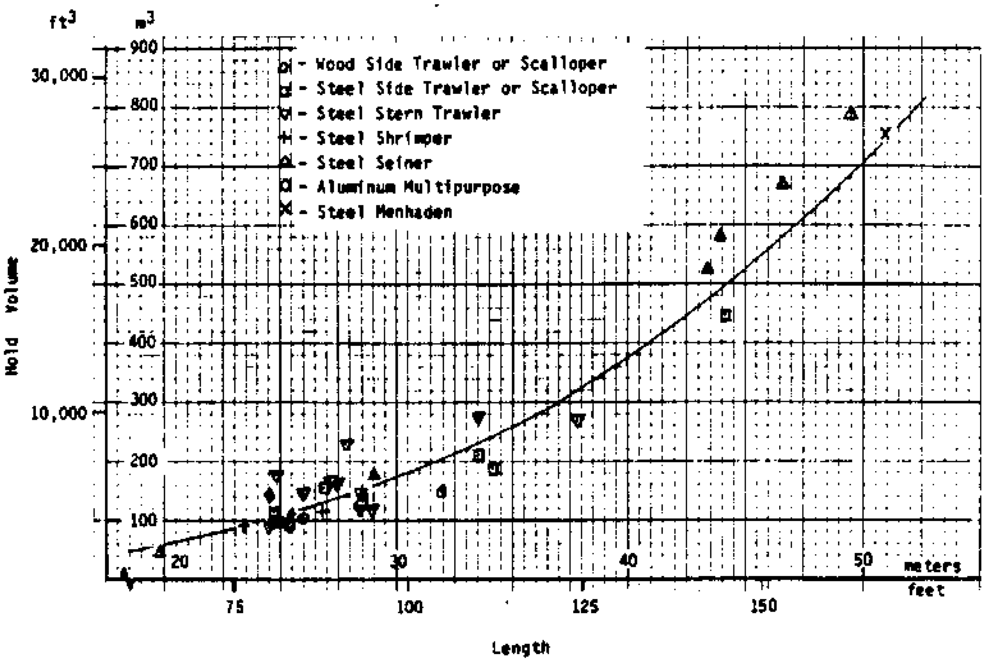


Figure 1. Hold Volume vs. Length

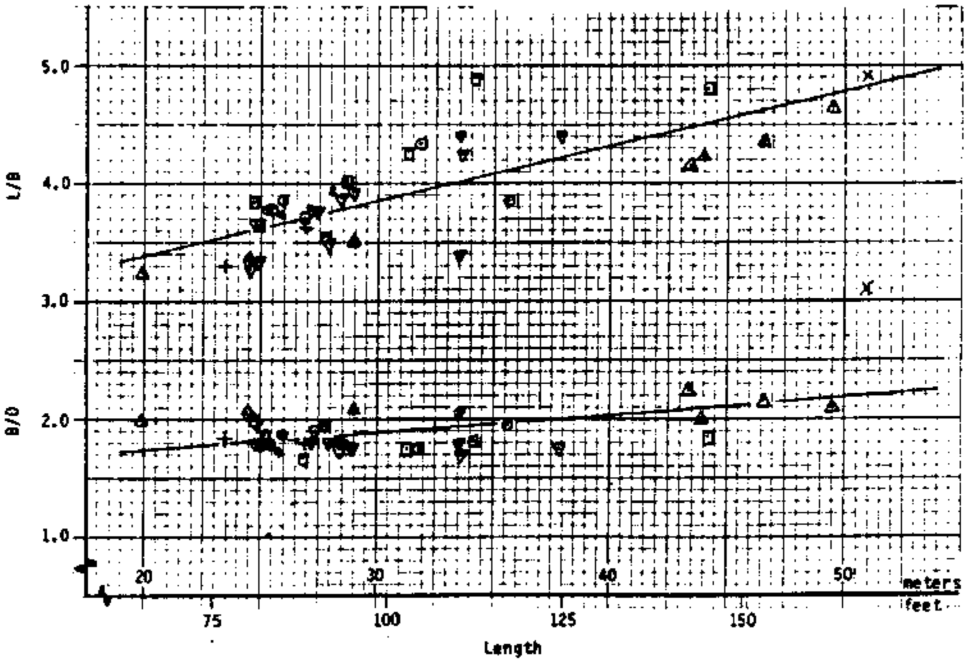


Figure 2. Molded L/B and B/D vs. Length

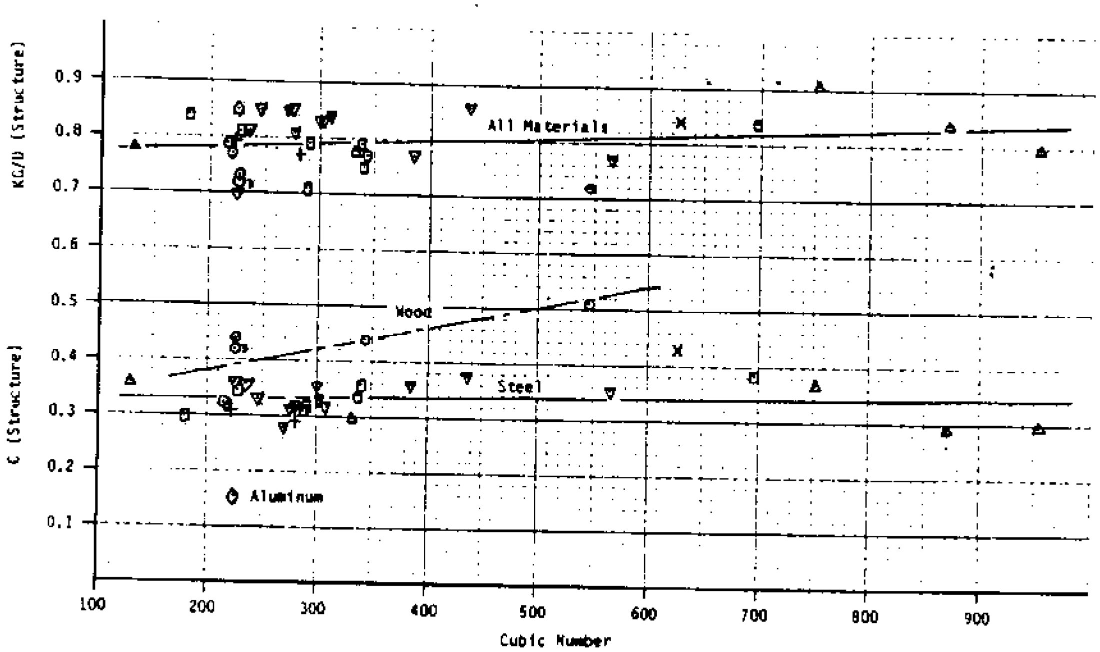


Figure 3. Structure Weight Coefficient and KG/D vs. Cubic Number

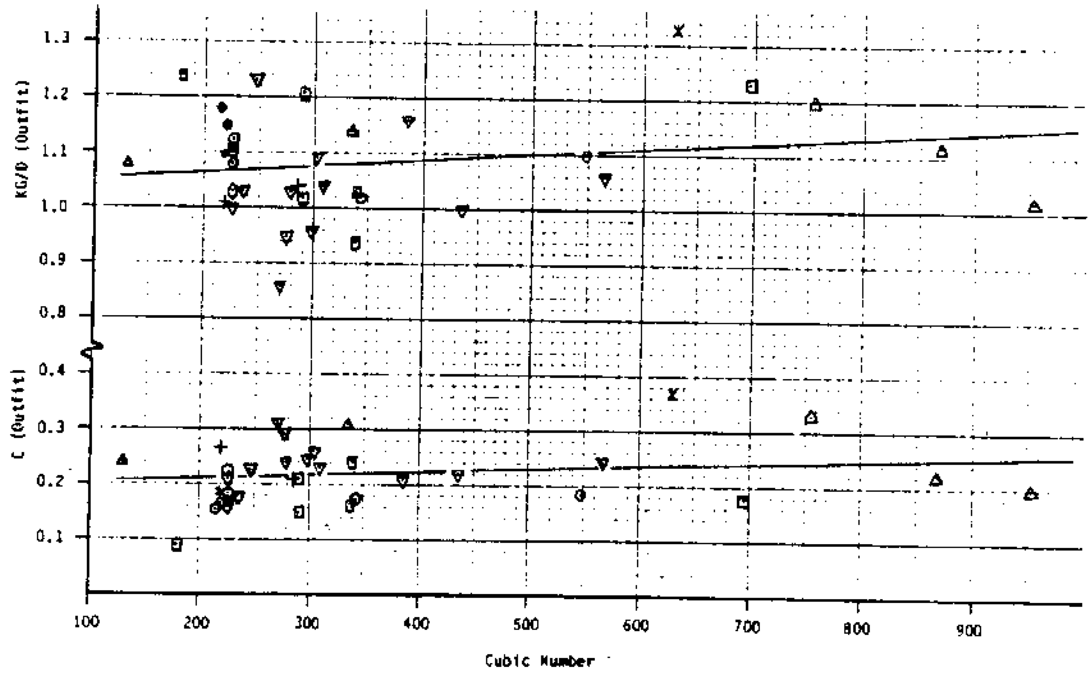


Figure 4. Outfit Weight Coefficient and KG/D vs. Cubic Number

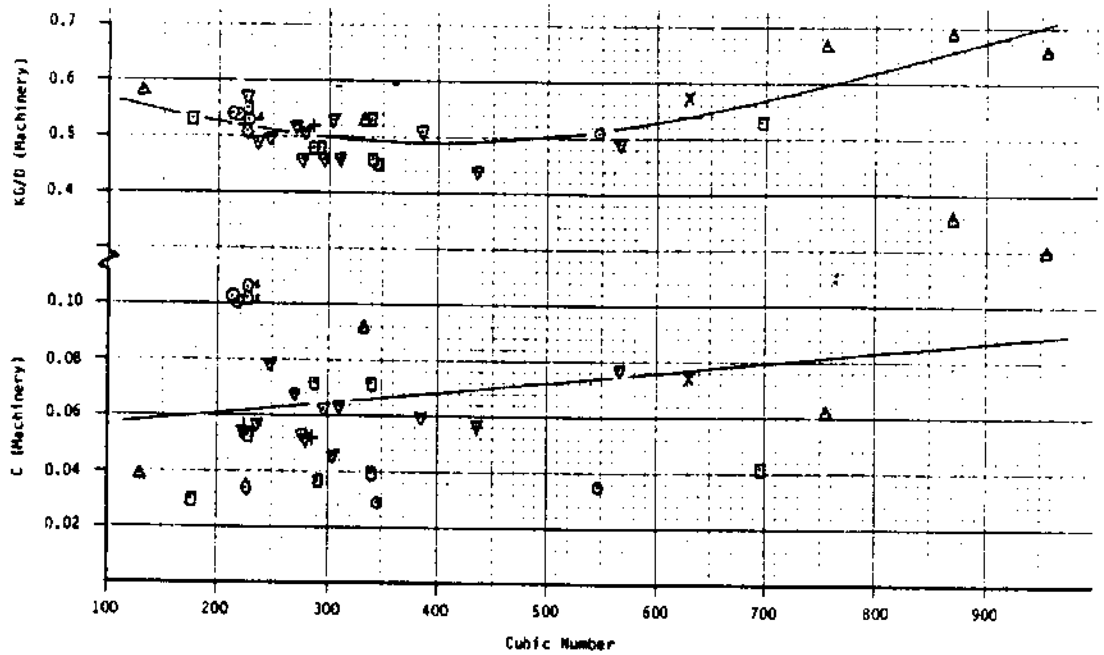


Figure 5. Machinery Weight Coefficient and KG/D vs. Cubic Number

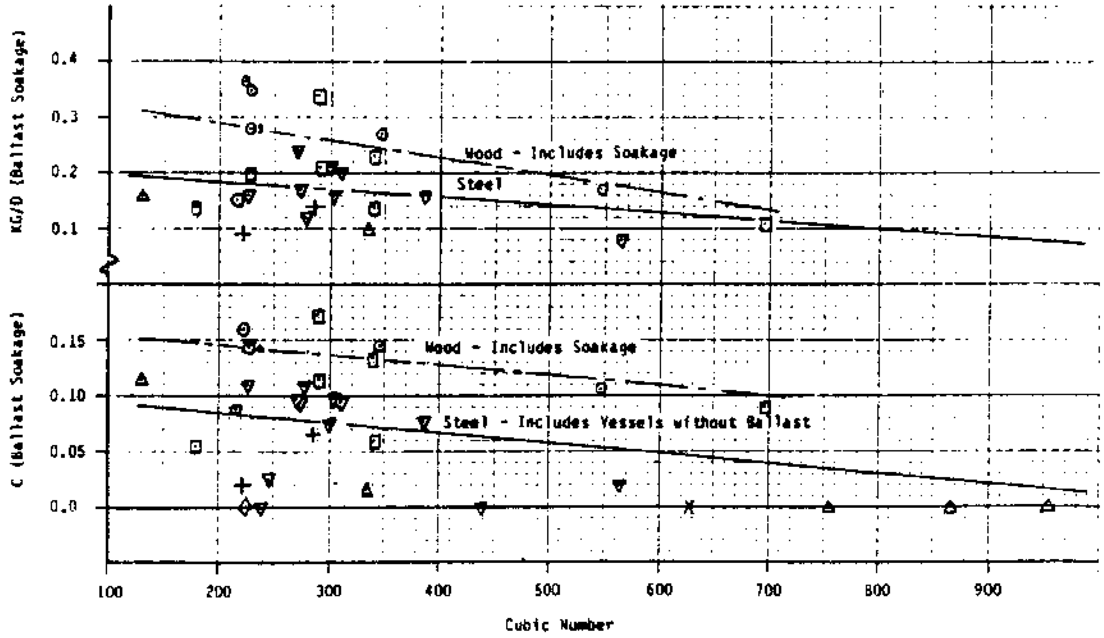


Figure 6. Ballast and Soakage Weight Coefficient and KG/D vs. Cubic Number

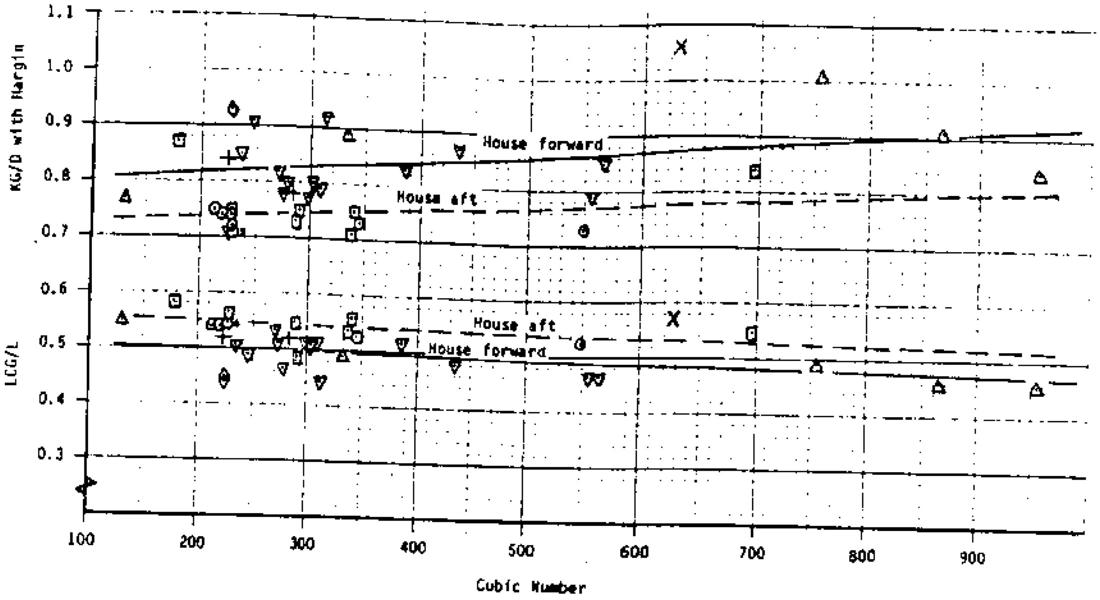


Figure 7. Lightship KG/D and LCG/L vs. Cubic Number

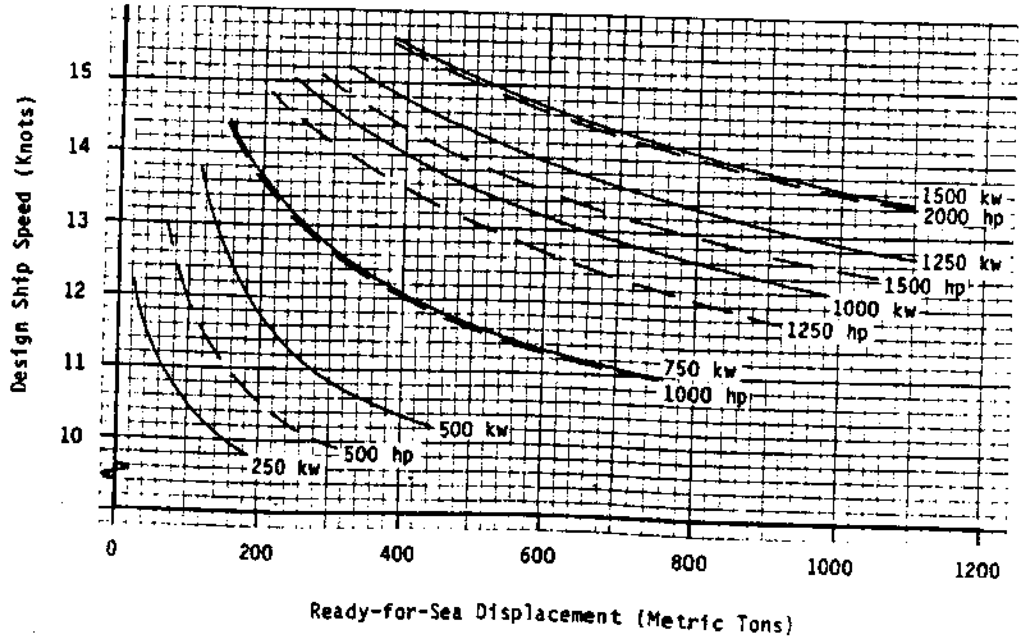


Figure 8. Installed Maximum Continuous Rated Power vs. Displacement

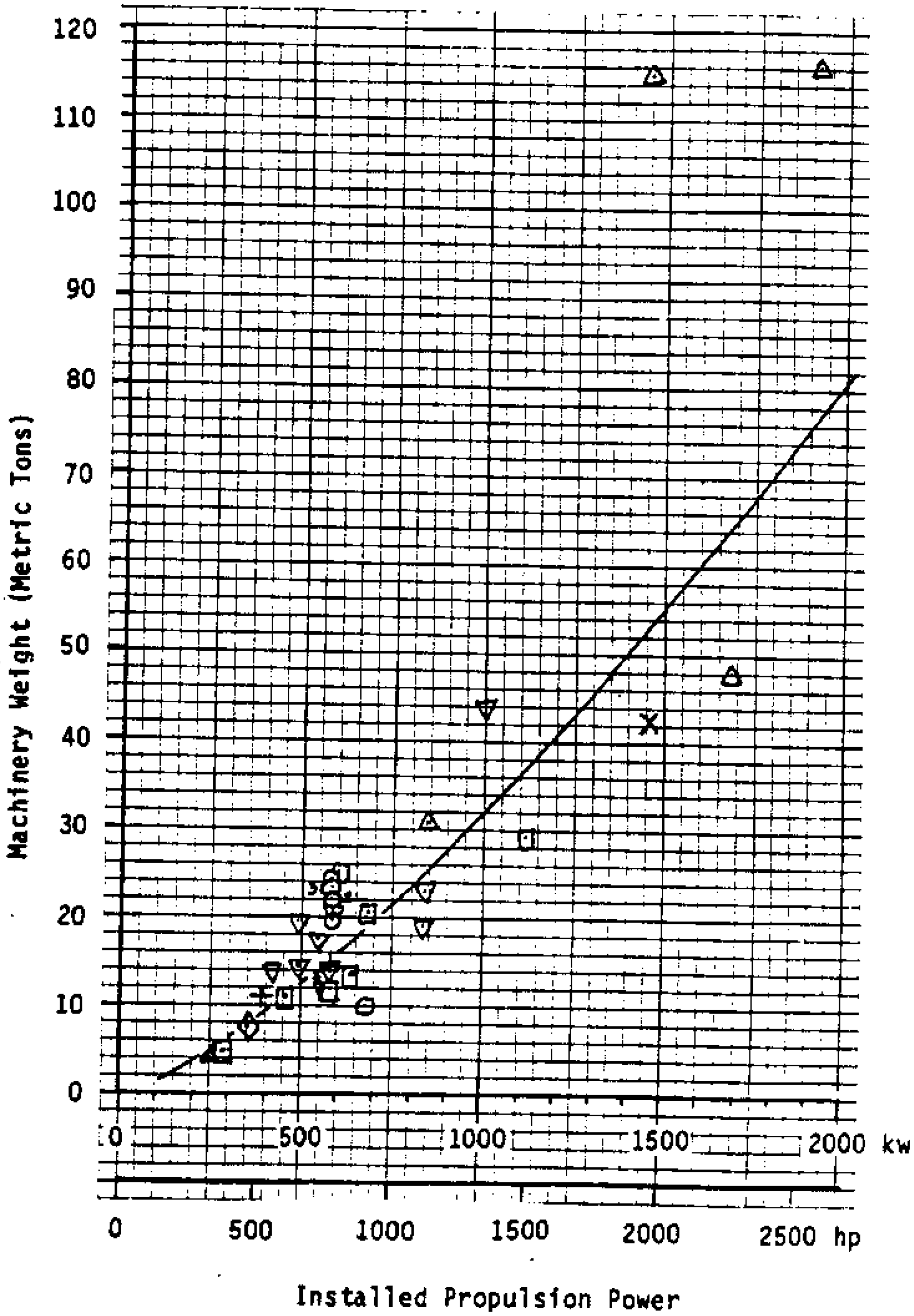


Figure 9. Machinery Weight vs. Power

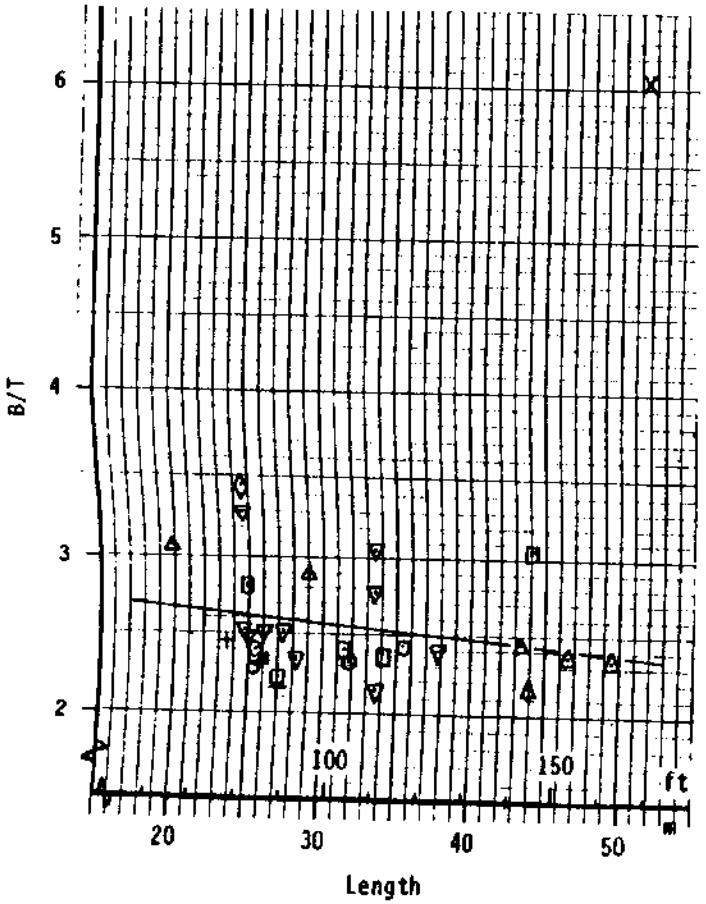


Figure 10. B/T vs. Length

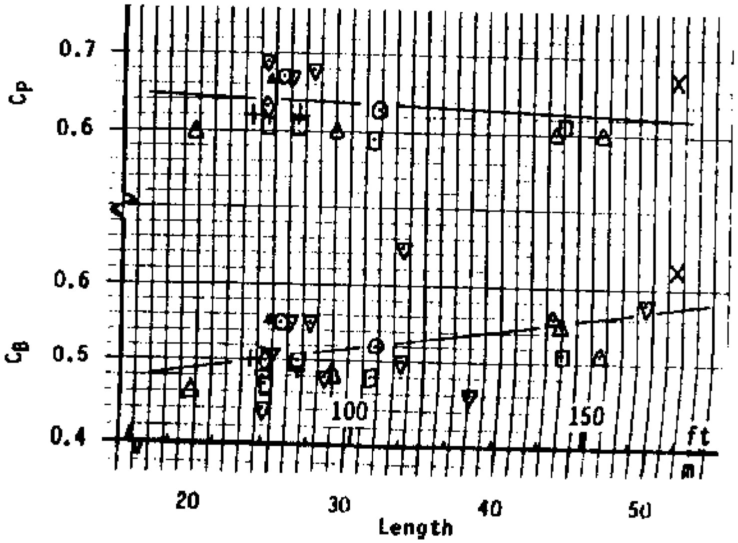


Figure 11. C_B and C_p vs. Length

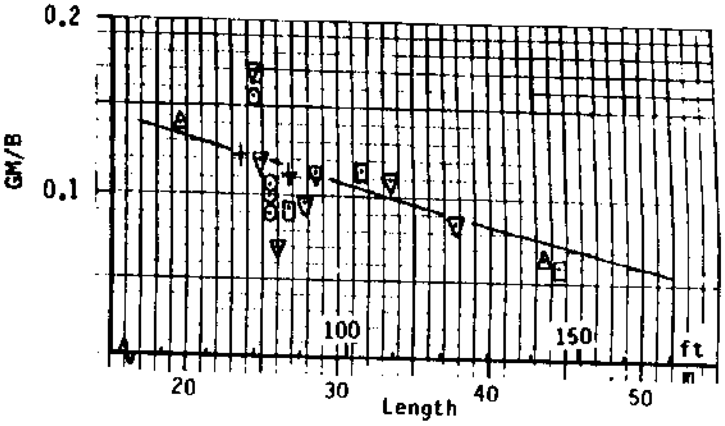


Figure 12. GM/B vs. Length

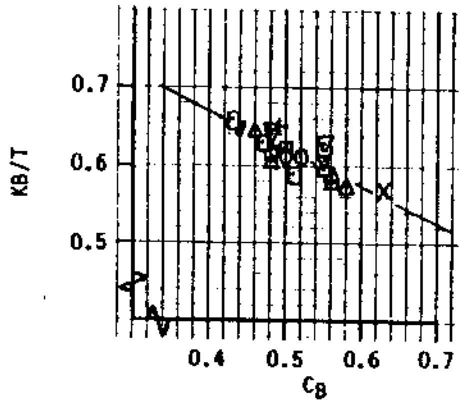


Figure 13. KB/T vs. C_B

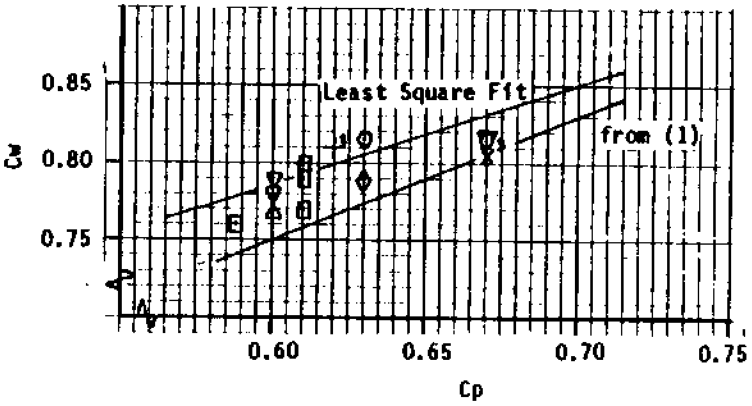


Figure 14. Cw vs. Cp

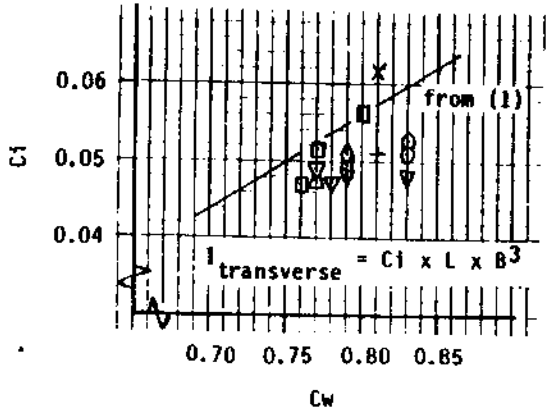


Figure 15. Ci vs. Cw

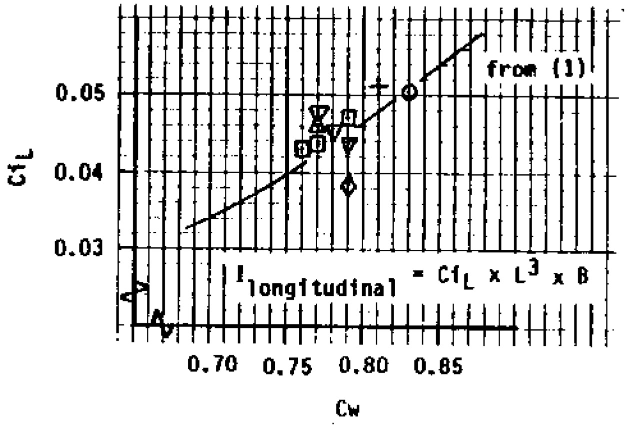


Figure 16. Ci_L vs. Cw

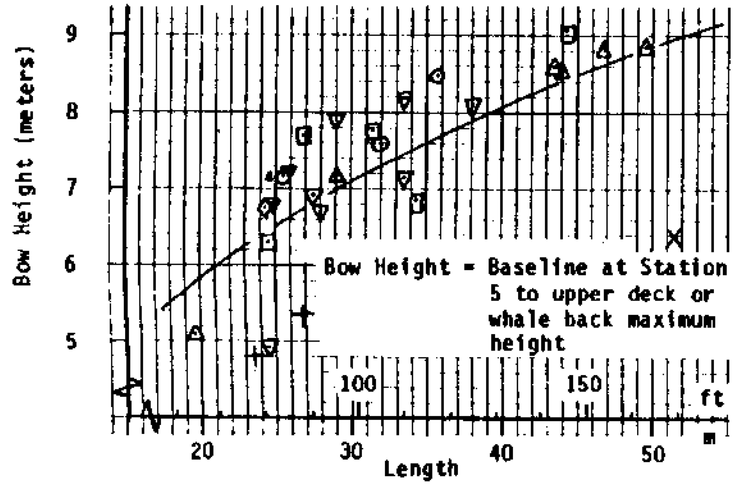


Figure 17. Bow Height vs. Length

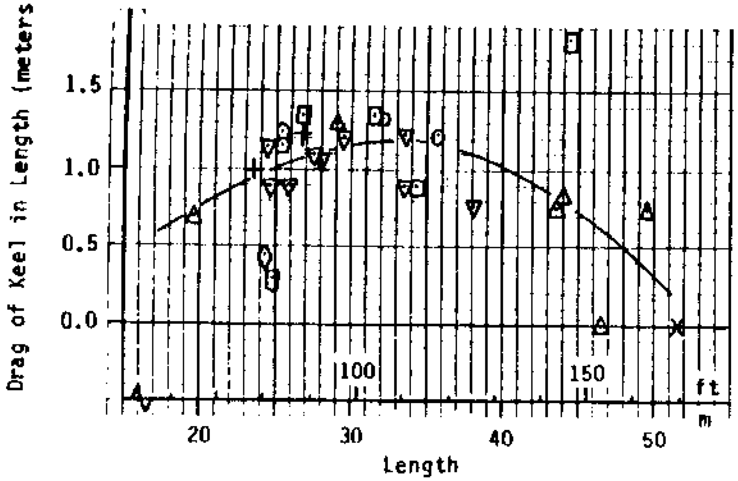


Figure 18. Drag of Keel vs. Length

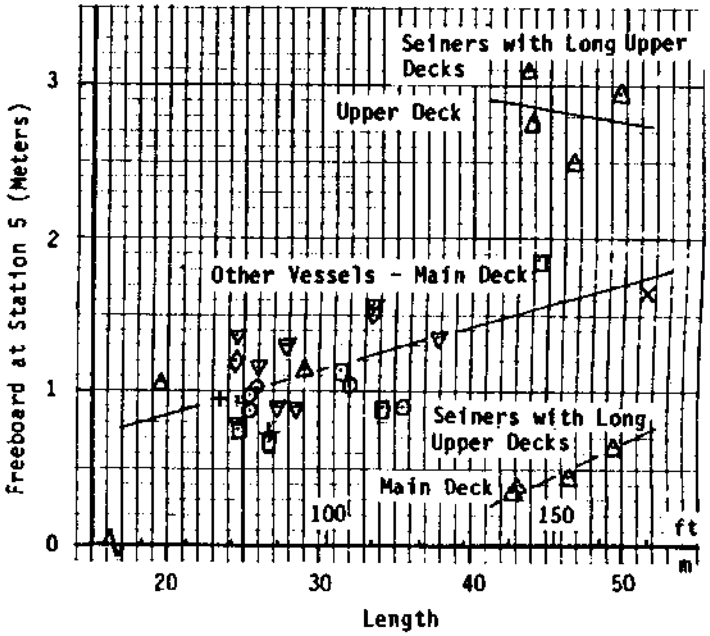


Figure 19. Freeboard vs. Length

APPENDIX - EXAMPLE DESIGN USING CURVES

Procedure

Develop characteristics to support a construction cost estimate for a steel stern trawler. The required hold volume is 100 cubic meters.

Step 1: Estimate L from Figure 1.
For hold volume = 100 m^3, obtain L = 24.0 m

Step 2: Estimate initial B and D from Figure 2. With L = 24.0 m, L/B = 3.60 and B/D = 1.80, yielding B = 24.0/3.60 = 6.67 m and D = 6.67/1.80 = 3.70 m.

Step 3: Calculate cubic number and lightship component weights and centers.
Cubic no. = L x B x D/2.834 = 209.0

Structure (Figure 3)
Cs = 0.33 KG/D = 0.785
Weight = 0.33 x 209.0 x 1.0163 = 70.1 metric tons
KG = 0.785 x 3.70 = 2.90 m

Outfit (Figure 4)
C_o = 0.21 KG/D = 1.07
Weight = 44.6 metric tons
KG = 3.96 m

Machinery (Figure 5)
Cm = 0.062 KG/D = 0.52
Weight = 13.2 metric tons
KG = 1.92 m

Ballast (Figure 6)
Cb = 0.085 KG/D = 0.18
Attempt to avoid use of ballast if possible.

Step 4: Calculate Lightship weight and centers using a 5% weight margin and a 0.1-m KG rise. From Figure 7, for house-forward, LCG/LWL = 0.50.

Item	Weight, m.t.	KG, m	LCG, m
Structure	70.1	2.90	
Outfit	44.6	3.96	
Machinery	13.2	1.92	
	127.9	3.17	12.0
Margin	6.4	+0.1	--
Total	134.3	3.27	12.0

Note that KG/D Lightship = 0.88.

Step 5: Make an initial estimate of ready-for-sea displacement based on the following additional requirements:

Design speed = 12 kn

Crew = 10

Endurance = 7 days, 5 @ full power, 2 @ half power

Crew and Stores - Estimate 10.0 metric tons

Fresh Water - Use 40 gallons/day/crew +10% = 11.7 metric tons

Fuel - As a first guess, assume 500 kw installed power at a fuel rate of 0.3 kg/kw-hr = 21.6 metric tons

Assume lube oil is negligible and no ice.

Ready-for-sea displacement, $\Delta = 177.6$ metric tons

Step 6: Re-estimate machinery weight using Figure 8. For 12 kn and 180 m. tons, obtain approximately 450 kw installed power. Keep the previous value of 500 kw and from Figure 9 obtain a machinery weight of 13 m. tons, so use the 13.2 value estimated earlier.

Step 7: Obtain C_B and C_p from Figure 11.

With L = 24.0 m, select $C_B = 0.48$ (below curve) and $C_p = 0.63$.

Then $T = \Delta/(L \times B \times C_B \times \text{specific gravity of sea water})$

$= 177.6/(24.0 \times 6.67 \times 0.48 \times 1.025)$

$= 2.25$ m

So that B/T = 2.96. This is within the spread of data shown on Figure 10.

Step 8: Estimate GM from Figure 12. Assume that this is a required GM value for L = 24.0 m, GM/B = 0.12, GM = 0.80 m.

Step 9: Obtain hydrostatic stability parameters.

Figure 13: With $C_B = 0.48$, KB/T = 0.63, so that KB = 1.42 m

Figure 14: With $C_p = 0.63$, $C_w = 0.80$

Figure 15: With $C_w = 0.80$, $C_i = 0.056$

Then $I_{\text{transverse}} = C_i \times L \times B^3 = 398.8 \text{ m}^4$

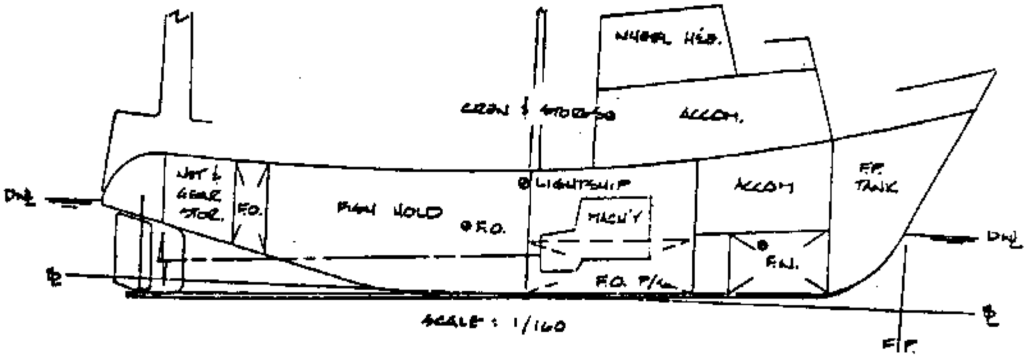
$BM = I_{\text{transverse}}/\nabla = 398.8/(177.6/1.025) = 2.30$ m

Step 10: From Figure 17, obtain bow height = 6.0 m (below curve).

From Figure 18, select drag of keel = 1.0 m

Freeboard = D - T = 3.70 - 2.25 = 1.45 m, which is high compared to Figure 19, but acceptable.

Step 11: Draw a small sketch to locate centers.



Step 12: Trim and Stability

Item	Weight, mt	KG, m	LCG, m
Lightship	134.3	3.27	12.0
Crew and Stores	10.0	5.40	9.1
Fuel Oil	21.6	2.0	13.2
Fresh Water	11.7	1.75	4.3
Ready for sea	177.6	3.13	11.5

GM = KB + BM - KG - free surface
= 1.42 + 2.30 - 3.13 - 0.01 (estimated)
= 0.58 m

This is below the 0.80 value from Step 8, so add ballast (See Step 3)

Ballast weight = 0.085 x 209 x 1.0163 = 17.8 metric tons
KG = 0.18 x 3.70 = 0.66 m
locate at LCG = 14.0

Ballast	17.8	0.66	14.0
Ready for sea	195.4	2.90	11.70

Assume propulsion power is still sufficient for the new displacement.

Recalculate GM with revised hydrostatics and weights.

Displacement = 195.4 metric tons
Assume C_B = 0.48 as previously obtained
Then

$T = 195.4/(24.0 \times 6.67 \times 0.48 \times 1.025) = 2.48$ m
$KB = 0.63 \times 2.48 = 1.56$ m
$BM = I_{transverse}/\text{new } \nabla = 398.8/(194.5/1.025) = 2.10$ m

GM = 1.56 + 2.10 - 2.90 - 0.01 = 0.75 m

Consider this adequate until intact stability studies can be conducted later in the design.

LCG = 11.70 m = 0.49 x L

Principal Characteristics

L = 24.0 m on waterline
B = 6.67 m
D = 3.70 m

Ready for sea Δ = 195.4 metric tons
T = 2.48 m C_B = 0.48 C_p = 0.63 Cw = 0.80

Installed power = 500 kw

WEIGHT AND Kg ESTIMATES FOR FERRO CEMENT FISHING VESSEL HULLS

by

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Abstract

This paper presents formulas for estimating weight and Kg/D for ferro cement fishing vessel hulls of 30 Loa 60 ft. These formulas were derived by extending published data and adopting simplifying assumptions. A check on the weight estimate indicates agreement within 10%.

1. Introduction

Ferro cement has attracted much attention as an alternate material to steel for constructing fishing boat hulls. In addition to the lower material costs which represent savings of 4 to 7%, a well built ferro cement hull has additional advantages including:

1.	Ruggedness:	Able to withstand service abrasion.
2.	Strength	Increased strength as cement cures.
3.	Corrosion Resistance:	No rust or corrosion protection required.
4.	Capacity:	In smaller vessels, a larger fish hold is possible due to less space taken up by scantlings and thermal insulation required. This advantage is lost as vessel size increases.

Ferro cement as a boat building material is like wood and fiberglass. It requires some skill and previous experience to obtain the desired results. The continued publication of books and reports /1/-/6/ has overcome this lack of information and promoted the use of ferro cement in fishing vessel construction.

In the present study an attempt was made to correlate published information to obtain formulas for estimating weight and Kg/D values for ferro cement fishing vessel hulls of $30 < \text{Loa} < 60$ ft. This represents an attempt to reduce the data to the format used in the Maritime Administration's "Fishing Vessel Design Data" report /7/. The final stage of using existing vessel data to confirm these formulas was not attempted. This was an unfortunate necessity in order to finish the study as a class project /8/ in NA 402, Small Commercial Vessel Design, taught by the second author at the University of Michigan.

2. Nomenclature

A_s : Hull surface area ft^2

B : Beam ft

C_h : Hull weight coefficient

C_I : Hull cubic index = (Loa x B x T)/100

C_n : Hull cubic number = (Loa x B x D)/100

C_s : Hull structure coefficient

D : Hull depth ft.

Kg : Vertical center of gravity of hull ft.

Kg_s : Vertical center of gravity of hull and structure ft.

Loa : Hull length overall ft

Lwl : Hull length on waterline ft

Sd : Ferro cement hull surface density lbs/ft^2

T : Hull draft ft

W_d : Weight of decks and brackets tons

W_h : Weight of hull (Table 1) tons

W_h^* : Weight of steel hull (Table 1) tons

W_s : Weight of hull and structure tons

W_{uw} : Weight of hull and upperworks tons

w : Weight of component tons

Z : Vertical center of gravity of component ft.

3. Basis for Study

During the 1960's a number of fishing vessels were constructed by the U. S. Fishing Fleet Improvement Act. The Maritime Administration's division of small ships created technical files for these vessels which were subsequently stored. In preparing fishing vessel designs for the National Oceanographic and Atmospheric Administration (NOAA) these files were brought out of storage and published as a series of graphs and design equations for preliminary vessel design /7/.

The present study stems from an effort to develop guidelines similar to those in /7/ for estimating the weight and Kg_s/D of ferro cement fishing vessels of length $30 < $ Loa < 60 ft. The format follows Fig. 1 from /7/ where

$$W_S = C_s C_n \text{ tons} \tag{1}$$

$$Kg_s = Kg_{s/D}\ D \tag{2}$$

Here the subscript "s" refers to structure which includes the fishing boat hull, house, welding, and fastenings (for wooden vessels).

With the possible combinations of a ferro cement hull, and upperworks constructed using wood, steel or ferro cement, it is difficult to obtain a reasonable estimate for weight and Kg/D for all the different cases.

Therefore the present study developed equations for estimating weight and Kg/D for ferro cement, steel and aluminum hulls of $30 <$ Loa $<$ 60 ft.

$$W_h = C_h Cn \tag{3}$$

$$KG_h = KG_{h/D}\ D \tag{4}$$

4. Fishing Vessel Hull Weight and Kg/D Estimates

4.1 Components Included in Hull Weight, Kg/D Estimates.

The components included in the hull weight and Kg/D estimates are summarized in Table I. The individual component weight is presented in terms of the hull weight W_h for ferro cement, steel and aluminum fishing vessel hulls based on a 100 ft fishing vessel designs /3/. The structural weight W_s is then assumed to be:

$$W_s = W_h + W_d + W_{uw} \tag{5}$$

where

W_h : Hull weight, broken down in Table 1.
W_d : Main deck, forecastle deck and bracket weight.
W_{uw} : House and upper works weight.

In /6/ W_{uw} is not included. For the present comparison, it is assumed:

$$W_{uw} = 0.10\ (W_h + W_d) \tag{6}$$

giving

$$W_s = 1.10\ (W_h + W_d) \tag{7}$$

Further analysis resulted in a ratio of W_h/W_s for ferro cement, steel, and aluminum hulls given in Table 1.

4.2 Kg/D Estimates.

The governing equation for obtaining the Kg/D of each hull is:

$$Kg/D = \frac{\sum (w/W_h)_i \times (z/D)_i}{\sum (w/W_h)_i} \tag{8}$$

Equation (8) is used to obtain Kg/D for the ferro cement, steel, and aluminum hulls in Table 1. The results are summarized in Table 2.

4.3 Development of Fishing Vessel Hull Weight W_h Estimates.

Equation (7) was used to develop the vessel hull weight W_h estimates. For the range of 30 to 60 ft hull length, it was assumed that the structure weight W_s of a ferro cement would be equal to the equivalent weight of a wooden hull. A hypothetical hull of Loa =52.5 ft., L_{WL} = 47.6 ft., B = 14.5 ft.,D=7.4 ft.,T=5.7 ft. was used as a basis. This hull has a cubic number C_n = 56.33 and a cubic index C_I = 43.39. Assuming the C_s equation for wooden hulls /7/, is valid for ferro cement hulls with C_n = 56.70, then from Fig. 1

$$C_s = \frac{C_n}{2.500} + 0.30 \tag{9}$$

giving

$$C_s = 0.322$$

From equation (1):

$$W_s = C_s C_n = 18.14 \text{ tons}$$

Using the results in Table 1 W_h/W_s = 0.54 for ferro cement hulls results in:

$$W_h = 0.54\ (18.14) = 9.8 \text{ tons.}$$

In this manner it was possible to obtain the coefficients for equation (3) which are summarized in Table 2.

5. Check of Ferro Cement Weight Estimates

It is possible to check the weight estimated using eq. (1) and the C_h coefficient for ferro cement hulls in Table 2. The hypothetical hull has C_I = 43.39 which corresponds to a surface area A_s = 1015 ft^2

Table 1. Hull Weight W_h and Kg/D Breakdown Based on /6/

Notes: $W_h^* = 55.26$ tons for Steel Hull

( ) : Kg/D Value

Hull Material	Ferro Cement	Steel	Aluminum
Component	w/W_h (Z/D)	w/W_h^* (Z/D)	w/W_h (Z/D)
Keel & sternbar	0.013 (.03)	0.018 (0.01)	0.023 (0.01)
Rudder & stern frame	0.030 (0.35)	0.037 (0.35)	0.040 (0.35)
Shell plating	0.58 (0.30)	0.555 (0.50)	0.586 (0.50)
Shaft tunnel	0.027 (0.20)	0.035 (0.20)	0.033 (0.22)
Bottom shell stiffness	0.015 (0.19)	0.014 (0.19)	0.011 (0.19)
Bulkheads	0.25 (0.67)	0.254 (0.60)	0.216 (0.60)
Engine seats	0.085 (0.15)	0.087 (0.15)	0.091 (0.15)
w/W_h (Kg/D)	1.00 (0.373D)	1.00 (0.465D)	1.00 (0.46D)
W_h/W_h^*	1.20	1.0	0.448
W_h/W_s	0.54	0.47	0.27

Table 2. Coefficients for C_h and Kg/D in Equations (3) and (4)

Hull Material	Ferro Cement	Steel	Aluminum
C_h	0.174	0.15	0.044
Kg/D	0.373	0.465	0.460

using curve "1" in Fig. 2 for normal hulls. Assuming a ferro cement hull will have a surface density sd = 19.45 lbs/ft^2, then

$$W_h = (sd)\ A_s \text{ lbs} \tag{10}$$

$$W_n = (19.45)\ (1015) = 19741.75 \text{ lbs} = 8.81 \text{ tons}$$

$$\text{Diff} = \frac{\text{Est. eq. (10)} - \text{Est eq. (1)}}{\text{Est. eq. (1)}} \times 100\% \tag{11}$$

Which gives a difference of - 10%.

6. Discussion and Recommendations

This paper has presented the results of a limited study on weight and Kg/D equations for ferro cement hulls in the range of $30 < \text{Loa} < 60$ ft. The calculation and assumptions introduced in obtaining the coefficients in Table 2 for equations (3) and (4) are summarized. The results of an independent check on the weight of the ferro cement indicates the hull weight estimates are within 10% of each other.

The preliminary Kg/D estimates indicate the hull Kg/D for ferro cement is lower than steel and aluminum hulls. This calculation should be refined and a more detailed analysis made to establish the actual Kg/D position for several fishing vessel designs.

If these initial Kg/D estimates are confirmed, then there is an additional advantage of larger GM to be claimed for ferro cement versus conventional hulls.

As an extension of this study it is recommended the final stage of using existing vessel data to confirm the coefficients in Table 2 be made.

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FIG.1 STRUCTURE WEIGHT COEFFICIET C_S AND $KG_{S/D}$ FOR U.S.FISHING VESSELS 1960-1970 /7/

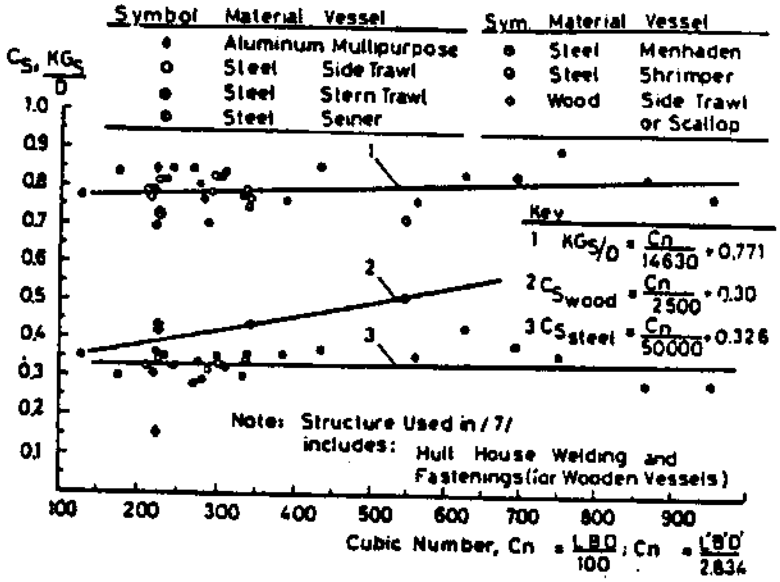
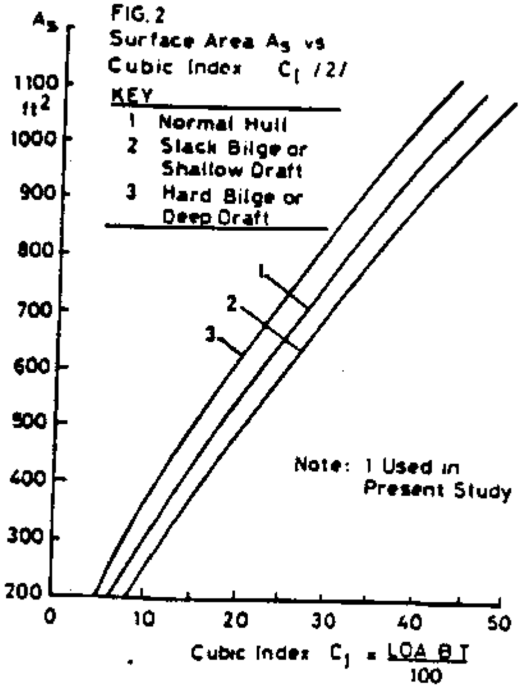


FIG. 2 Surface Area A_S vs Cubic Index C_I /2/



HYDRODYNAMIC DESIGN OF TUNA CLIPPERS

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Miguel Moreno
Manuel Carlier
Manuel O'Dogherty
C.E.H. de El Pardo

Abstract

The great increase experienced in the cost of marine fuels has a large influence on the service speed of many vessels, which tends to decrease. Nevertheless, in the case of Tuna Clippers, the high value of the catch brings about the need of maintaining a rather high service speed for a better economy of exploitation. Due to this design requirement, the hydrodynamic studies, contributing to a maximum fuel economy become of prior importance.

In this paper, the hydrodynamic aspects of the design of Tuna Clippers are analized, including:

- Computer aided choice of main Design parameters.
- Hull Lines Optimization.
- Appropriate choice of propelling system.

Two empirical Power/Speed Estimation methods are also presented. Both of them can be applied, with satisfactory results, to the tuna clippers. The first one is an early design method, being the second one a more accurate method, based on regressional analysis of the tests results of similar vessels, stored in the Data Bank of El Pardo Model Basin.

Introduction

The attention to be paid to the hydrodynamic aspects of the design of tuna clippers is of utmost importance taking into account the following considerations:

- These ships have to stay at sea during long periods of time, so that it becomes essential to assure that the ship has good seakeeping conditions, being able to sustain relatively high speeds in adverse weather conditions.

- The presence of holds with live bait and large free surfaces renders the stability of these vessels a point of utmost importance, that must be studied in great detail, in order to avoid service conditions that may endanger ship safety.

- The study of ship lines and the appropriate choice of the propulsion equipment and/or the reduction gear in order to decide favourable screw operational rpm's have a large incidence on fuel consumption and so on the economy of exploitation.

- The screw design may have a great bearing on the fuel consumption, so that it becomes essential to avoid undesirable cavitation phenomena which may develop into vibrations problems, leading to a loss in propeller efficiency and the production of vibration noise, unfavourable for the fishing activities.

For the design of ship lines, having tuna clippers relatively small dimensions as compared with those of merchant ships, preference must be paid to seakeeping in comparison with the behaviour in calm water, so that it is necessary to consider the prevailing weather conditions on the areas of ship operation.

Bearing this in mind the following general guidelines are of application:

- Moderate length/breadth ratios, generally in the range 4.5 to 5, are advisable due to stability requirements and seaworthiness.

- Extreme V sections should be avoided from the stability point of view, since they tend to increase the loss of stability in following seas. Besides very pronounced V stern frames may lead to shipping water on the stern deck, with a reduction on the ship safety.

- A good sheer forward, the adoption of satisfactory freeboard values and the existence of deck superstructures, forecastle, etc. improve the seaworthiness, specially with respect to shipping water on deck.

- The counterstern profile should have a slight rising slope in the aft direction, what is favourable for damping the pitching motions of the ship, contributing also to diminish the propeller suction.

- The shape of the sternpost is also important for obtaining sufficient hull-propeller clearances, in order to avoid excessive excitation to the hull and/or the propeller shaft.

- Using nabla-type bulbous bows is advisable for these ships since they allow an increase in service speed in fair and rough weather due to the reduction of pitching motions that they produce.

- The use of stern bulbs may be advisable, specially for those ships provided of high propelling power, in order to obtain a more uniform flow entrance to the screw and a lower vibration level transmitted by the propeller.

Tuna Clipper Ship Lines Design

As for any other ship type, the choice of ship lines is closely related to the most essential design characteristics such as :

- Ship main particulars
- Deadweight
- Required power for the design speed
- Stability and seaworthiness
- Manoeuvrability
- Ship cost

It can be stated that almost all of the characteristics of a tuna clipper are mutually dependent, in such a way that a modification of any design parameter will affect some other ship qualities requiring the reconsideration of other parameters. Therefore, the process to get an optimum design corresponds to an iterative scheme that leads the designer to a satisfactory compromise between the different design characteristics.

A flow-chart of the process for ship forms design in this type of vessel may be seen in figure 1. The investigation of all the possible alternatives in that diagram would require a forbidding work, if there is lack of sufficient information on ships similar to the design. On the other hand, that information may greatly simplify the design process as it may permit to restrict the scope of the study, under the following assumptions:

- Ship main dimensions choice, having in mind the design trends suggested by the service results of similar ships.

- Different aspects of the ship performance may be foreseen, having into account the knowledge of model and full scale results of those ships taken as the basis for the design.

From that point of view, a Model Basin will be in an optimum position to study the ship lines for a new design meeting satisfactorily all the design requirements.

Stability and Seakeeping

Ship Stability is a factor of primary importance, as it is intimately related to ship seaworthiness and safety. In Ref. 1, Paulling studied the stability conditions of American tuna clippers, including some capsizing problems. The Stability study of these vessels should take into consideration the following problems :

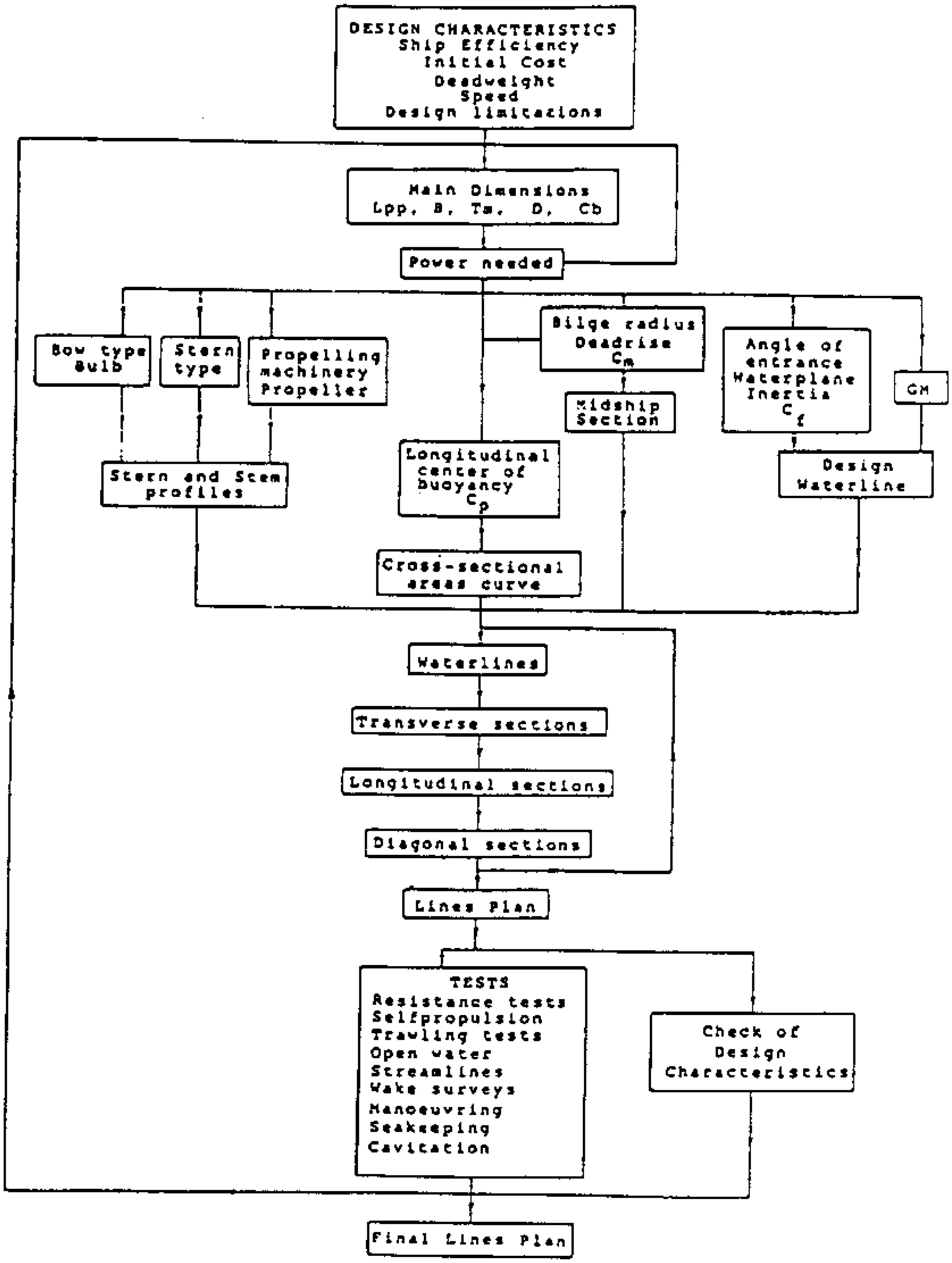

DESIGN CHARACTERISTICS
Ship Efficiency
Initial Cost
Deadweight
Speed
Design limitations
Main Dimensions
Lpp, B, Tm, D, Cb
Power needed
Bow type
Bulb
Stern type
Propelling machinery
Propeller
Bilge radius
Deadrise
Cm
Midship Section
Angle of entrance
Waterplane Inertia
Cf
GM
Stern and Stem profiles
Longitudinal center of buoyancy
Cp
Design Waterline
Cross-sectional areas curve
Waterlines
Transverse sections
Longitudinal sections
Diagonal sections
Lines Plan
TESTS
Resistance tests
Selfpropulsion
Trawling tests
Open water
Streamlines
Wake surveys
Manoeuvring
Seakeeping
Cavitation
Check of Design Characteristics
Final Lines Plan

FIG. 1

- Tuna clippers sail very often with large trims that produce righting arms that are different from those calculated from the cross curves deduced for the design trim.

- When the ship is operating at low speeds in following seas, the righting arms are largely modified by the waves action. That modification brings about a severe reduction when the ship is in the hogging condition, with a crest amidships and hollows forward and aft (fig.2). On this figure, three sections 1, 2 and 3, represent typical sections, forward, midships and aft. WoLo represents the waterline for the ship inclined to a large angle, in still water, while W_2L_2 is the local water level amidships and W_1L_1 represents the water level at the forward and aft sections. In comparison with the still water line of flotation, the ship has an additional buoyancy, E_2, amidships, while it experiences losses of buoyancy, E_1, E_3, forward and aft. The combined effect is a heeling moment which reduces the righting arms of the vessel.

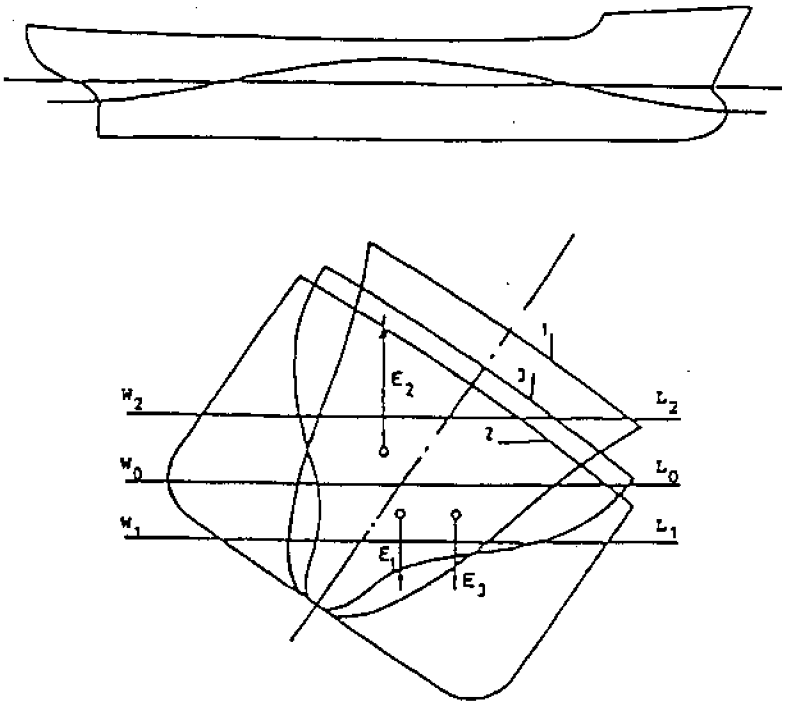


FIG. 2

- Great losses of stability due to large free surfaces may occur in some cases, as a consequence of the necessary holds with live bait.

In Ref. 2, a Stability Criterion for fishing boats was established. It has been checked by applying it to a large number of fishing boats, including both, vessels of satisfactory stability and sea behaviour and ships that have been lost at sea, through capsizing produced by defective stability. The Criterion has shown a good correlation, in order to establish a safe standard of stability.

In order to evaluate the seakeeping behaviour of fishing vessels, it is essential to investigate the rolling motions of the ship. The rolling period of any vessel may be expressed as:

$$Tb = f\ B/\sqrt{GM}$$

where f is a constant for any ship and loading condition, that takes values from about 0.73 to 0.83, if B and GM are expressed in meters. The values of f depend on the loading condition, the fullness of the ship and its weight distribution. In general, it may be assumed that f increases when the ship has empty spaces, permanent ballast, and/or large superstructures and top hamper. On the contrary, f tends to diminish when the ship is loaded, or if the ship lines are very fine, or the weights are concentrated around the rolling axis, or the ship has no superstructures or very little top weight.

Kempf established the rolling nondimensional constant, $Ck = Tb\sqrt{g/B}$, that relates the rolling period and the beam. Ck gives a measure of the accelerations produced by the rolling motions, so that small values of Ck, say less than 7, imply that the ship is very stiff, being submitted to violent rolling motions. On the other hand, large values of Ck, say more than 12, are associated to large rolling periods, with low initial stability of the ship, that may become tender and unsafe at sea, unless it is provided with ample freeboard and/or a large amount of watertight superstructures.

In order to avoid unsatisfactory rolling motions, what may occur either with slow motions, due to insufficient initial stability, dangerous for the ship safety, or with violent rolling motions, if GM is excessive, making very difficult the work on board, the values of Ck, in all operating conditions of the ship, should comply with the following criterion:

$$5.4 + 0.05\ LBP < Ck < 9.2 + 0.06\ LBP$$

Tuna Clipper Dimensions and Proportions

Table 1 shows main particulars of several tuna clippers tested at El Pardo Model Basin. A multiple linear regression analysis of those data furnishes linear relationships among the different variables to be used in the design.

Table 1.

LBP(m)	B(m)	T(m)	Cb	ServSp	EHP(cv)	Fnd
66.00	13.00	6.300	.6656	14.50	2042.	.608
47.00	11.70	5.400	.6550	12.36	1120.	.574
52.00	12.10	5.650	.6338	13.76	1453.	.624
41.00	10.70	5.000	.6303	12.25	849.	.603
63.60	13.00	5.800	.6178	14.78	2128.	.641
52.25	11.60	5.500	.6490	13.69	1451.	.625
63.00	12.95	5.750	.5642	15.30	1917.	.676
56.10	11.80	5.450	.5918	14.44	1459.	.661
56.10	11.90	4.750	.5773	16.19	1978.	.760
60.00	12.70	5.650	.6071	15.26	2151.	.675
54.00	13.10	5.210	.6023	15.53	2344.	.686
66.00	13.50	6.000	.5380	15.96	2558.	.695
61.00	12.50	5.400	.5895	15.28	1841.	.684
61.50	12.90	5.700	.5801	15.08	1912.	.667
63.00	13.00	5.700	.5818	15.22	1915.	.669
64.00	12.85	5.600	.5810	15.37	1923.	.678

The following empirical formulae have been obtained, by regressional analysis of the data that corresponds to the most satisfactory designs of Tuna clippers:

$$B = 0.1\ LBP + 6.4$$

$$LBP/B = 0.044\ LBP + 0.9\ Fnd + 1.6$$

$$Cb\ B/LBP = -0.00152\ LBP - 0.16\ Fnd + 0.324$$

$$Cm = 0.53\ Cb + 0.607$$

For the half-angle of entrance, at the design water line, though not included in the above tabulation, the following relation has been obtained in the same way:

$$Y = -0.25\ LBP + 165\ Cb\ B/LBP + 14$$

Considerations on the Ship lines of Tuna clippers

No easy and safe rules can be established for ensuring a satisfactory and near optimum design of ship lines for these fishing boats. Nevertheless, some notes may be given as a general guidance for preliminary design:

- The experience of El Pardo Model Basin, through the analysis of information stored in our Data Bank, shows a clear trend for an improved performance of vessels of this type provided with bulbous bows. Great care should be exercised in designing the bulb, making sure that the bulb is designed to be effective at the design draughts of ship operation, as a poor knowledge of design draughts and trims of the vessel may have a large influence on the ship performance, rendering the bulb effect to become negligible or even negative.

The analysis of the results obtained with the ships of this type for which we have information in our Data Base suggests the following design considerations:

- For the bulb protuberance:

$$0.020 < Xb/LBP < 0.022$$

- For the bulb height:

$$0.40 < Hb/Tpr < 0.44$$

- For the bulb sectional area at the forward perpendicular:

$$As20/As10 = 0.354 - 0.088\ LBP/B + 0.222\ Cb$$

- The value of the parameter Cb B/LBP has a marked influence on ship resistance, so that satisfactory results are obtained with moderate values of this parameter. From the studied ships, those with better hydrodynamic behaviour for the design speed range, showed values in the range 0.115 - 0.125.

- Prismatic coefficient optimum values are in accordance with the high service speed of these ships, for which a minimum wavemaking resistance in calm and rough seas must be seeked. Values in the range 0.62 - 0.64 are advisable. For Cp's greater than the higher limit, the longitudinal distribution of volume is impaired and the resistance increases. For values below the lower limit, at equal block coefficients, the midship section coefficient Cm must increase what damages the hydrodynamic behaviour of the hull.

- The longitudinal position of the center of buoyancy is of some importance from the points of view of resistance and seakeeping. In general its position must be aft, but with a good choice of lines its situation is not critical admiting a variation of 0.5 % of LBP with respect to the estimated optimum position. From the point of view of seakeeping, it should be slightly forward of the optimum position for propulsion in calm water. With these considerations in mind, 3 - 3.5 % of LBP aft are the suggested values for this parameter.

- In order to obtain a favourable propeller inmersion with large screw diameters, corresponding to low rpm, associated to larger propulsive coefficient, a design trim is advisable in these vessels. That trim may range from 1 - 3 % of LBP. It must be realized that, in any case, a moderate after trim is advantageous from a resistance point of view.

- The shape of the midship section is related to the fullness and form of the hull. The use of deadrise is advisable for achieving forms of low resistance. Its amount must be combined with the choice of bilge radius to attain a satisfactory curvature at the bilge, in order to diminish the generation of bilge vortices.

- The ratio LBP/B has a great effect on resistance. In some cases, the design values of Cb and/or Cp are obtained by adopting an excessive value of the beam, in combination with relatively fine entrance and run, what originates strong local changes of curvature that may produce many undesirable effects like eddy-making, wave-breaking, separation and shoulder waves.

- The longitudinal position of the maximum sectional area must be somehow abaft of midships, its position being a function of the weight distribution and the angle of entrance.

- The use of transom sterns is very convenient since it allows for more ample space aft for fishing operations, favours ship stability and diminishes the exit angles of the waterlines near the surface, reducing the risk of undesirable flow phenomena.

- Closed sternframes are generally used, what is convenient to protect the net in the fishing operations.

As an example figure 3 represents the body plan, sectional area curve and design waterline of a modern tuna clipper of satisfactory design.

Power/Speed Estimates

The prediction of power for a fishing vessel is particularly difficult having in mind the following:

- The small size of these vessels makes very difficult to obtain reliable information, referring to systematic Tank tests, carried out for fishing vessels.

- These ships are designed to operate at rather high values of Fn, as they have usually small length/beam ratios, what in time, gives rise to very high values of wavemaking resistance, that may be accompanied by unpredictable increases in viscous pressure resistance, due to eddy making, breaking waves and/or shoulder waves.

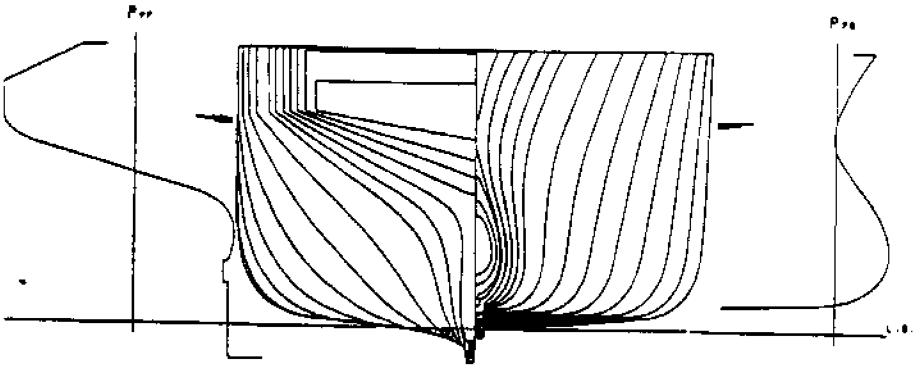
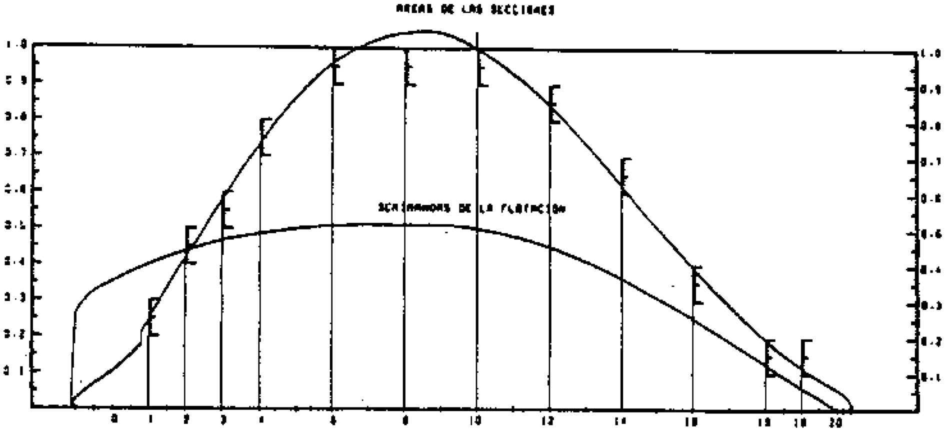


FIG. 3



- Due to the small size of many of these vessels, their service conditions are very influenced by environmental conditions, sea and wind, so that the seakeeping conditions of these vessels have also a large influence on their sustained sea speed, and their fishing ability.

- There are great differences in the technological design conditions of fishing boats of different countries, which depend not only on their variable technological levels, but also on the different fishing techniques and corresponding environmental conditions and economical criteria.

The information published about power estimates for trawlers is not applicable to tuna clippers due to the large differences in characteristics and dimensions. Similar comments may be stated when referring to methods for estimating the power of merchant ships, like those of Lap, Holtrop, Series 60, etc. As an example Table 2 includes normal values of CbB/LBP, Fnl and Fnd for the types of ships mentioned above.

Table 2.

	Merchant S.	Trawlers	Tuna Clipp.
CbB/LBP	0.10 - 0.14	0.11 - 0.17	0.11 - 0.14
Fnl	0.15 - 0.25	0.22 - 0.34	0.30 - 0.38
Fnd	0.35 - 0.65	0.45 - 0.70	0.55 - 0.70

Therefore, the best method to obtain accurate predictions and an optimum hydrodynamic design will be to carry out a Tank test program including, resistance, stream lines, wake survey, open water, selfpropulsion and cavitation tests.

Nevertheless, at the preliminary design stage, it is necessary to have an approximate prediction method available in order to obtain power predictions of reasonable accuracy.

Power/Speed Prediction By Means of a Data Base

This method, obviously, takes advantage of the knowledge of model tests and full scale results for a number of existing ships. In Ref. 4 a complete description of the procedure and Data Base can be found.

Fig. 4 represents the flow-chart that corresponds to this Power Prediction Method. In the course of this Prediction method, it becomes convenient to use empirical expressions corrected by means of the results obtained by its application to the basic-hull sample. What is actually done is to accept the general trend of the formula but correcting its "independent terms" in such a way that a good fit to the absolute values appearing in the analyzed basic-hulls is achieved.

This method is considered the most accurate one that may be used at the design stage, its typical error for tuna clippers being about 5%

Empirical Method of Power/Speed Prediction

From an analysis of the information on tuna clippers stored at the Data Base of El Pardo, the following approximate power prediction method has been derived.

The basis of the method is the knowledge of the ratio Crs/Cfs between the resistance coefficients defined as follows:

$$Cfs\ (ITTC\text{-}57) = 0.075/(\log Rn-2)^2$$

$$Crs = Crm = Ctm - Cfm$$

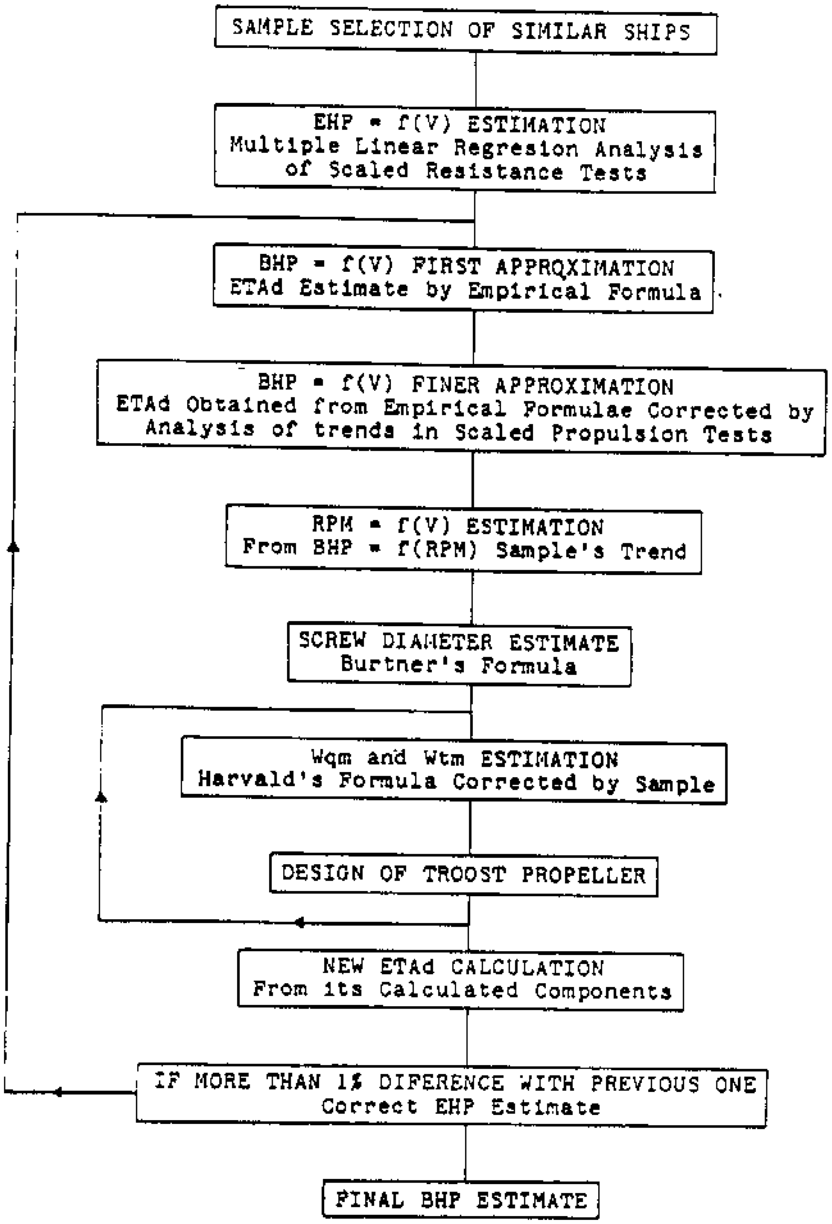


FIG. 4

The above ratio may be approximated by using a regressional formula given by:

$$Crs/Cfs = a\ Cb\ B/LBP + b\ B/Tm + c\ LBP + d$$

obtained from a linear multiple regresion analysis of the available data. The coefficients a, b, c and d have been determined for a standard Tuna clipper of 2,500 m3 displacement and for different values of Fnd. These values are shown on Table 3.

Table 3.

Fnd	a	b	c	d
0.550	3.606	-0.4505	-0.0188	2.719
0.575	8.438	-0.4411	-0.0160	2.002
0.600	19.523	-0.3331	-0.0047	-0.191
0.625	34.358	-0.3626	0.0132	-2.868
0.650	52.163	-0.4695	0.0400	-6.265
0.675	70.272	-0.7487	0.0757	-9.865

In order to obtain the power prediction , one must proceed as follows :

- Compute the value of Fnd for the design.

- Obtain from table 3 the values of the coefficients a, b, c and d corresponding to that Fnd.

- Calculate the ficticious length LBPf necessary to get a displacement of 2500 m3 with the proportions of the design.

- Compute Cfsf corresponding to the values Fnd and LBPf determined above.

- Compute Crs for the design by applying:

$$Crs = Cfsf\ (a\ Cb\ B/LBP + b\ B/Tm + c\ LBPf + d)$$

where all the values not otherwise stated correspond to the design.

- Compute Cfs for the design.

- Compute Cts = Cfs + Crs + Ca

- Compute Rts as:

$$Rts = 0.5\ \rho\ Scap\ V^2\ Cts$$

- Finally: EHP(cv) = Rts(kp) V(m/s) / 75.

By using the appropriate values of ETAm and ETAd the BHP value may be estimated.

Screw Design. General Considerations

In order to obtain a good propeller design, several considerations should be taken into account.

- As for any other ship type, the modification of the working conditions of the propelling system produced by the deterioration of hull, propeller and propulsion plant, should be accounted for by designing the screw to absorb a fraction of the nominal power of about 85% to 90% at the nominal rpm's, in trial conditions.

- The larger price of higher strength propeller materials is promptly paid back by the savings produced in fuel consumption and maintenance.

- Knowing the wake distribution when designing the propeller is of paramount importance for avoiding cavitation and vibration problems, obtaining a wake-adapted propeller of high efficiency. This requires to perform the corresponding wake survey tests.

Apart from the form of the distribution, it is necessary to know the absolute mean value of the effective wake Ws . As a first approximation, the following formula may be used.

$$Ws = 0.3\ Cb + 0.9\ B/LBP - 0.15$$

- The limitations imposed by the stern frame contour on the propeller design should be taken into account when designing the aft end of the ship, apart from the usual considerations on provision of spaces for fuel and ballast tanks or machinery elements.

Propeller Clearances

As it is known, providing scarce clearances between propeller and hull may produce serious vibration and cavitation problems.

Analyzing the minimum clearances required by the Classification Societies it can be seen that those requirements are lower the larger the number of blades, what is logical since torque and thrust fluctuations per blade decrease for increasing number of blades. On the other hand the optimum diameter for larger number of blades is lower, allowing the use of lower rpm at the propeller, what means an increase in efficiency. On the debit side, it must be taken into account that, generally, for larger number of blades efficiencies tend to decrease. Therefore an optimum must be found considering all the possible alternatives. Usually this optimum corresponds to 5 bladed propellers.

The Wake and its Influence on the Screw Working Conditions

The flow entrance conditions to the screw are decissive for the propeller-hull interaction, conditioning, on one side, the propeller efficiency, and on the other side, the possibility of cavitation phenomena, usually accompanied by vibrations and noises that may scare away the catch.

An estimation of the probability of appearance of this kind of phenomena may be obtained by the B.S.R.A. Criterion (Ref. 3). This Criterion makes clear the convenience of lowering the propeller rpm.

Propeller RPM Choice

From the considerations made above it may be stated that it should be chosen the minimum rpm value compatible with the following conditions:

- Possibility of using the optimum diameter, for which the maximum efficiency of the propeller can be attained.

- Provision of the necessary propeller-hull clearances, for which the requirements of Clasificaton Societies are usually a good guidance.

- Availability of the necessary reduction gear.

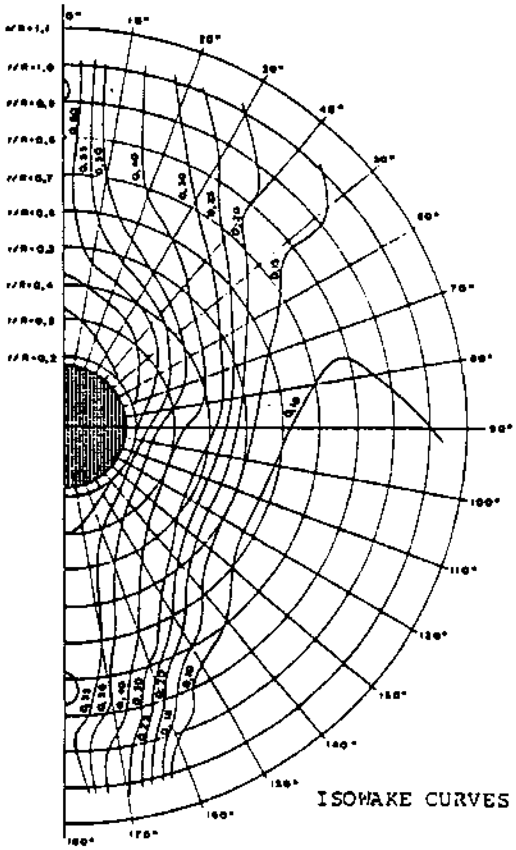


FIG. 5

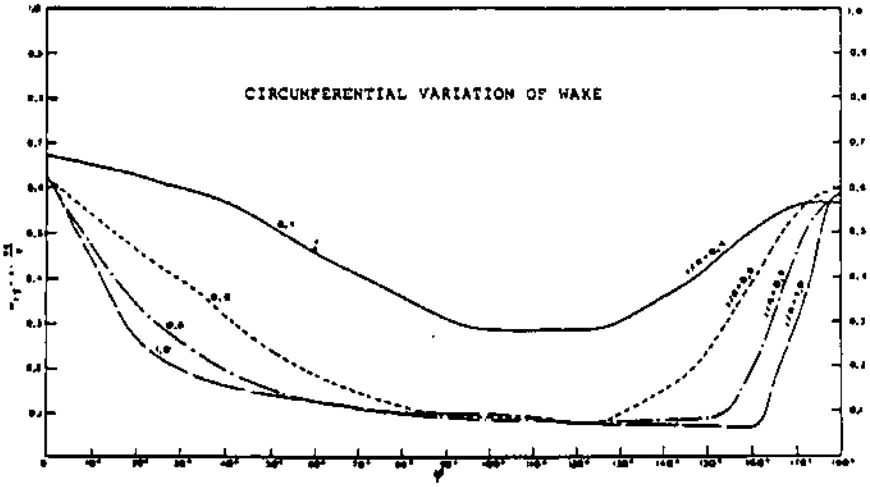


FIG. 6

- The propeller inmersion for all operating conditions must be satisfactory.

As an example of the above considerations, in Table 4 the characteristics of the propellers for a tuna clipper with 3700 BHP as a function of the chosen rpm are outlined. Figures 5 and 6 show isowake and circumferential variation of the wake for this ship.

Table 4.

Blade number	5	5	5	5	5
RPM	180	200	220	240	280
D (m)	3.35	3.22	3.10	2.98	2.65
Ae/Ao	0.60	0.62	0.65	0.68	0.77
V (knots)	0.922	0.873	0.826	0.786	0.804
ETAo	15.45	15.4	15.33	15.25	15.1
Margin	39%	23%	11%	0%	-7%

It can be seen that increasing the propeller rpm diminishes the service speed, the propeller efficiency and the B.S.R.A. Criterion safety margin. This last situation is represented on figure 7 where the cavitation number is given as a function of the parameter W_{Δ}, that gives an idea of the amplitude of variation of the wake values. For values in excess of 240 rpm, a high risk of cavitation and vibration problems can be foreseen.

Conclusion

The considerations outlined in this paper on the hydrodynamic design of tuna clippers reflect the present state of the art of the design technique for this type of ships in Spain, as it can be concluded from the information contained in El Pardo Data Base about ships tested in the last decade.

Though this information may be useful for the design of new tuna clippers, predictions made on the behaviour of ships of similar characteristics should be confirmed by the corresponding test programme that may allow, besides, the optimization of the design.

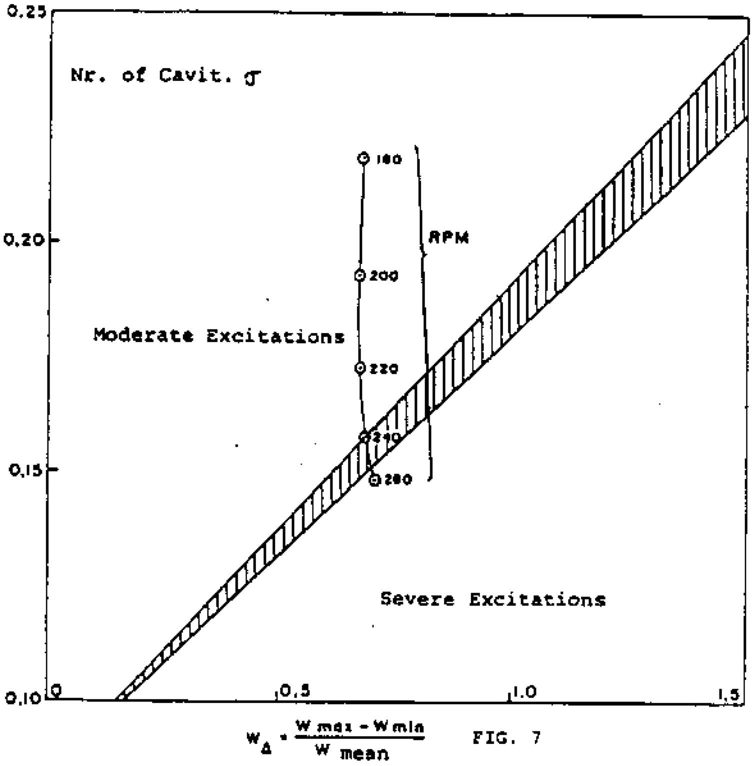


FIG. 7

Symbols

Ae/Ao - Blade Area Ratio
As10 - Maximum Cross Sectional Area
As20 - Cross Sectional Area at Fore Perpendicular
B - Moulded breadth
BHP - Propulsion Plant Power
Ca - Resistance Allowance Coefficient
Cb - Block Coefficient
Cfm - ITTC-57 Model Frictional Coefficient
Cfs - ITTC-57 Ship Frictional Coefficient
Ck - Kempf's Nondimensional Coefficient
Cm - Maximum Cross Sectional Area Coefficient

Cp - Prismatic Coefficient
Crm - Model Residuary Resistance Coefficient
Crs - Ship Residuary Resistance Coefficient
Ctm - Model Total Resistance Coefficient
Cts - Ship Total Resistance Coefficient
D - Screw Diameter
EHP - Effective Power
ETAd - Quasi-Propulsive Efficiency
ETAm - Line-Shaft Mechanical Efficiency
ETAr - Rotative-Relative Efficiency
ETAo - Propeller Efficiency
Fn - Length Froude Number
Fnd - Displacement Froude Number = V/ g
f - Constant for Rolling Period Calculation
g - Acceleration of Gravity
GM - Metacentric Height
Hb - Bulb Height
LBP - Length Between Perpendiculars
R - Screw Radius
r - Radius for a Section of the Screw
Rn - Reynolds Number
RPM - Speed of Rotation in rev/min
Rts - Ship Towing Resistance
Scap - Wetted Surface with Appendages
T - Draught
Tb - Rolling Period
Tm - Mean Draught
Tf - Draught at Fore Perpendicular
V - Ship Speed
Wqm - Model Wake Fraction at Equal Torque
Ws - Ship Effective Wake Fraction
Wtm - Model Wake Fraction at Equal Thrust
W_{Δ} - B.S.R.A. Criterion Parameter
Xb - Bulb Protuberance
γ - Design waterline Half-Angle of entrance
ρ - Fluid Density
σ - Cavitation Number

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ADDED RESISTANCE OF FISHING VESSELS IN HEAD SEAS

BY

STEPHEN R. JUDSON, P.E.*

ABSTRACT

Models of three typical New England fishing vessels were towed in calm water and in three different fully developed head seaways. Total drag was measured in each case which allowed the calculation of added resistance due to motion in waves. The same three designs were used in the M.I.T. Five-Degree-of-Freedom Ship-Motions Program that predicts added resistance. All of the speeds and seaways used in the model tank tests were duplicated in the ship-motions program, and so the accuracy of the added resistance prediction of the program was checked for this type of vessel.

The data from the towing tank test show that motion in waves may increase the drag of these vessels by more than 300%. In one case, the data indicate that the added resistance reaches a limiting value with increasing seaway severity. Comparison added-resistance calculation shows that the ship-motions program did not predict that quantity with any degree of accuracy or reliability for typical New England fishing vessels. The results are presented in a fashion thought to be most useful to designers who can use the results to estimate powering for this type of vessel.

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Opinions in this paper reflect those of the author and not necessarily those of the U.S. Coast Guard.

INTRODUCTION

One of the great unknowns in estimating powering is the added resistance of the vessel in waves. This added resistance can become a very significant part of the total resistance. Indeed, if the waves become large enough or if the vessel is small enough, the vessel is slowed greatly or even stopped. Obviously, an attempt to gather data for added resistance in waves for all types of vessels and wave conditions would be a project of incredible proportions. This paper describes an effort undertaken to quantify added resistance for a group of small vessels. The vessels considered were New England fishing trawlers. This group was selected for two reasons; there is very little resistance data of any kind available on these vessels and because of the author's proximity to these vessels.

Three models of these vessels were used in the M.I.T. Towing Tank to obtain resistance data in both calm water and waves. They were also used in the M.I.T. 5-Degree-of-Freedon Ship-Motions Program to check the accuracy of the computer predictions for added resistance for this type of vessel. The results of their effort showed that added resistance in waves may increase the total drag of these vessels by more than 300% over calm water resistance. It also showed that the M.I.T. Ship-Motions Program does not predict added resistance for New England fishing trawlers with any degree of accuracy or reliability.

SELECTION OF VESSELS

The author contacted John W. Gilbert Associates, Inc. in Boston for assistance in selecting appropriate vessels. The Gilbert design firm was chosen since it has had a great deal of experience in designing fishing vessels. Mr. Jerry Gilligan of Gilbert reconfirmed the concept of great similarity between typical New England trawlers and he provided representative vessels for this work.

The specific requirements for vessel selection were to include a range of size from largest to smallest vessel and to be as representative as possible of the vessels actually in use. These requirements were met by five vessels; four fishing trawlers and a research vessel. The research vessel was included for two reasons; first, it was basically a fishing vessel with its outfit changed for a different purpose and secondly, a towing tank model was already available for use. The lines, proportions, center of gravity and gyradii of the research vessel all were in the same range as the type of vessel being investigated.

At this point, it became necessary to decide which of the five vessels were to be towed in the model testing tank. There was neither time nor funds to do all five, so the largest, smallest and mid-sized vessels were chosen. The largest was a 139' research vessel which also had a model ready to use. The others were a 119' trawler and a 76' trawler. The author also decided to use the M.I.T. Five-Degree-of-Freedom Ships-Motions Program (SMP) to compare the results of it to the towing tank results. This had added potential benefit since if the program predicted the tank results accurately, then the two vessels not towed might also be used in the program to obtain data of interest. Particulars and body plans of the vessels used in the tank are included in Appendix I.

EXPERIMENTAL SET-UP

Models of the smallest and mid-sized vessel were constructed of fiberglass by Applied Model Technology in Needham, Massachusetts. The models were built as lightly as possible commensurate with maintaining rigidity. This ensured that during the ballasting procedure, the mass distribution of the model would accurately reflect the actual vessel. This was important since the mass distribution affects vessel motion, which in turn affects the resistance in waves.

The towing tank and wavemaking apparatus at the M.I.T. Ship Model Towing Tank were used without modification. During testing, the models were free to move in heave and pitch, but were restrained in roll, yaw, surge and sway. Model resistance was measured on a force block which incorporates a linear differential transformer. The force block was mounted on the models at the LCG and as close to the propeller-shaft line as possible. This location results in the best estimate of what shaft horsepower (SHP) will be required.

Waves were generated by a flat-plate wavemaker at one end of the tank which was hinged at the tank bottom and driven by a hydraulic ram. Due to the geometry of the tank, this allows only the case of head and following seas to be tested; however, the author feels that head seas is the most severe case. The waves in which the models were towed were synthesized Pierson-Moskowitz fully developed sea spectra. The spectra were created at M.I.T. using random noise as a source. Each spectrum is on magnetic tape with a real time sample of approximately 47 minutes. The model were run in each seaway long enough to exposes them to roughly 20% of the running time. This provided a significant sample for each wave condition.

The models were all ballasted to correspond to the maximum load condition and level trim. They were towed in calm water to obtain standard resistance in waves. Then each model was towed at three different speeds in each of three different sea states. In every case, the highest sea state was the maximum that the tank wavemaker could generate. The numerical value of the full scale significant wave height in each of these cases varied with the model, since the models were all built to different scales. The speeds were selected to provide a range that the fishing vessels would encounter in service.

Table I summarizes the test conditions. In addition to using models in the towing tank, the M.I.T. Five-Degree-of-Freedom Ship-Motions Program was also used to generate added resistance data. The SMP has many outputs, one of them being the calculation of added resistance due to motion in head seas. The program was utilized to predict values of added resistance for a wider range of seaways and speeds than were tested in the tank. The specific conditions were included and the author felt that if those results correlated well with the tank results, the wider range of seaways and speeds would provide more data for more complete estimates of added resistance.

TABLE I
TOWING TANK TEST CONDITIONS

Model 1 - 76' Trawler
Calm Water Drag 2-12 Kts Full Scale
Seaway Significant Wave Heigts: 1.92', 5.0', 8.16'
Speeds in Seaways: 3.04 Kts, 5.03 Kts, 7.05 Kts

Model 3 - 119' Trawler
Calm Water Drag 2-14 Kts Full Scale
Seaway Significant Wave Heights: 3.0', 7.0', 10.86'
Speeds in Seaways: 3.14 Kts, 5.05 Kts, 6.96 Kts

Model 5 - 139' Research Vessel
Calm Water Drag 2-12 Kts Full Scale
Seaway Significant Wave Heights: 2.9', 6.31', 10.2'
Speeds in Seaway: 2.99 Kts, 4.95 Kts, 7.21 Kts

DATA ANALYSIS

The drag raw data were recorded as a digitized frequency and converted to pounds via the force-block calibration constant. These drag forces were expanded up to full-scale vessel drag and effective horsepower (EHP) using standard model-test data-analysis procedures. The International Towing Tank Conference (ITTC) formulation for calculation of frictional resistance coefficient (C_f) and a correlation allowance (C_a) of .0004 were used in calculating full-scale drag for the vessel in 5º F seawater. A short program was written for a TI-59 programmable calculator to perform the calculations. Sufficient runs in calm water were made to make a smooth running plot of R /V versus V.

For the data in waves, enough runs were made at each speed to ensure that approximately 20% of the duration of each sea state tape was encountered by the model. During the higher speeds, this sometimes resulted in as many as 14 runs for one data point. For this portion of the analysis, in one sea state at one speeds, the drag forces form all runs were averaged to obtain the drag at that point. These drag forces were expanded in the same manner as calm water drag to obtain full scale drag and EHP. No corrections were used to account for changing wetted surface since there is no way of estimating the wetted surface at any given time. Added resistance was simply calculated by subtracting the value of calm water resistance off of the faired drag curve from the value of drag in waves at the same speed.

The output of the SMP is directly in added resistance for the full scale vessel at each sea state/speed point. This allows direct comparison of the model results and SMP results.

RESULTS

The results of the model tests are presnted in graphical form in Figures 1-6. These plots show drag versus speed and EHP versus speed for all three vessels for calm-water and wave tests.

As can be seen by the figures, there is a significant increase in drag and EHP in waves. This comes as no surprise to anyone who has ever been to sea. What is of particular interest is the amount of the increase. The tests indicate that for a given speed, the drag in waves may double or even triple compared to drag in calm water. The same is of course true for powering. Conversely, for a given power level, the speed may be reduced by so much as several knots. Again, this is no surprise, but it has not been quantified before for this type of vessel, to the author's knowledge.

Figures 7-9 show the values of added resistance obtained by model test and by SMP plotted against speed. This illustrates graphically that the SMP does not predict added resistance for these type vessels accurately or with any predictable error. It was for this reason that the SMP was not used to predict added resistance for the 2 vessels for which no model tests were done.

CONCLUSIONS

This series of tank tests dramatically quantifies the substantial added resistance fishing vessels encounter when operating in waves. It may be of use to designers in estimating required horsepower for propulsion. It should be noted again that the powering curves are in EHP, so a designer needs to apply a specific propulsive coefficient.

The range of differences between tank test values of added resistance and SMP values of added resistance clearly indicates that the SMP does not predict these values for these type of vessels well at all. The author feels that this is attributable to two main reasons. The first is that the vessels were well beyond the range of linear ship motion theory. There was large rigid body motion taking place, with green water being shipped over the bow very frequently. The second reason is that the SMP is more suited to much larger conventional hull vessels such as freighters and tankers which have much larger length-beam ratios which better satisfies strip-slender body theory.

It is interesting to note the.results of the tests on the 76' hull. The added resistance values for the significant wave height of 8.16' are less then those for the significant height of 5'. The author feels this is due to the fact that as waveheights get larger, so do wavelengths, and the vessel is riding easier. It is analogous to a lifeboat riding relatively comfortably in waves that batter large ships. This would indicate that there may be some peak value of added resistance for any vessel at a particular speed. Added reistance may increase with increasing waveheight up to this maximum point after which the added reistance may fall off. This seems reasonable since as the waves get larger and larger, the vessel only experiences smaller and smaller waves in its immediate vicinity even though it may be climbing the back of a mountainous wave.

It should be pointed out that there are several factors that have not been included in this analysis. The first is the added drag of a propeller when the vessel is being towed but the propeller is not turning or if it is freewheeling. The first case is worse than the second since the blades of a non-turning propeller will act as flat plates being dragged through the water. A reasonable estimate of this drag is 10-20% of the calm water resistance. In the higher sea states, it can be seen that value becomes insignificant compared with the total drag.

The second factor not considered was wind resistance. Accurate determination of wind resistance was beyond the scope of this work and would add yet another variable in the total resistance formulation.

In summary, the work done here was quantified what seamen and naval architects have known; that there is added resistance in waves. It has indicated that the values of added resistance are quite significant; as much as 300% or more of the calm water resistance. It provides designers of fishings vessels usable data for input to various decisions.

FIGURE 1

76' TRAWLER
DRAG vs SPEED

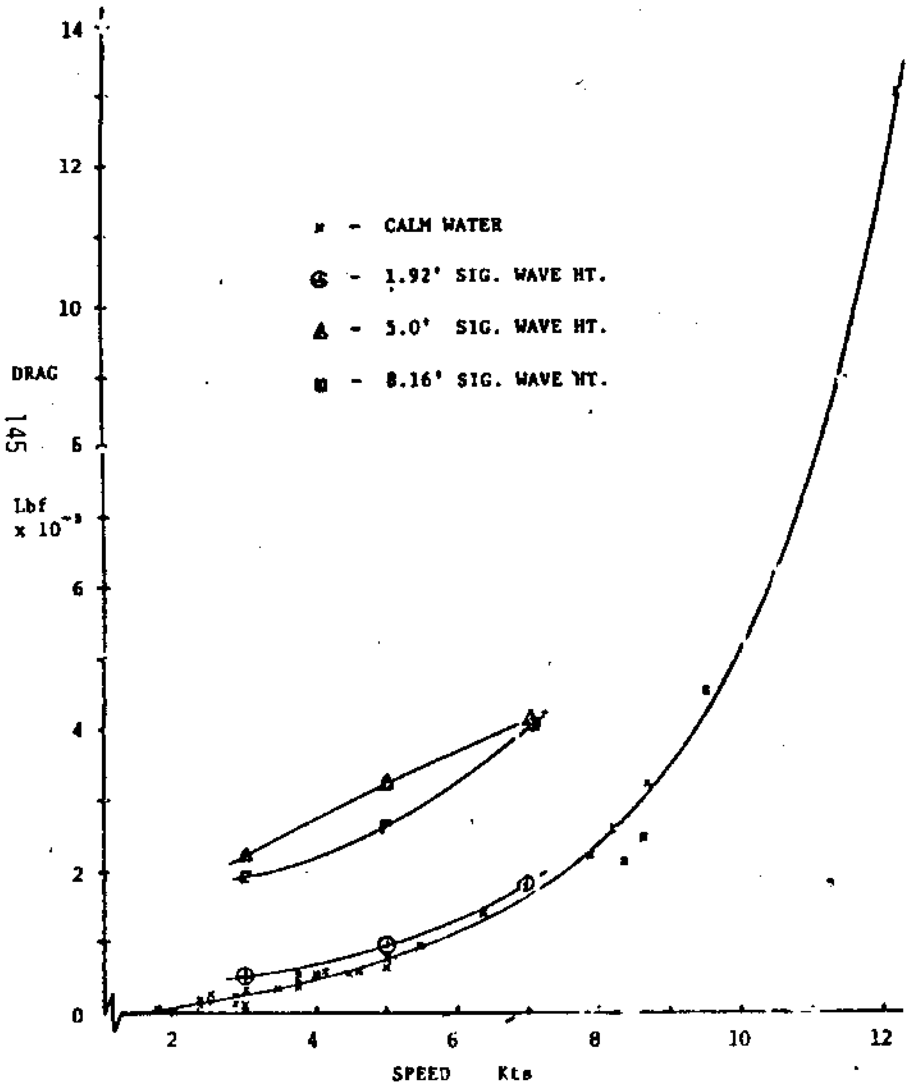


FIGURE 2

119' TRAWLER
DRAG vs SPEED

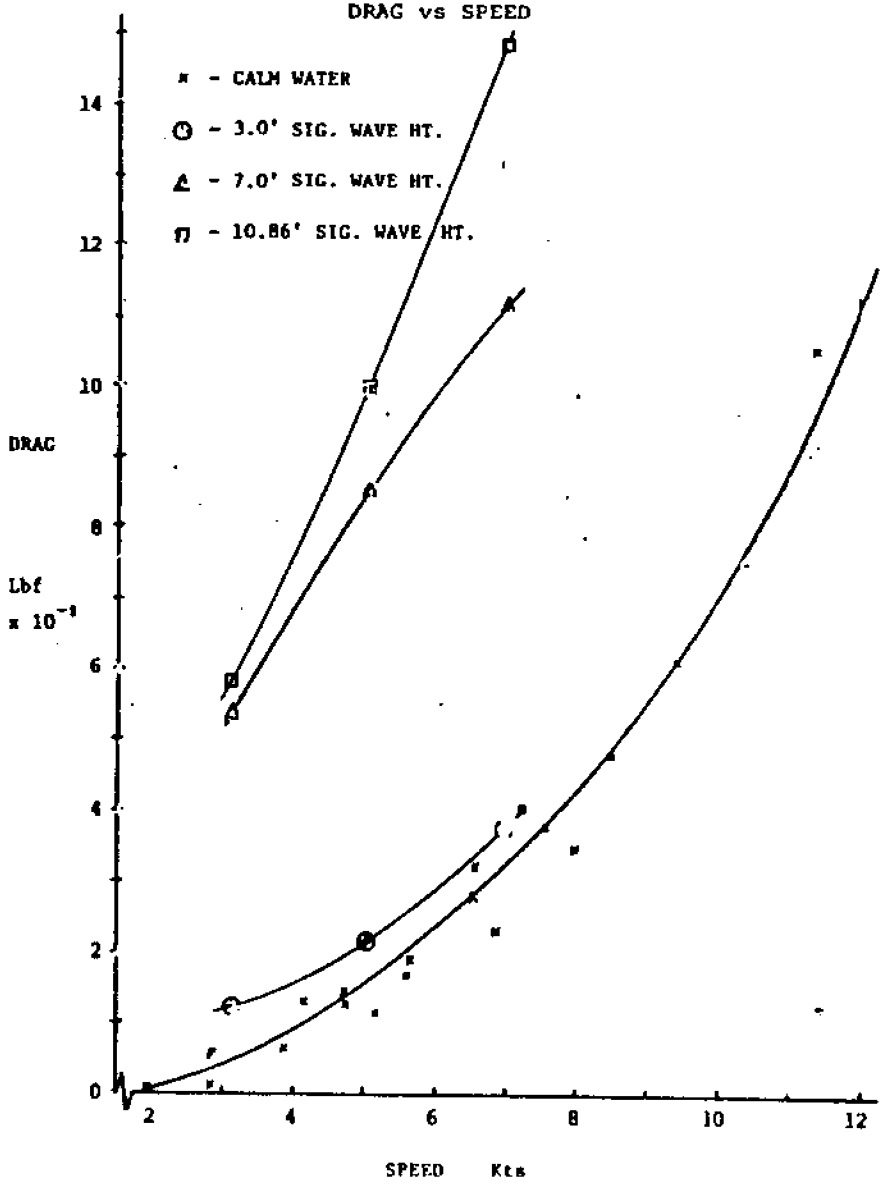


FIGURE 3

139' RESEARCH VESSEL
DRAG vs SPEED

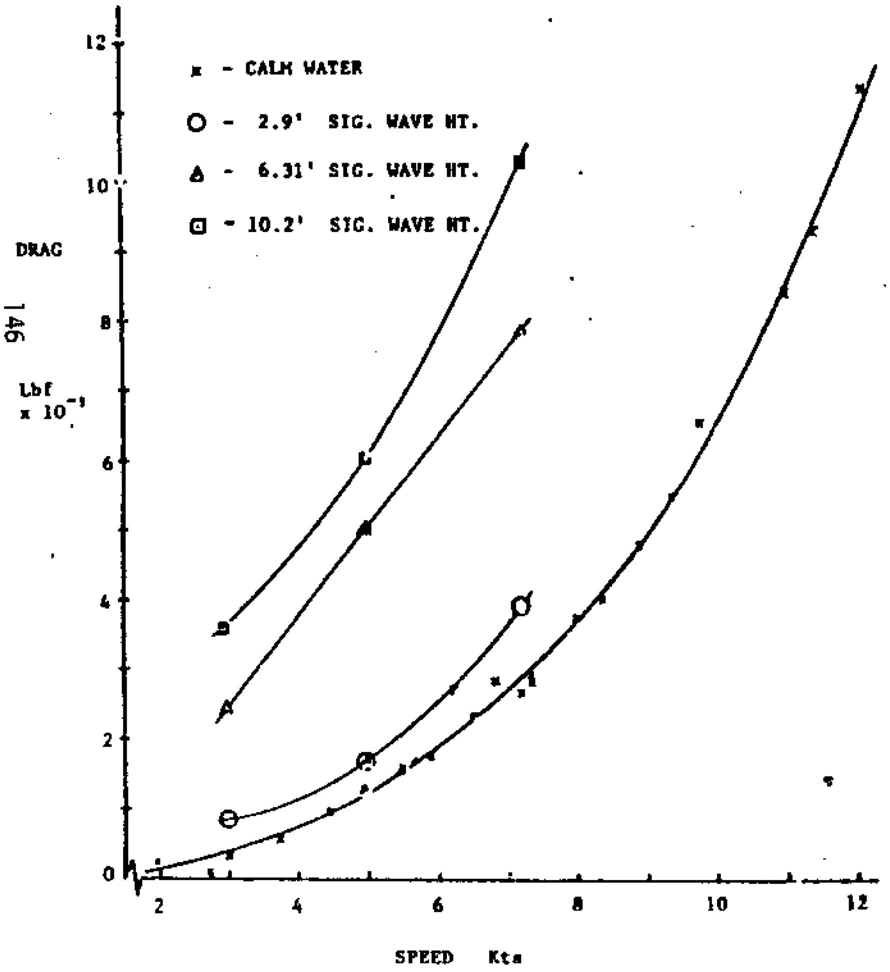


FIGURE 4

76' TRAWLER
EHP vs SPEED

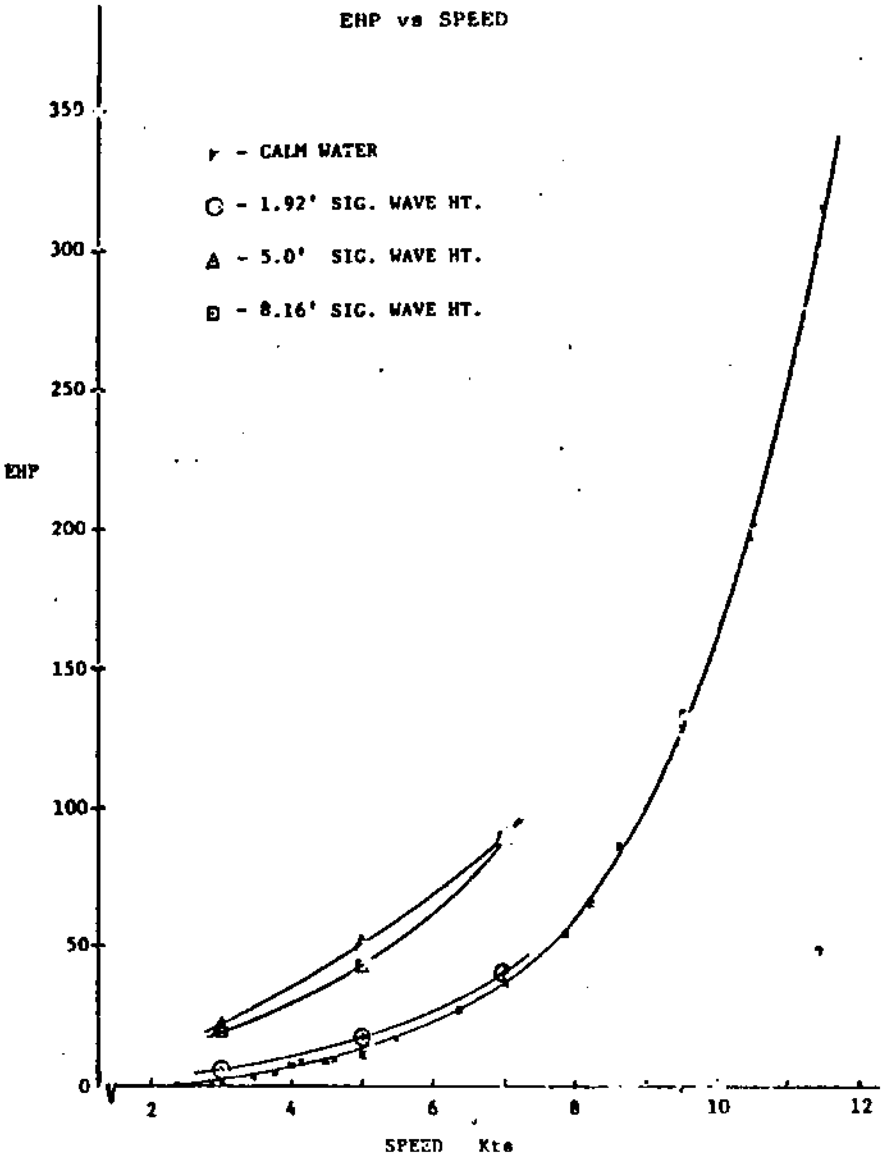


FIGURE.5

119' TRAWLER

EHP vs SPEED

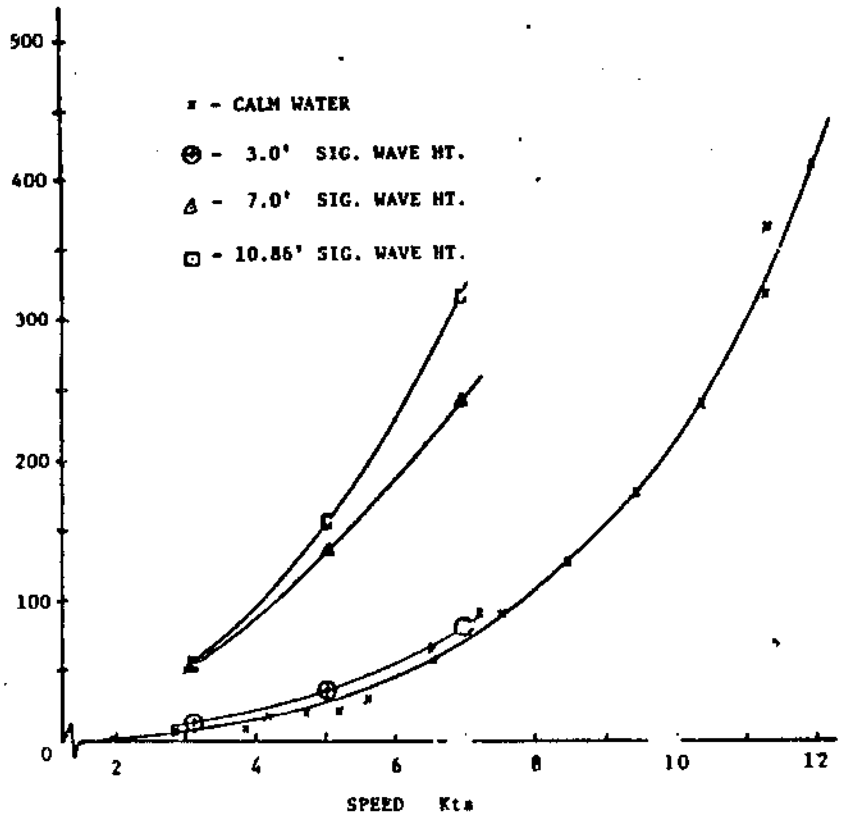


FIGURE 6

139' RESEARCH VESSEL

EHP vs SPEED

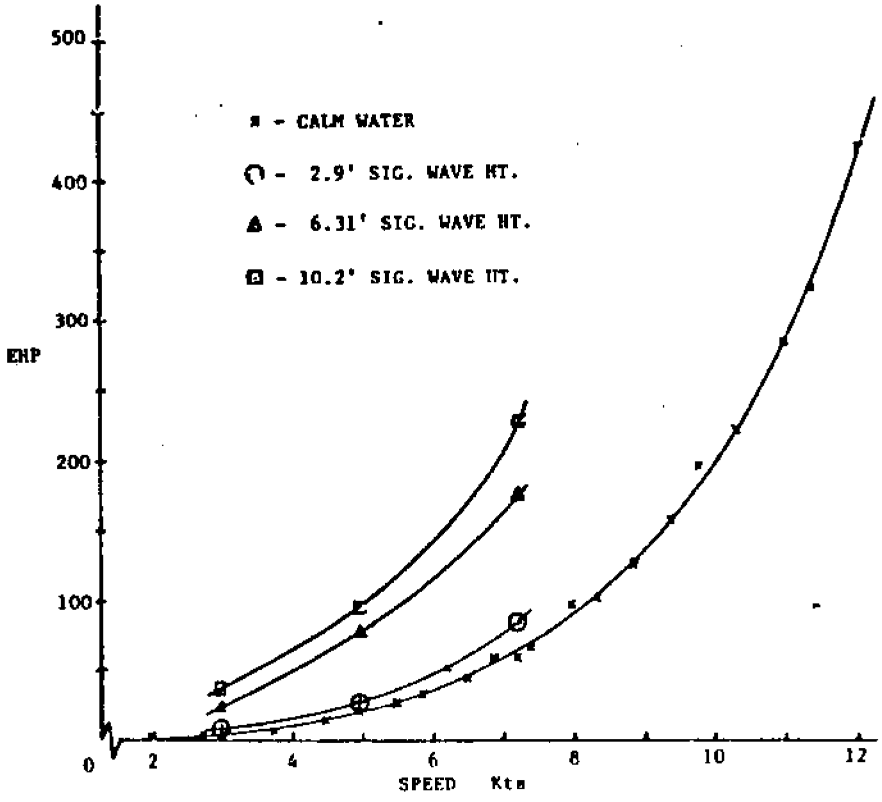


FIGURE 7

76' TRAWLER

ADDED RESISTANCE vs SPEED

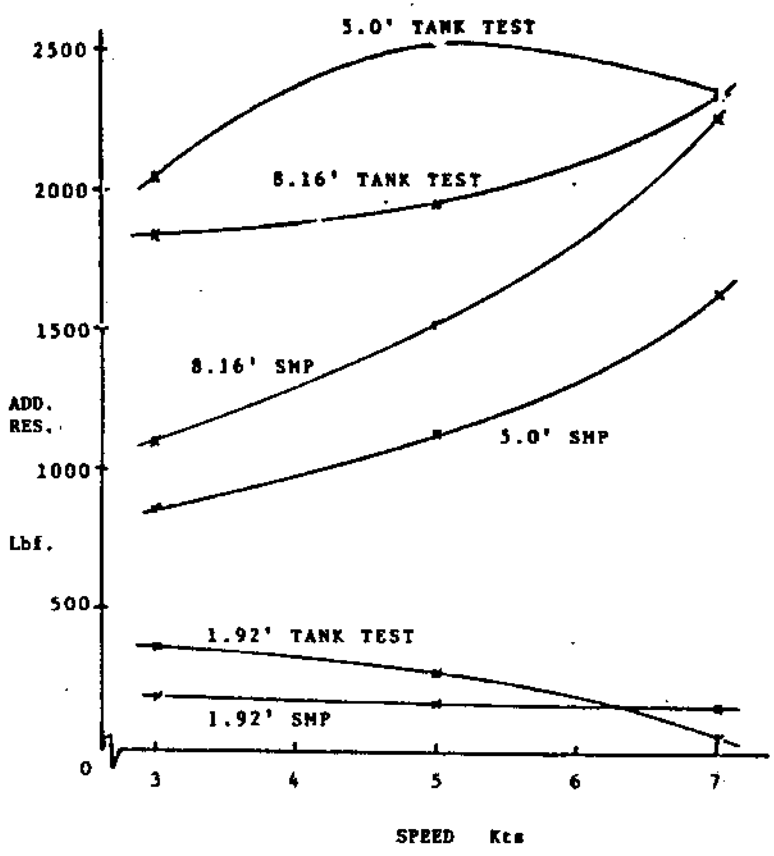


FIGURE 8

119' TRAWLER

ADDED RESISTANCE vs SPEED

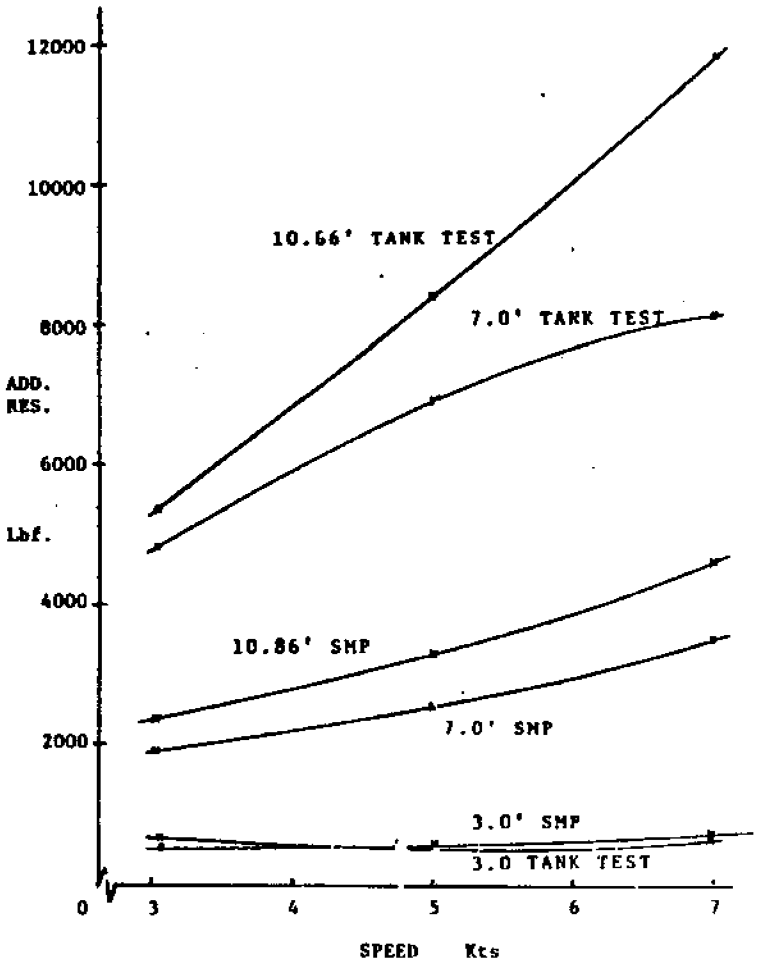
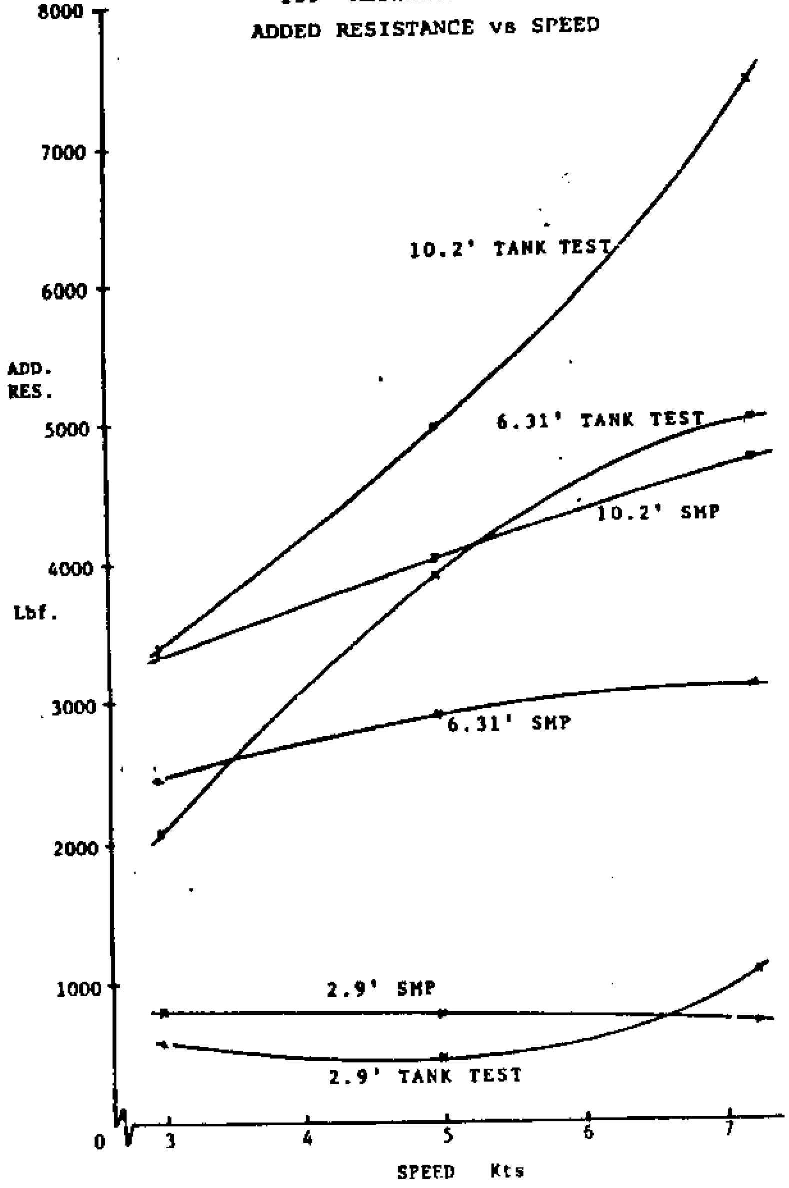


FIGURE 9

139' RESEARCH VESSEL
ADDED RESISTANCE vs SPEED



VESSEL DATA

Vessel No.1 - 76' Trawler

	Ship		Model
LOA	76' 7 1/2"		4' 6"
LWL	72' 0"		4' 2"
Beam	21' 9"		1' 3"
Draft	9' 0"		6 1/3"
Displacement	214.2 Tons		94.7 lbs.
Scale Ratio		17.03	

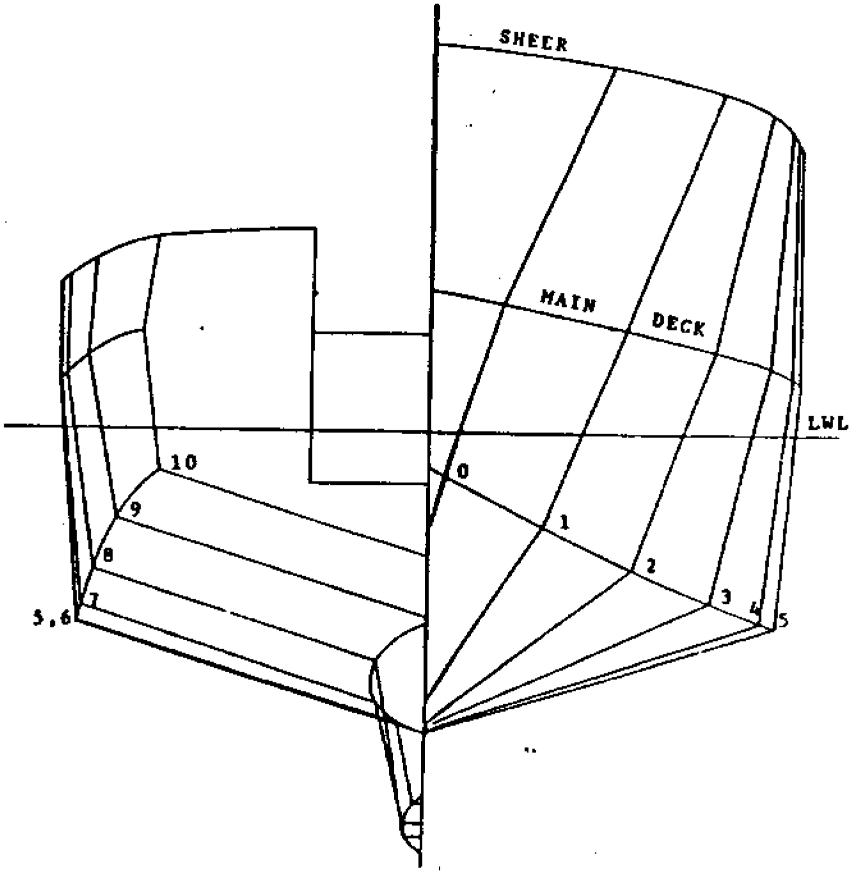
Vessel No. 3 - 119' Trawler

	Ship		Model
LOA	119' 4 3/4"		4' 6"
LWL	111' 6"		4' 2"
Beam	23' 0"		1' 1/2"
Draft	11' 10"		5 1/3"
Displacement	573.2 Tons		67.2 lbs.
Scale Ratio		26.53	

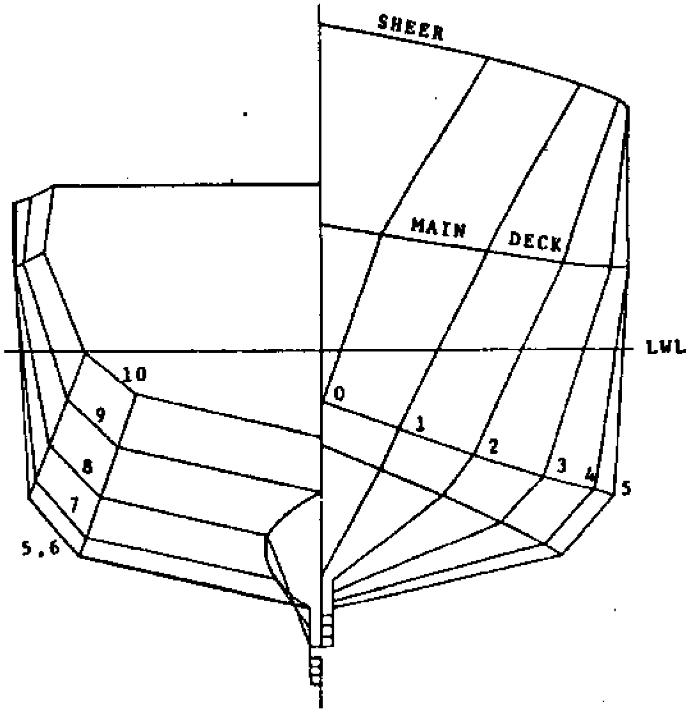
Vessel No. 5 - 130' Research Vessel

	Ship		Model
LOA	130' 4"		5' 9 3/4"
LWL	127' 9"		5' 4"
Beam	31' 4"		1' 4"
Draft	11' 2"		5 1/2"
Displacement	648.4 Tons		102.9 lbs.
Scale Ratio		23.97	

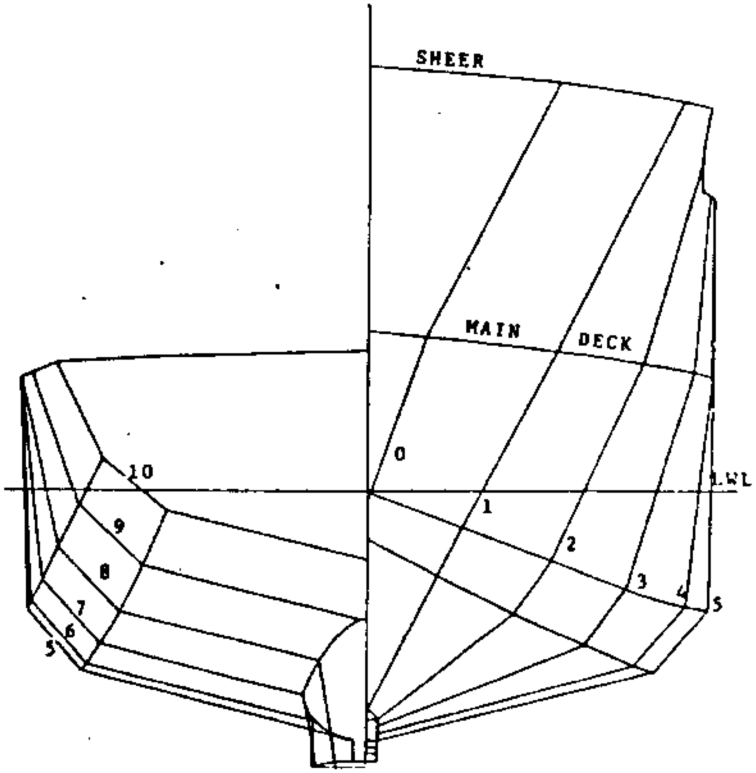
BODY PLAN OF
76' TRAWLER



BODY PLAN OF
119' TRAWLER



BODY PLAN OF
139' RESEARCH VESSEL



THE EFFECT OF VARIATIONS IN MAJOR HULL PARAMETERS OF FISHING VESSELS UPON DIRECTIONAL STABILITY

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Abstract

Directional stability is a term associated with the course-keeping ability and maneuverability of marine vehicles. In the design of large ships, directional stability and other maneuverability parameters are often evaluated using model tests or rather complex analytical methods. Unfortunately, the cost of such tests and analyses cannot normally be justified in the design of fishing boats or working vessels. Nevertheless, the need exists for the capability to make some assessment of the maneuvering characteristics of a fishing boat early in its design. The naval architect may be faced with requirements for high maneuverability for fishing operations or, conversely, for a large degree of directional stability to counteract situations where broaching might occur. This paper describes the development of a simple technique for determining whether a given fishing boat hull will be directionally stable, and for evaluating the effects of changes in major hull form parameters on directional stability.

Based on standard analytical methods for estimating hydrodynamic coefficients, a numerical model was written to compute the controls-fixed directional stability index specifically for working vessel hull forms. The numerical model was then run for variations in length-to-beam and beam-to-draft ratios, block coefficient, longitudinal center of gravity, and drag of keel. The results are plotted as "stability threshold" curves that can be easily used by the naval architect in the early stages of design to determine if a vessel configuration is directionally stable.

Introduction

Fishing boats and other working vessels frequently operate in services requiring greatly varying amounts of maneuverability and/or course-keeping ability. High maneuverability may be desired for fish-finding as well as fishing (seining, long-lining, trawling, etc.), while the ability to maintain course may be necessary for steaming to and from the fishing ground. The latter requirement is beneficial from the standpoint of minimizing strain on the helmsman and may be crucial for other reasons in severe sea conditions. In particular, course-keeping ability may substantially affect the resistance of a boat to broaching. Broaching may be defined as the loss of course-keeping ability and control when a vessel is travelling in a following sea at or near to the wave velocity. Since broaching is often accompanied by or can lead to a severe reduction in hydrostatic transverse stability, catastrophic consequences can result.

Course-keeping ability is dependent upon a number of factors [1] that include the magnitude and nature of the forces acting to disturb the vessel from its intended path; the speed and accuracy with which such disturbances can be detected and corrected through the helmsman, steering gear, and rudder; and the responses of the vessel to the disturbances independent of helmsman and rudder. The latter are normally assessed through a determination of controls-fixed directional stability characteristics. To the extent a vessel exhibits controls-fixed stability, its course-keeping ability will be enhanced. Instability is not inherently disastrous, but because more than one yaw angle (or ship's heading) can arise from a given rudder angle, the control problem is made more difficult.

In ship design, directional stability is frequently evaluated using model tests in sophisticated tank facilities or with complex analytical methods. Unfortunately, the time and expense associated with these tests and analyses cannot normally be justified during the design of a fishing boat. A need, therefore, exists for a means to quickly and inexpensively estimate the directional stability properties of fishing boats and other working vessels.

Nomenclature

B - ship beam, m or ft
C - controls-fixed stability criterion
Ca - correlation allowance
C_B - block coefficient, on waterline length
C_D - drag coefficient for Jacobs formula
Cf - frictional resistance coefficient
Cp - prismatic coefficient
Cr - residuary resistance coefficient
C_s - 2-dimensional added mass coefficient
$Ctot$ - total resistance coefficient
h - local draft, m or ft
h_f - height of deadwood at trailing edge of deadwood, m or ft
K - rotational added moment of inertia for a spheroid
K_1 - longitudinal added mass for a spheroid
K_2 - lateral added mass for a spheroid
L - length, on waterline, m or ft
LCB - longitudinal center of buoyancy from amidships, m or ft
m - mass of ship, kg or slugs
m_2 - added mass of hull in sway
m_z - rotational added mass of hull
N_r - derivative of yaw moment w/respect to rotational velocity
N_v - derivative of yaw moment w/respect to sway velocity
$Rtot$ - total resistance, N or lbs
S - wetted surface of hull, m^2 or ft^2
T - draft amidships, m or ft
t - time, s
V - ship speed, knots
v - ship speed, m/s or ft/s
$\bar{X}$ - centroid of added mass in sway from amidships, m or ft
X_f - centroid of skeg lateral area from amidships, m or ft

X_g - center of gravity from amidships, m or ft
Xo - Cp/L
Xp - center of lateral effort of hull profile from amidships, m or ft
Y_r - derivative of sway force w/respect to rotational velocity
Y_v - derivative of sway force w/respect to sway velocity
σ - stability index
ρ - density of water, kg/m^3, or sl/ft^3
∇ - volume of displacement of hull, m^3 or ft^3

Superscripts and Subscripts

' - prime indicates nondimensionalization
f - as in $(N_v)_f$ indicates the skeg hydrodynamic coefficient
h - as in $(N_v)_h$ indicates the bare hull hydrodynamic coefficient
. - indicates differential with respect to time (as in $\dot v$ and $N_{\dot v}$)

Equations of Motion and the Stability Criterion

The nondimensional equations of motion for a ship in the horizontal plane and neglecting surge are [1]:

$$-Y'_v v' + (m' - Y'_{\dot v})\dot v' - (Y'_r - m')r' - (Y'_{\dot r} - m'X'_g)\dot r' = 0 \qquad \text{(Sway)}$$

$$-N'_v v' - (N'_{\dot v} - m'X'_g)\dot v' - (N'_r - m'X'_g)r' + (I'_z - N'_{\dot r})\dot r' = 0 \qquad \text{(Yaw)}$$

In accordance with the presentation in [1], the solutions to the above simultaneous differential equations can be assumed to take the following form:

$$v' = v_2 e^{\sigma_1 t} + v_3 e^{\sigma_2 t}$$

$$r' = r_2 e^{\sigma_1 t} + r_3 e^{\sigma_2 t}$$

For controls-fixed stability, the sway velocity (v') and yaw angular velocity (r') must decrease with time, which necessitates that both σ_1 and σ_2 be negative. If the solutions for v' and r' are inserted back into the equations of motion, a quadratic equation in terms of σ results:

$$A\sigma^2 + B\sigma + C = 0$$

Solving,

$$\sigma_{1,2} = \frac{-B/A \pm \sqrt{(B/A)^2 - 4C/A}}{2}$$

For ships, A and B are always large and positive, so that C must be positive in order to obtain negative values for σ_1 and σ_2. Thus, the criterion for controls-fixed stability can be shown to be

$$C = Y'_v(N'_r - m'X'_g) - N'_v(Y'_r - m') > 0 \quad .$$

Determination of Hydrodynamic Coefficients

The approach outlined in [1] was employed to predict the hydrodynamic coefficients (or derivatives) for use in the above stability criterion equation. Essentially, this approach assumes the bare hull of a ship to be a low aspect ratio foil, where the aspect ratio is defined as draft/length, operating at a Froude number of 0.25 or less. The Froude number restriction is imposed in order that the influence of wave making can be neglected. Therefore, the hull can be considered to be the lower half of a double body mirrored about the free surface horizontal plane. The velocity-dependent bare hull hydrodynamic coefficients can then be estimated with the following equations:

BARE HULL VELOCITY-DEPENDENT SWAY FORCE

$$(Y'_v)_h = \frac{(Y_v)_h}{\frac{\rho}{2} LTv} = -\frac{\pi T}{L} + (C_D)_h$$

The first term, $\pi T/L$, is the slope of the lift coefficient versus angle of attack curve based on Jones' formula for low aspect ratio foils, and is generally applicable only where the hull's effective aspect ratio, $2T/L$, is less than 1/5. Most working vessels satisfy this criterion, although not by a significant amount. The range of effective aspect ratio investigated in this report is 1/3 to 1/10.8. Jones' formula is used for the higher aspect ratios anyway, since from Figure 41 on page 501 of [1], it does not appear to be greatly in error until the effective aspect ratio exceeds 1/2.

The second term, $(C_D)_h$, is the drag of the vessel at zero angle of attack, or the total calm water resistance. Using the Webb

Standard Trawler Series resistance data [2], $(C_D)_h$ was determined using ITTC friction and a correlation allowance, Ca of 0.0004. The residuary resistance coefficient was approximated for $V/\sqrt{L} = 1.2$ by the following exponential linear fit:

$$Cr = 0.0427\ Cp^{4.343}$$

This relationship was derived from Figure 1 which is taken from [2]. A displacement-length ratio of 467 was assumed in order to develop the equation for Cr. Note that the effect of displacement-length ratio upon Cr is small compared to the effect of Cp. Since the total term $(C_D)_h$ is not large, neglecting the effect of the displacement-length ratio will not introduce significant error into the final results.

Having obtained Ctot = Cr + Cf + Ca, it is necessary to convert it to $(C_D)_h$, which is nondimensionalized by different parameters. Note that:

$$Ctot = \frac{Rtot}{\frac{\rho}{2} Sv^2} \qquad \text{but} \qquad (C_D)_h = \frac{Rtot}{\frac{\rho}{2} LTv^2}$$

For the Webb Series, Nevitt used the Taylor wetted surface coefficient, $C = S/\sqrt{\nabla L}$, where C is a constant varying from 2.65 to 2.75. Using C = 2.70 as a mean, the conversion can be made as follows:

$$(C_D)_h = Ctot \times 2.70\sqrt{BC_B/T}$$

As a final point, the high speed-length ratio of 1.2 was selected as being typical of a trawler or other fishing vessel in the free-running condition. This violates the earlier restriction on wave making since it corresponds to a Froude number of approximately 0.36. However, the high speed-length ratio affects only the $(C_D)_h$ term, which is very small. Therefore, the hydrodynamic coefficients for the free-running condition or for lower speeds that do satisfy the wave-making requirement will not be substantially in error.

BARE HULL VELOCITY-DEPENDENT YAW MOMENT

$$(N'_v)_h = \frac{(N_v)_h}{\frac{\rho}{2} L^2 Tv} = -\ (m'_2 - k_1 m') + \frac{Xp}{L}\ (Y'_v)_h$$

The first term is the velocity-dependent moment of the hull in an ideal fluid in which

$$m_2' = \frac{\text{added mass of hull in sway}}{\frac{\rho}{2} L^2 T} = \frac{K_2}{\frac{\rho}{2} L^2 T} \int_0^L \frac{\rho}{2} \pi C_s (h)^2 dx$$

k_2 = lateral added mass coefficient for a spheroid in infinite flow

h = local hull draft

C_s = the 2-dimensional sectional added mass coefficient, equal to 1.00 for an ellipse

and

$k_1 m'$ = added mass of the ship in surge (nondimensional)

$$m' = \frac{\text{mass of ship}}{\frac{\rho}{2} L^2 T}$$

k_1 = longitudinal added mass coefficient for a spheroid in infinite flow.

In order to simplify and generalize the expression for m_2', the hull form is assumed to consist only of elliptical sections. This allows a constant value of Cs to be used and also enables major hull form variations to be made without requiring a special means to generate sectional half-breadths for each variation. The elliptical section yields a midship section coefficient of 0.785, which is typical for working vessel types. Given a reasonable sectional area curve and underwater hull profile, an equally reasonable design waterline results.

It should be noted that the inclusion of L/T in the integrand of the last formula on page 521 of [1] is apparently in error.

The second term in the yaw-moment equation is due to the formation of vortices on the downstream sides of both bow and stern as a result of viscous flow. These vortices can be associated with a reduction in the pressure distribution over the hull from that of the ideal flow case. Xp is taken as the distance from amidships to the center of lateral effort of the hull.

BARE HULL ROTATIONAL VELOCITY-DEPENDENT SWAY FORCE

$$(Y_r')_h = \frac{(Y_r)_h}{\frac{\rho}{2} L^2 T v} = -K_1 m' + \frac{Xp}{L} (Y_v')_h$$

Here, the first term is the outward force exerted by an ideal fluid on the hull undergoing a constant rotational velocity, and the second term again accounts for viscous effects.

BARE HULL ROTATIONAL VELOCITY-DEPENDENT YAW MOMENT

$$(N'_r)_h = \frac{(N_r)_h}{\frac{\rho}{2} L^3 T v} = - m'_z \frac{\overline{X}}{L} + \left(\frac{Xo}{L}\right)^2 (Y'_v)_h$$

Here

m'_z = the rotational added mass coefficient

= $(K'/K_2)m'_2$

where

k' = the rotational added moment of inertia for a spheroid in infinite flow

$\overline{X}$ = the centroid of added mass in sway from amidships

$\frac{Xo}{L}$ = Cp/2.

SKEG HYDRODYNAMIC COEFFICIENTS

Using Jones' formula and the method of [1]:

$$(Y'_v)_f = - \frac{\pi (h_f)^2}{LT}$$

where h_f = height of skeg at trailing edge of skeg

$$(N'_v)_f = X_f \cdot (Y'_v)_f$$

where X_f = distance from center of lateral effort of skeg to amidships

$$(N'_r)_f = X_f^2 \cdot (Y'_v)_f$$

$$(Y'_r)_f = (N'_v)_f$$

Finally, the total hydrodynamic coefficients are (hull plus skeg):

$$Y'_v = (Y'_v)_h + (Y'_v)_f$$

$$N'_v = (N'_v)_h + (N'_v)_f$$

$$Y'_r = (Y'_r)_h + (Y'_r)_f$$

$$N'_r = (N'_r)_h + (N'_r)_f$$

Development of Directional Stability Threshold Curves

A numerical model was written to compute the hydrodynamic coefficients and the controls-fixed stability index. The model was used to run a large number of hull form variations as a means to develop a simple graphical method for determining directional stability.

A typical working vessel will have a length/beam ratio of around 4.0, a beam/draft ratio of 2.5, drag of keel equal to 0.03 times the length, and a block coefficient of 0.50. In order to investigate the effect of changes in these parameters and also LCB and drag of keel, a matrix of hull form variations was generated. Based on the Webb Trawler Series hull W-18 from [2], three separate hull and skeg profiles were constructed corresponding to zero drag of keel, drag equal to 0.03L, and drag equal to 0.06L. Sketches of these profiles and tabulated values of the local hull and skeg draft nondimensionalized by draft amidships are shown in Figure 2.

For each drag of keel, nine different sectional area curves were drawn, as shown in Figure 3. For these curves, C_B was varied from 0.40 to 0.60 by 0.10 increments. Then, for each C_B, three LCBs (0.05L aft of amidships, amidships, and 0.05L forward of amidships) were developed. Trial and error were used to obtain sectional area curves with the proper C_B and LCB.

The numerical model includes an option to produce automatic variations in L/B and B/T from 3 to 6 by increments of 1, and from 2.0 to 3.6 by increments of 0.40, respectively. With the three hull profile variations, nine LCB-C_B variations, and 20 L/B-B/T variations, a total of 540 different hull forms were investigated. These 540 hulls were divided into 27 ship sets for cataloguing purposes, with each ship set having only one drag of keel, LCB, and C_B, but 20 different L/B and B/T ratios.

Throughout, a standard hull length of 100 feet was chosen as being typical of a mid-sized fishing boat. The initial run of each ship set was arbitrarily selected at L = 100, B = 40, and T = 20. This is a particularly fat hull of unusual dimensions used only to initialize each ship set, and the resulting stability output was ignored.

Plots of stability criterion, C, versus B/T for the four L/B ratios were developed from the numerical model output for each ship set. An example of these plots is given in Figure 4 for zero drag of keel, LCB at amidships, and $C_B = 0.50$. The general trend in these graphs is for the low L/B hull to be stable at higher B/T values. This is primarily due to the fact that B/T and L/B are not independent, and therefore, what actually is occurring is a strong dependence of the stability criterion upon T/L. From these graphs, the values of B/T at C = 0 were read and then replotted in Figures 5, 6, and 7.

Figures 5, 6, and 7 are plots of C = 0 lines, or "stability thresholds" against L/B and B/T. Each figure is for a single block coefficient, but includes the three drag of keel and three LCB variations. The concept of the "stability threshold" graphs is to allow the design naval architect to enter with major hull form parameters and quickly determine if the vessel in question will be directionally stable. Any point lying to the left of a threshold line indicates stability, any point to the right, instability. Stability characteristics for intermediate values of C_B, drag of keel, or LCB can be approximated by interpolation.

The behavior of the threshold lines is reasonable. Low block coefficient, additional drag of keel, and LCB forward all increase the "stable region." The biggest improvement in stability occurs for increases in drag of keel--the effects of LCB changes and C_B changes are about equal. In fact, a drag of keel of 0.06L appears to guarantee directional stability for normal fishing boat hulls, where B/T seldom exceeds 3.0.

Conclusions and Recommendations

An easy-to-use graphical method for determining the effects of variations in major hull parameters of fishing boats on controls-fixed directional stability is presented. The method is based on the results of a numerical model that computes hydrodynamic coefficients following a published theoretical approach with adjustments to suit fishing boat hull forms. With the graphs, the fishing boat designer can quickly and inexpensively estimate whether a given hull will be directionally stable and also determine the changes that can be made to alter stability characteristics if required.

The graphs represent a first step in providing the fishing boat design community general information relative to directional stability and course-keeping. As a recommendation, data from model tests or full-scale maneuvering trials, if available, should be collected in order to validate the curves shown on the graphs.

References

1. Comstock, John P., ed., Principles of Naval Architecture, The Society of Naval Architects and Marine Engineers, New York, 1967, pp. 518-529.

2. Ridgely-Nevitt, Cedric, "The Resistance of a High Displacement-Length Ratio Trawler Series," 1967 SNAME Transactions, New York, 1967.

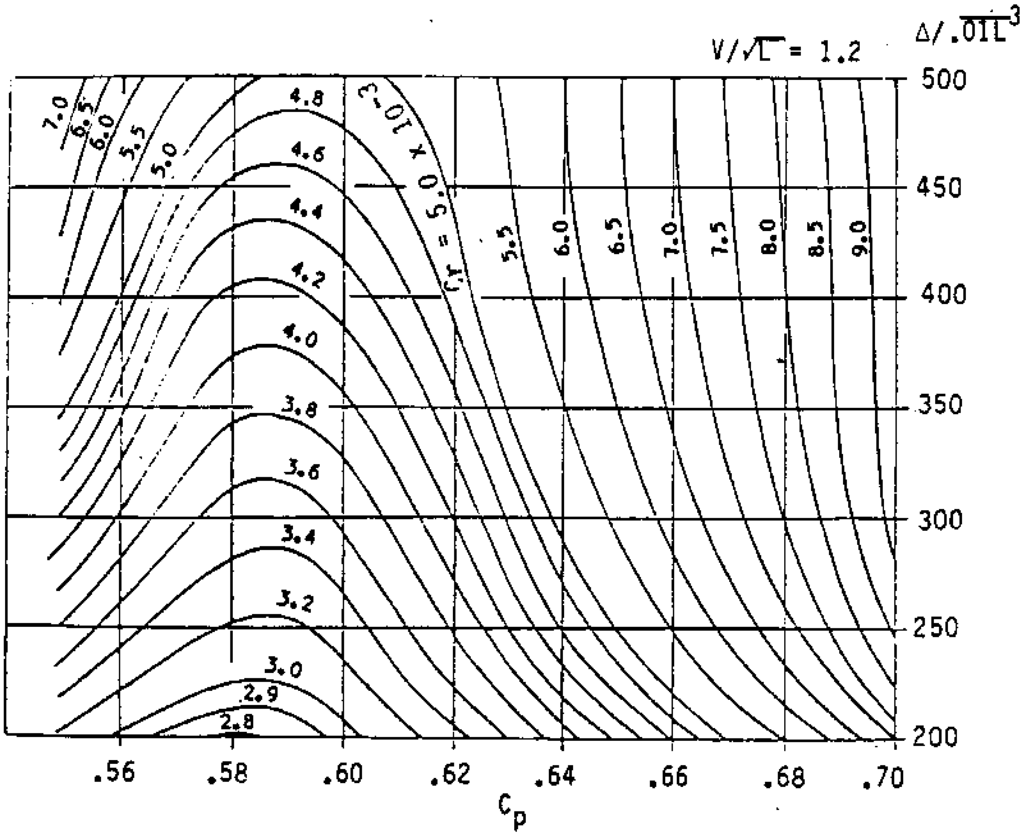
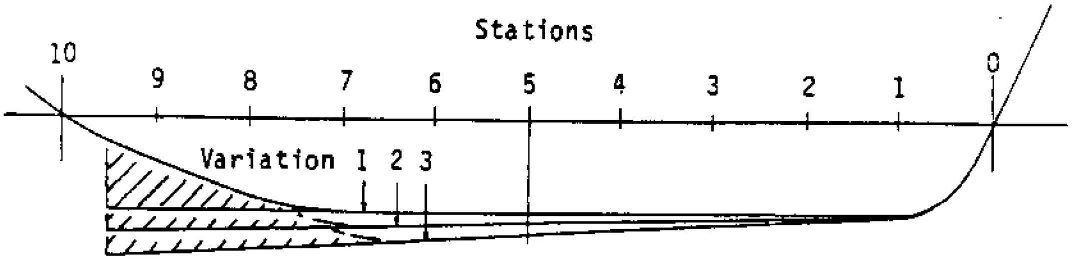


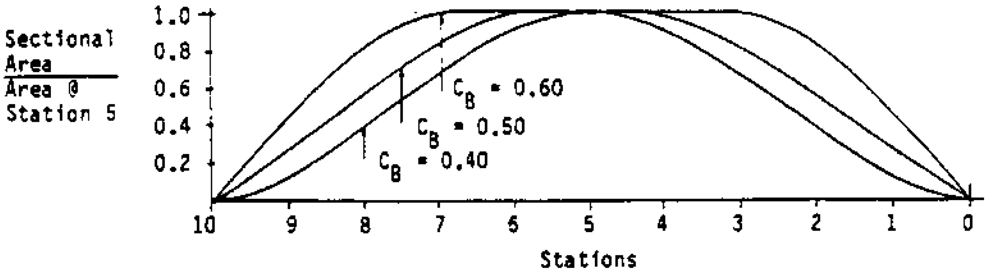
Figure 1. Webb Trawler Series Residuary Resistance Contours for V/√L = 1.2



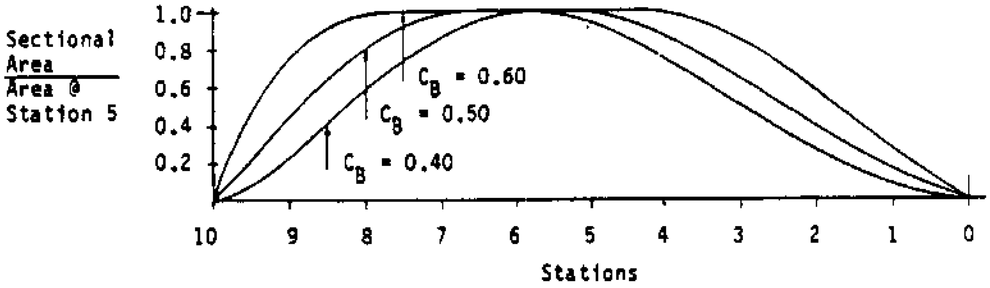
Ordinates of Hull and Skeg Profiles $\left(\frac{\text{Hull Draft} \div \text{Draft Sta. 5}}{\text{Skeg Draft} \div \text{Draft Sta. 5}}\right)$

Station	0	1/2	1	2	3	4	5	6	7	8	9	9 1/2	10
Variation													
1 - No Drag	0/0	0.8/0.8	1.0/1.0	1.0/1.0	1.0/1.0	1.0/1.0	1.0/1.0	1.0/1.0	1.0/1.0	0.65/1.0	0.45/1.0	0.25/1.0	0/0
2 - 0.02L Drag	0/0	0.71/0.71	0.90/0.90	0.93/0.93	0.95/0.95	0.98/0.98	1.0/1.0	1.02/1.02	1.02/1.05	0.76/1.07	0.4/1.1	0.22/1.11	0/0
3 - 0.04L Drag	0/0	0.64/0.64	0.82/0.82	0.87/0.87	0.91/0.91	0.96/0.96	1.0/1.0	1.04/1.04	1.0/1.08	0.60/1.13	0.36/1.16	0.20/1.20	0/0

Figure 2. Drag of Keel Variations



Curves for LCB = 0.50L



Curves for LCB = 0.55L
(Reverse for LCB = 0.45L)

Figure 3. Sectional Area Curve Variations

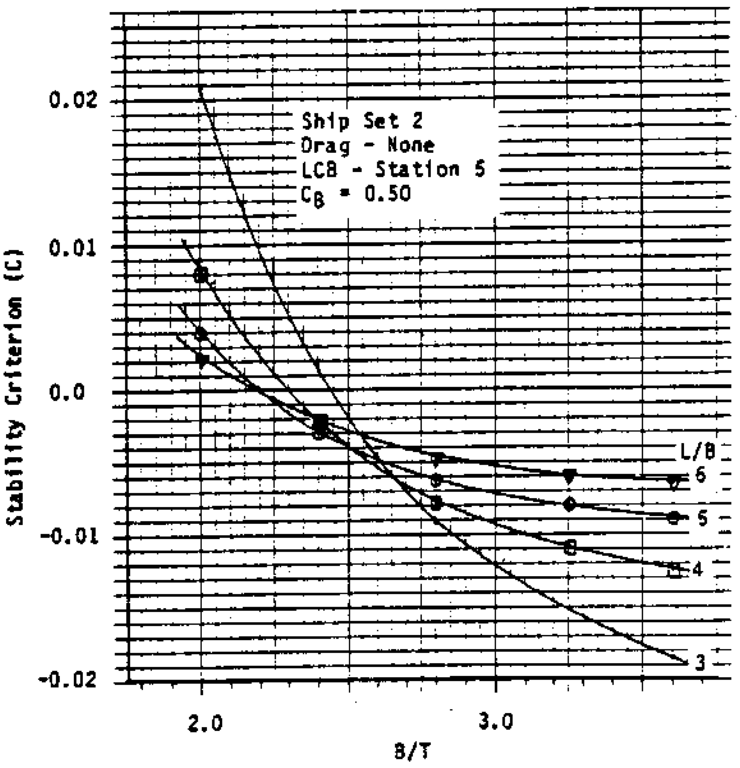


Figure 4. Example Plot of Stability Criterion vs. Beam to Draft Ratio

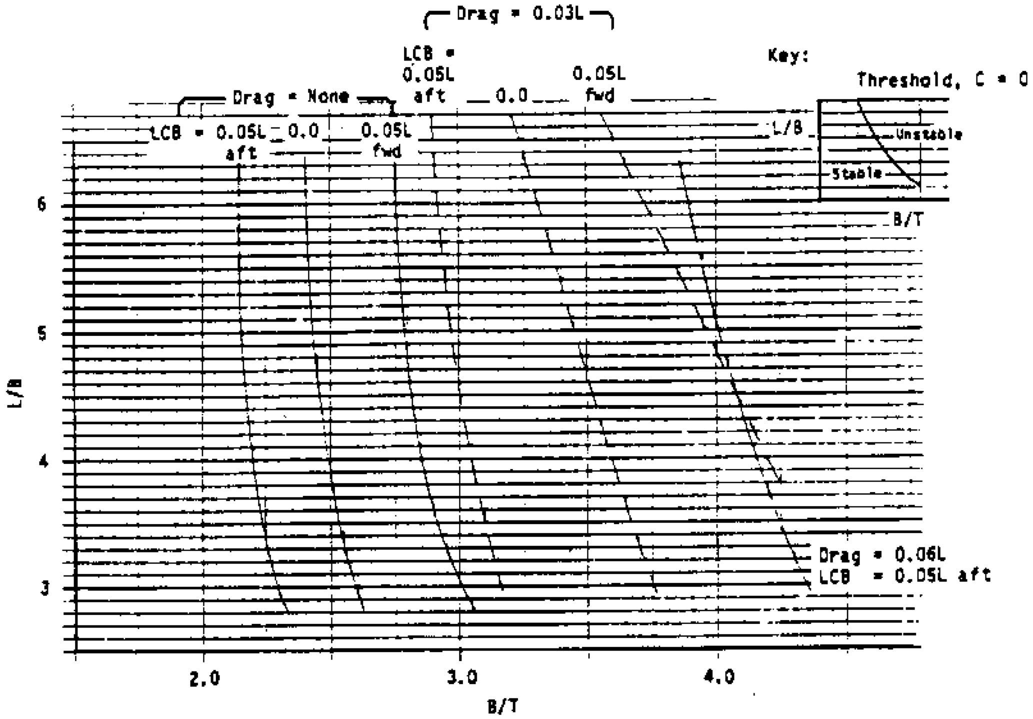


Figure 5. Directional Stability Thresholds for C_B = 0.40

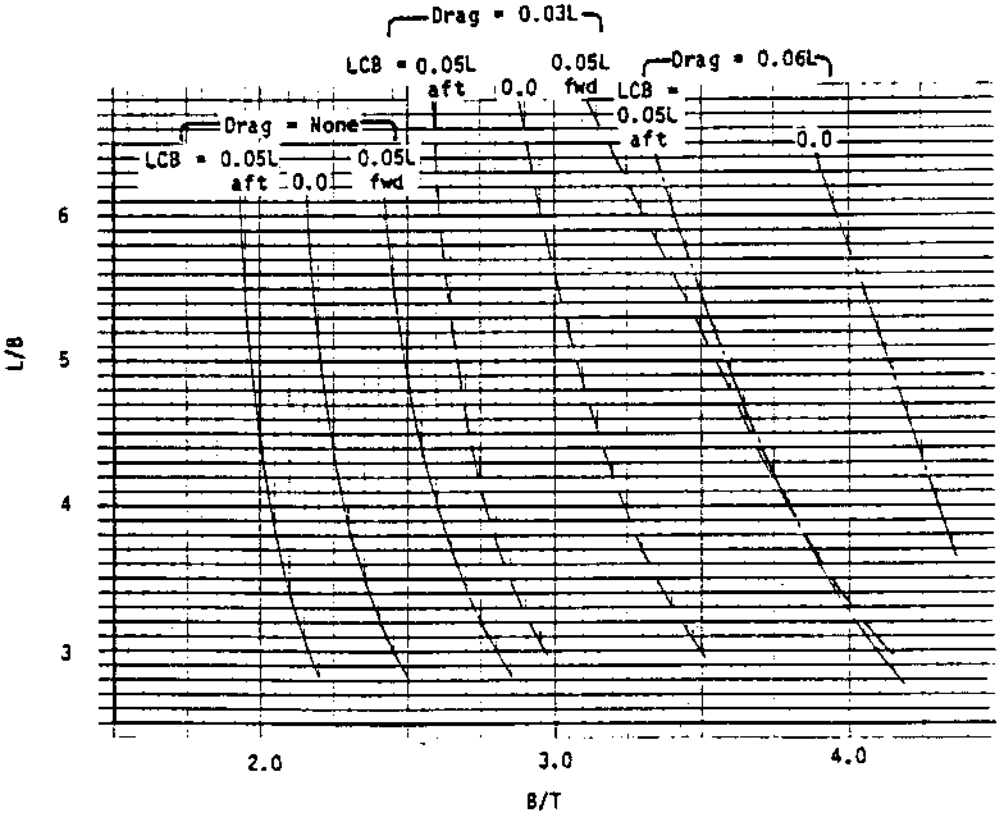


Figure 6. Directional Stability Thresholds for C_B = 0.50

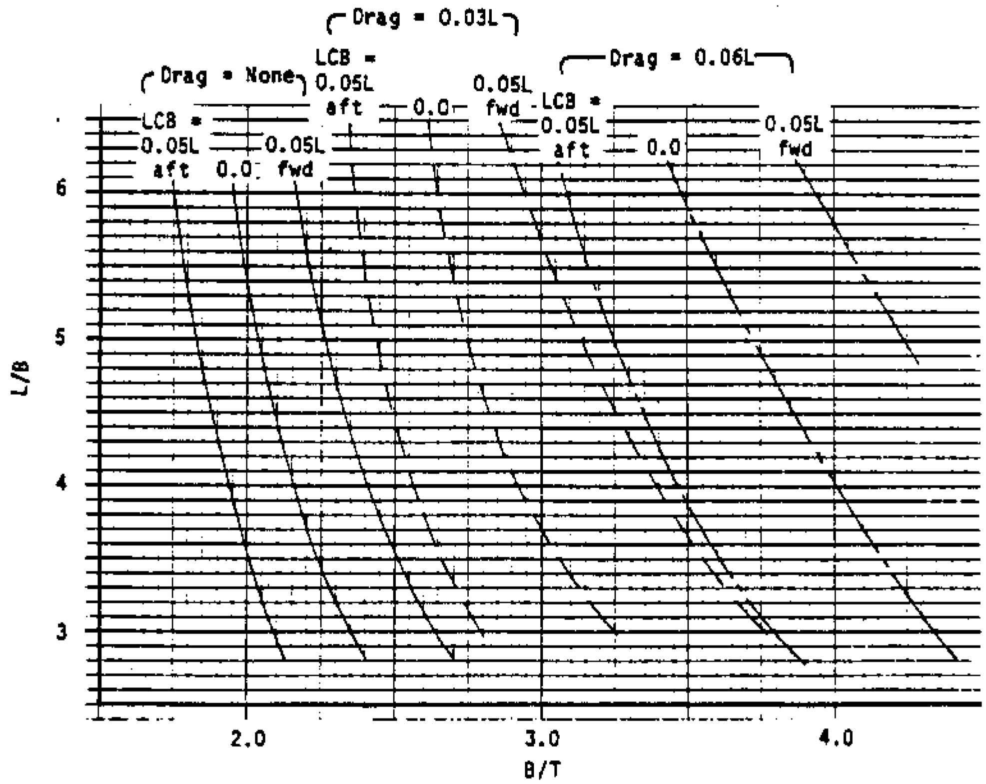


Figure 7. Directional Stability Thresholds for $C_B = 0.60$

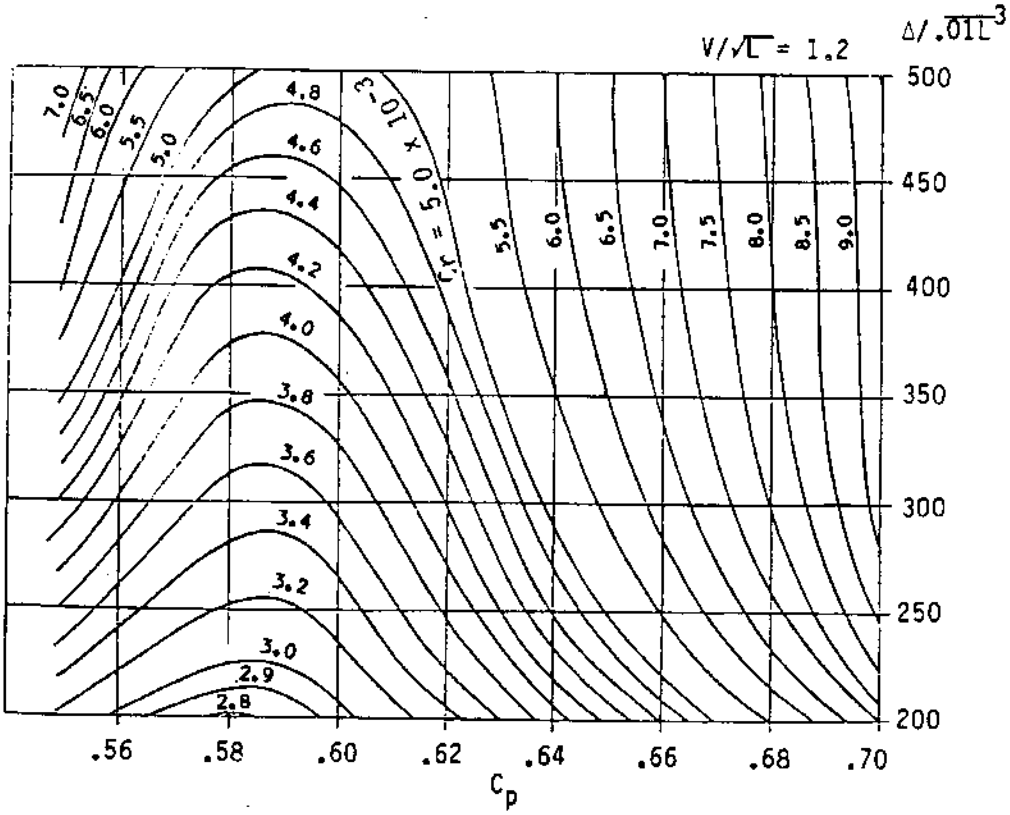
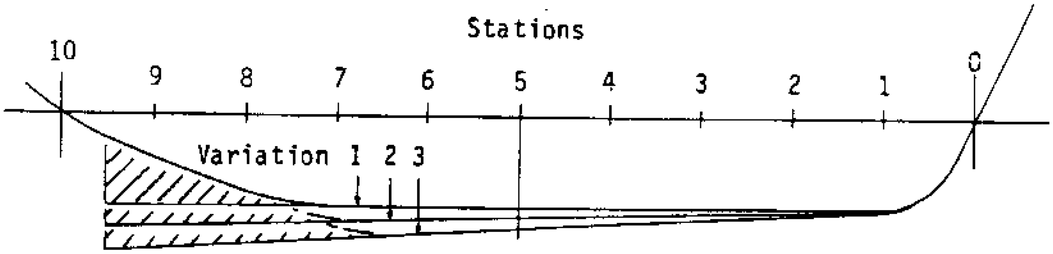


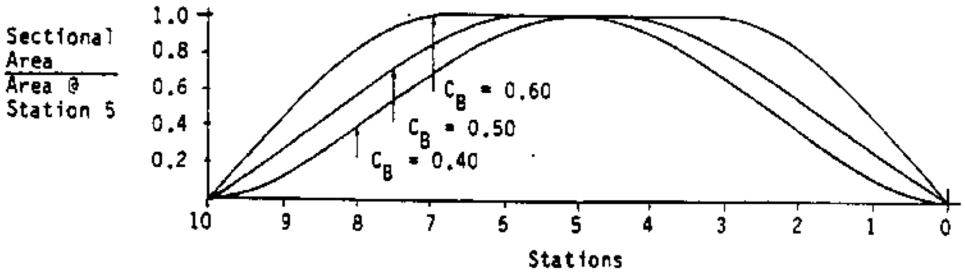
Figure 1. Webb Trawler Series Residuary Resistance Contours for $V/\sqrt{L} = 1.2$



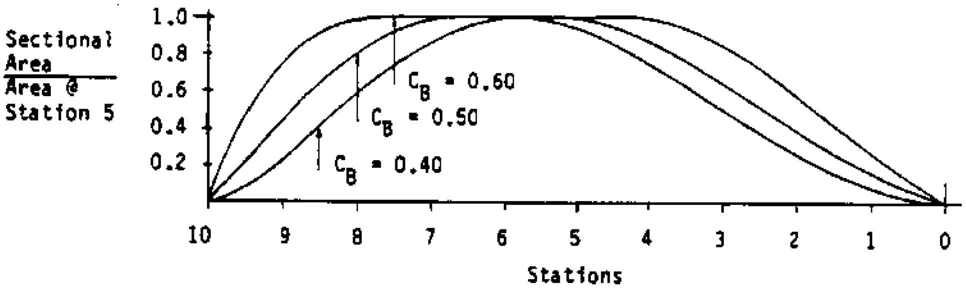
Ordinates of Hull and Skeg Profiles $\left(\frac{\text{Hull Draft} \div \text{Draft Sta. 5}}{\text{Skeg Draft} \div \text{Draft Sta. 5}}\right)$

Station	0	1/2	1	2	3	4	5	6	7	8	9	9 1/2	10
Variation													
1 - No Drag	0/0	0.8/0.8	1.0/1.0	1.0/1.0	1.0/1.0	1.0/1.0	1.0/1.0	1.0/1.0	1.0/1.0	0.85/1.0	0.45/1.0	0.25/1.0	0/0
2 - 0.03L Drag	0/0	0.71/0.71	0.90/0.90	0.93/0.93	0.95/0.96	0.98/0.98	1.0/1.0	1.02/1.02	1.02/1.04	0.78/1.07	0.4/1.1	0.22/1.11	0/0
3 - 0.06L Drag	0/0	0.64/0.64	0.82/0.82	0.87/0.87	0.91/0.91	0.96/0.96	1.0/1.0	1.04/1.04	1.0/1.09	0.60/1.13	0.36/1.18	0.20/1.20	0/0

Figure 2. Drag of Keel Variations



Curves for LCB = 0.50L



Curves for LCB = 0.55L
(Reverse for LCB = 0.45L)

Figure 3. Sectional Area Curve Variations

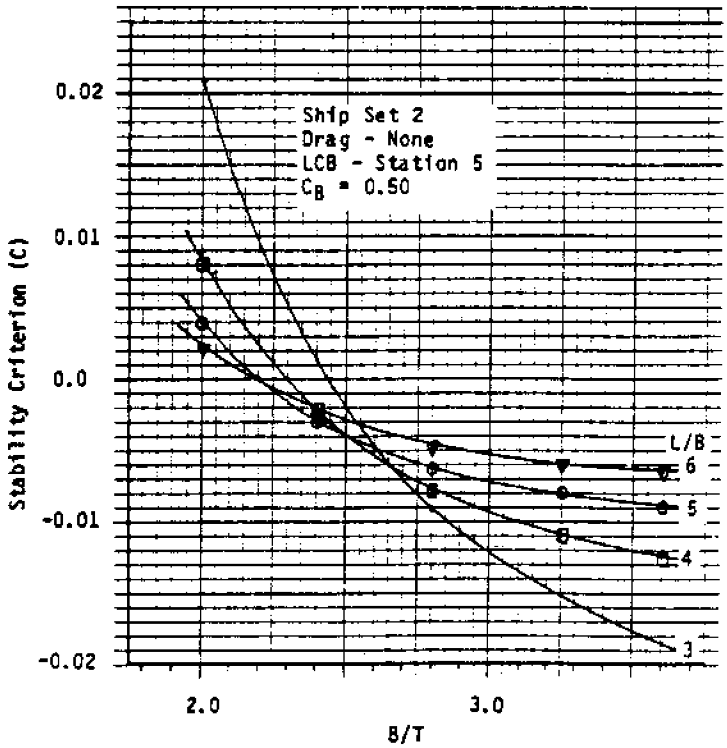


Figure 4. Example Plot of Stability Criterion vs. Beam to Draft Ratio

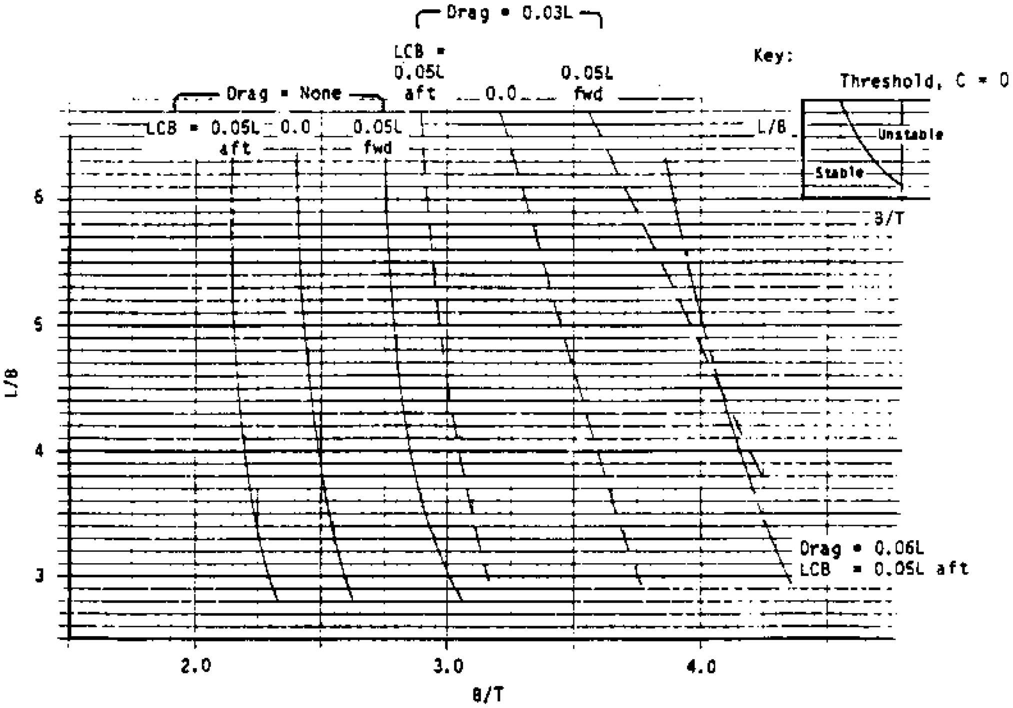


Figure 5. Directional Stability Thresholds for $C_B = 0.40$

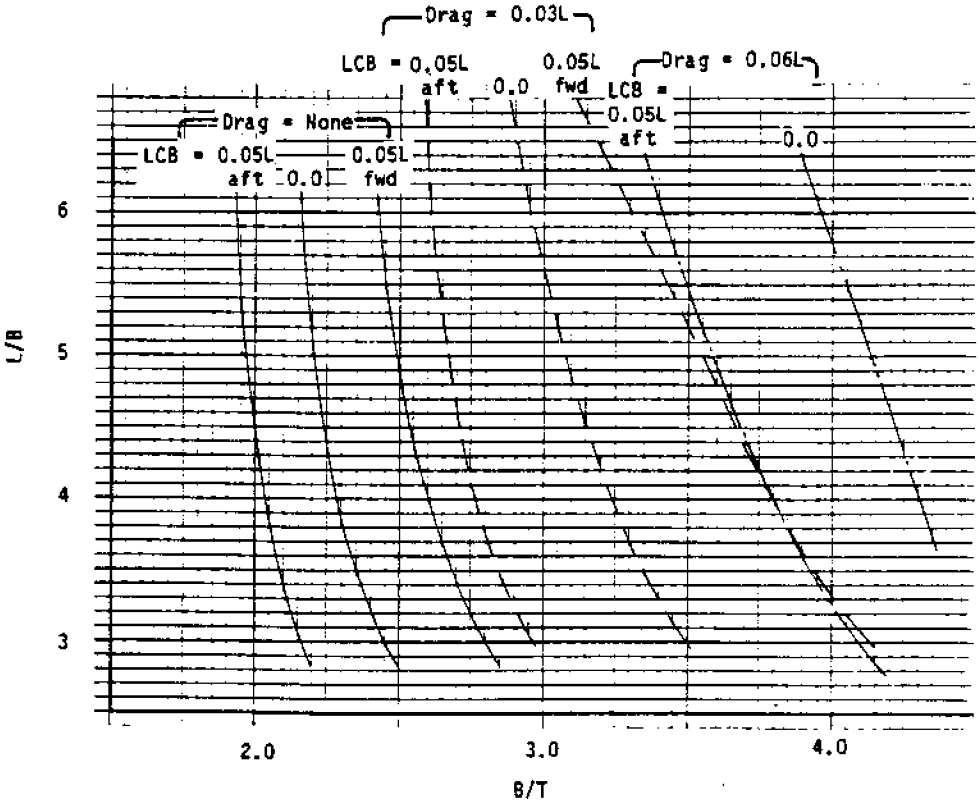


Figure 6. Directional Stability Thresholds for $C_B = 0.50$

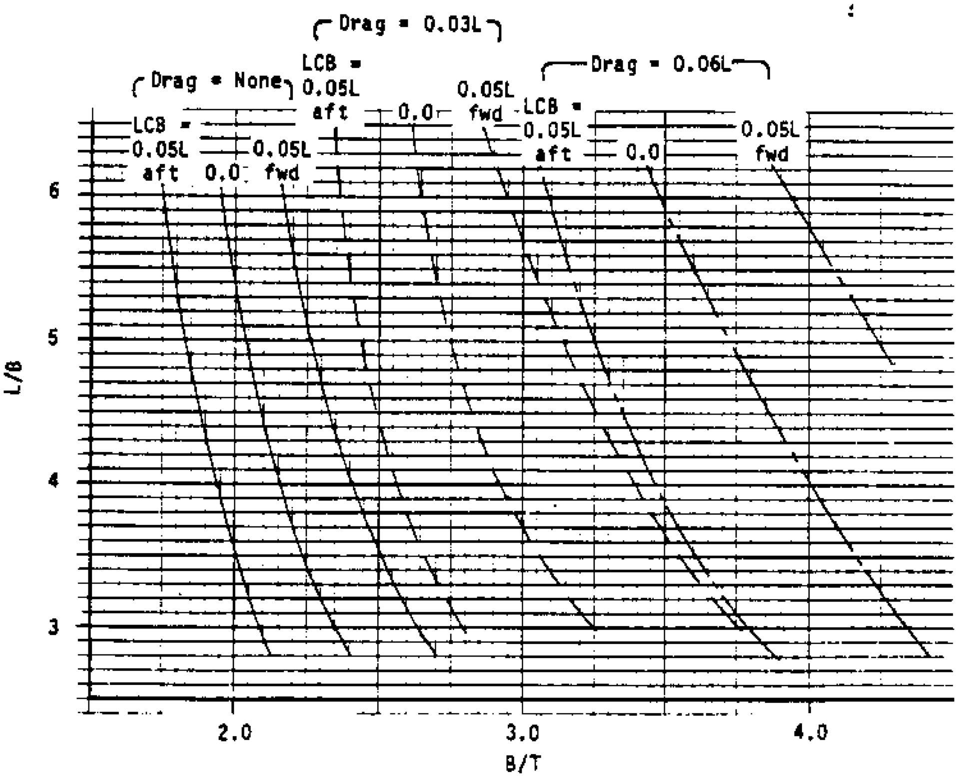


Figure 7. Directional Stability Thresholds for $C_B = 0.60$

FISHING AND SHIP MOTIONS - DESIGN CONSIDERATIONS BASED ON OBSERVATIONS OF OPERATIONS

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Abstract

Ship motions obviously impact on the ability to fish. Not only is production time lost when weather precludes fishing, the weather dependent fluctuations of the landings across the industry cause chaos and inefficiency all down the line in processing, distribution, marketing and consumption. The resultant loss is significant.

This paper presents notes based on a study of Seakeeping and Human Engineering. Practical design considerations are reviewed.

Introduction

Fishermen enjoy a well deserved prestige because they work in small ships subject constant motion from the sea. Designers of fishing vessels share in this prestige because, more often then not, fishing vessels do return safely to port. But just how good a job have designers been able to do, and is there room for improvement?

The New England Goundfish industry presents a good example. When weather precludes fishing production time is lost. Lost time increases the capital costs of fishing. Since this industry deals with a fresh product having a shelf life of only 8 to 14 days, fish prices vary inversely as the amount of good weather. Bad weather yields good prices. Because of the weather dependent fluctuations of landings, the entire system of processing, distributing, and marketing fish is very flexible and quite inefficient. Dockside prices in the summer are less than half the prices for the same fish in the winter. This is not because the fish have changed, but because the industry can not handle both situations with equal effectiveness. Essentially if fish were dependably available year round, the system could be set up to yield the same high added value in the summer as it does in the winter. All of this can not be laid at the door of ship motions, but Figure 1 shows the remarkable

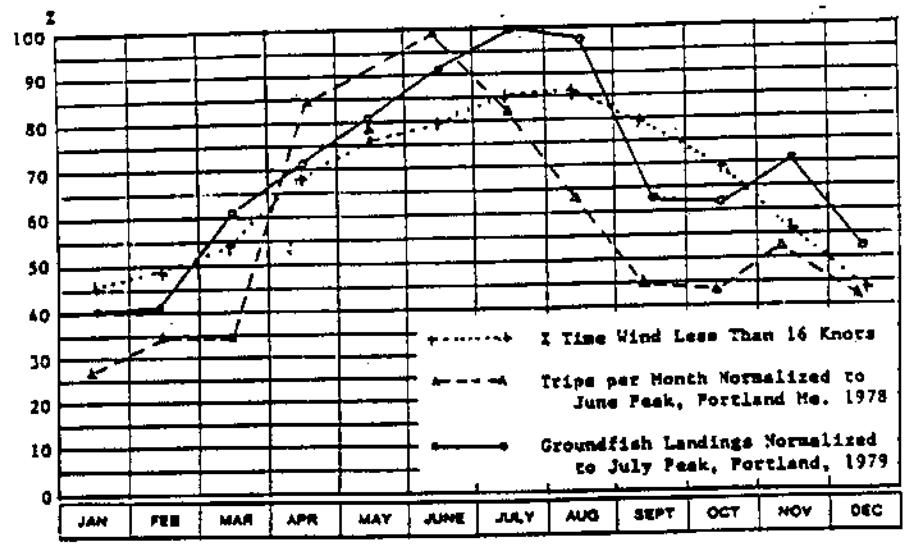


FIGURE 1

COINCIDENCE OF WEATHER
FISHING TRIPS AND LANDINGS

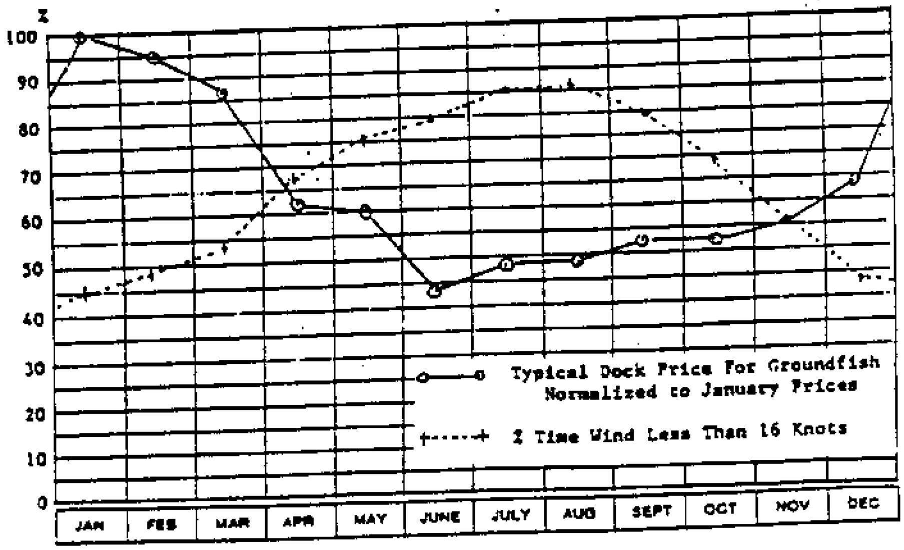


FIGURE 2

SEASONAL FISH PRICE AND WEATHER VARIATION coincidence of good weather, number of fishing trips and fish landings. Note the close correlation of monthly landings to number of trips. This indicates a relatively constant catch per unit effort.

Figure 2 shows the inverse relationship of price and weather. Taking number of trips in the most active month as an indication of fleet capacity, the figures indicate that fleet operates at about 60 percent of its annual capacity. Taking the winter time price as an indication of the obtainable value for the fish, the fisherman recieves an average of 60% of this value. It is probably not overstating the case to say that weather (ship motion) restrictions cost the New England fishing system at least 50% of its potential economic efficiency.

Weather Dimensions

What is this weather that so hampers the industry? Statistics from shipboard observations (large merchants ships) give the following for the Gulf of Maine:

Figure 3
Weather Conditions in Gulf of Maine

% Occurance	Wind Speed knots	Wave Height (sig) feet	Wave Period (sec)
66 %	< 16	< 4 [1.2m]	< 6
81 %	< 22	< 6 [1.8m]	< 7
95 %	< 33	< 9 [2.8m]	< 9

The 16 knot conditions correspond very well with the fishing activity as shown in Figure 1. It appears that "weatherproofing" the operations so that they could be conducted in the 33 knot wind conditions would essentially eliminate the effects of weather upon the system.

These weather conditions seem surprisingly mild when one considers that the industry statistics cited are based on vessels 40 to 130 feet in length [12 to 40 m] fishing from 3 to 200 miles from port [5 to 320 km]. It should also be stated that the fishermen's perception of the situation is not as severe as implied by these statistics. A recent spot sampling indicated that fishermen felt that lack of weather restrictions would add only 20 to 60 extra trips to seasons that already encompassed 150 to 200 trips.

Motion Interference With Fishing

The following descriptions of reasons that motions interfere with fishing are based on observation trips with fishermen, interviews with fishermen and the start of a survey of fishermen on the subject. These all refer to studies of New England fishermen. The importance of one influence or another refers to the frequency with which it seems to interfere with fishing on an industry wide basis.

The first category of motion interference with fishing is: DANGER TO THE VESSEL. This includes swamping, capsize in waves, loss of directional control while running with following waves, and structural damage from slamming. Fishing vessel designers have been concentrating on preventing these events and in a survey of fishermen only the last of these was given as a reason that fishing was stopped because of weather. Designers should perhaps be comforted; danger to the vessel is an important consideration but is not the usual reason that fishing ceases when it does.

The second category of motion interference with fishing is: GEAR DOESN'T FISH. Observations indicate that this is a very important aspect of the problem. It is particularly a problem for trawlers as the ship motions (heave and pitch) are translated to pulsing motions of the doors and net. Experience indicates that the catch rate drops off as the motion level of the vessel increases. This is an important factor in causing vessels to cease fishing.

The third category of motion interference with fishing is: VESSEL CAN'T STAY ON GEAR. This, too, is a very important aspect. Fixed gear fishing methods such as gillnetting and longlining require that the vessel excel at station keeping so that the gear can be worked over slowly (speed less than one knot). As ship motions increase, especially surface drift and loss of control of heading, large strains are placed upon the fishing gear. Also the speed at which the vessel can work its way down the gear is decreased. This, too, is an important factor in causing vessels to stop fishing.

The fourth category of motion interference with fishing is: LOOSE GEAR ON DECK POSES THREAT. This threat can be to the vessel, to the gear itself or to the crew. Arrangements are easily made to secure most gear. There is obviously some danger to the crew while in the process of securing the gear, but the time periods for this danger are quite short and are accepted as a normal part of the the job. While this is an important consideration it was not observed to be a major reason that fishing was halted.

The fifth category of motion interference with fishing is: WATER ON DECK POSES THREAT. This threat is to gear on deck, to the fish being processed on deck, and to the crew. Again this is an important consideration, especially for vessels with very low freeboard, but does not appear to be a major reason that fishing ceases.

The sixth category or motion interference with fishing is: MOTION IMPACT ON CREW ABILITY TO WORK. This includes the traditional seasickness category, which does not appear to be very important in stopping fishing. Slipping and sliding on the tilting deck appears to be a "medium" level problem in causing fishing to cease. A more important reason appears to be the "jerking around" that causes difficulty for the crew in maintaining their body position relative to the ship. Another important aspect is the exhaustion from fighting motion. These last two would have to be classes as major reeasons that fishing activities ceased.

Figure 4 presents a summary of the motion influences that interfere with fishing and ranks them in order of decreasing frequency and importance.

Figure 4
Motion Influences Interfering With Fishing
Ranked by Frequency of Occurance

MOST FREQUENT REASONS
- Gear Doesn't Fish
- Vessel Can't Stay on Gear
- Crews Gets Jerked Around
- Exhaustion from Fighting Motion

MODERATELY FREQUENT REASONS
- Crew Slips and Slides on Inclined Decks
- Water On Deck Poses Threat

LEAST FREQUENT REASONS
- Loose Gear On Deck Poses Threat
- Danger to Vessel

Mechanics of Crew - Motion Problems

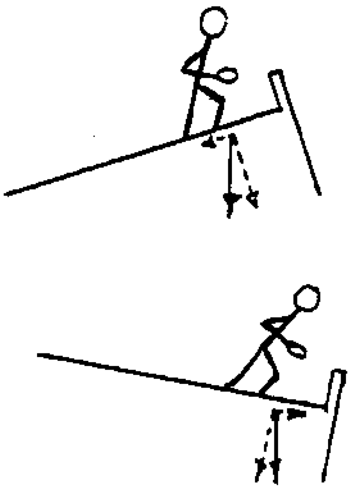
A study of Seakeeping and Human Engineering was recently completed for SNAME panel H - 7, Seakeeping Characteristics. The study illustrated the complexity of the influence of motion of man's ability to work at sea. It also showed that very little is really understood about this interaction.

The study illustrated two important extensions to the commonly used "model" of man interacting with motions. Previous studies had not considered either the predictability of motions nor the amount that the crew was jerked around. During observation trips with fishermen both of these were observed to be important. The amount that the crew is jerked around has a physical measurement, jerk, which is the time derivative of acceleration. Previous studies have made much of acceleration levels, which are related to the frequency squared for sinusoidal motion. Jerk, however, may be more important and is related to the frequency cubed.

The quasi-static picture in Figure 5 shows a good deal about man working on the deck of a moving vessel. Within broad boundaries the force levels themselves are not problems, the man can provide a wide variety of opposing forces to maintain his position relative to the ship. However, unlike a rigid body, say a statue at this position, the man must actively use his brain and muscles to maintain this position. This interferes with the work he is doing, such as securing loose gear on deck, which also requires active use of the brain and muscle system. Thus it is the time variation of the forces and accelerations that is the problem rather than the forces themselves.

FIGURE 5

Change of Deck Tangent And Normal Components of Gravitational Force as Vessel Heels



The jerk problem can be immediate, in that the crewmen may not be able to simultaneously hold their position and perform their tasks, or the problem may be cumulative, showing up as fatigue and exhuastion. Both immediate jerk and fatigue were observed to be important reasons that fishing ceased. Fishermen who have installed paravanes (flopper stoppers) have reported that, of all of the effects from the 30 to 50% reductions in roll amplitude, the biggest improvement is in the endurance of the crew.

Design Considerations

It is not likely that any set of simple design considerations are going to extend the fleet operations to 33 knots wind conditions. However, some boats are reputedly good sea boats, while others are not. There are reported differences even amoung sister ships. Fishermen offer many theories as to why this vessel is better than that, but no definative work seems to have been done on this subject.

Too little is understood to be able to offer numerical guidelines or methods of analysis for designing "good" boats as far as motions are concerned. The best that can be offered as design considerations are really only common sense items. It should be noted, however, that many fishing vessels fall outside the guidelines of these common sense considerations. The results are always predictably bad. Many vessels have pilothouses way forward where pitching motions are extreme. Perhaps this is acceptable in the waters for which these vessels were originally designed, but vessels today are mass produced and marketed and moved around the world.

Design considerations fall into three categories, designing the vessel to reduce motions, arranging which tasks take place in which motion environments on the vessel, and arranging details of body supports, etc., at the work stations.

In the category of reducing ship motions roll and pitch can be reduced somewhat by increasing the damping influence of fins, bilge keels, chines and paravanes. There is ample fishermen testimony to the effectiveness of these measures. Interestingly, although fishermen like chine boats, there are complaints about vessels with too sharp a chine possessing too "jerky" a motion as the sharp chine suddenly comes into play during a roll. $\overline{GM}$ (metacentric height) also comes into play, although in a complicated way. Increased $\overline{GM}$ is needed for stability safety, and it also tends to reduce the roll amplitude. The reduced roll amplitude is helpful in reducing accelerations and jerk levels, but seems to be more than offset by the fact that increased $\overline{GM}$ leads to higher roll frequency and thus higher acceleration and jerk. In general increased $\overline{GM}$ is associated with a more "jerky" motion. $\overline{GM}$ in excess of whatever is really needed for safety is probably detrimental to the vessel's performance. This of course opens a whole debate about the appropriate trade off between ultimate vessel stability and operating concerns.

Closely related to the motion design considerations are those of station keeping in rough weather. Windage should be kept down and the vessel's heading relative to the wind should be easily maintained. Boats with high bows and low sterns are difficult to keep on course when working over fishing gear at low speeds in high winds unless the underwater profile is designed to match the windage profile. Extra maneuvability is desirable for vessels fishing fixed gear and side thrusters would probably be helpful.

The location of various tasks should be arranged with consideration of the motion environment. The placement of pilot houses way forward or very high are common examples of arrangements that could be improved from a motion design standpoint. Everything is a compromise, however. For maneuverability the hauling of fixed gear over the side should be located as far forward of the vessel's turning pivot point as possible. From a purely motion standpoint the hauling station should be located as close to amidship as possible in order to minimize the motion levels and also the relative motion between the station and the sea surface.

Probably the area with the most immediate potential is in the detailed design of workstations and tasks for crewmembers. More attention could be given to providing convenient handholds and body supports. Tasks could be designed to eliminate the handling of heavy objects and to cut down on the amount of "travel" required on a pitching deck. Modifications like these do not need to be expensive and will address one of the most important motion influences that interferes with fishing.

Conclusion

There certainly is room for improvement in the design of fishing vessels so that ship motions will be less of a restriction to operations. There is also a great deal yet to be learned.

The fishing industry has perhaps more to gain than anyone else from the study of how ship motions interfere with operations at sea. Who else goes down to work the sea in such small small ships and expects to do so in all kinds of weather. Perhaps this is why the fishing industry already has a leadership role in the study of these things. Some of the most practical published work on this subject is that of Capt. Walter Möckel in the "Fishing Boats of the World," Volumes I and II. Unfortunatley very little seems to have been done since then.

One has to hope that discussions today and future work on this subject will yield better understanding of this issue. Perhaps we can look forward to more quantative guidelines at future conferences.

A NEW APPROACH TO THE PROBLEM OF SHEATHING WOODEN HULLS

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Abstract

In the past, attempts to sheath large wooden vessels with reinforced plastics have largely been unsuccessful due to delamination of the sheathing from the wood in a relatively short time. This paper describes a new technique of using an elastomeric adhesive to bond the sheathing to the wood. The flexibility of the bond allows some movement between the sheathing and the planking, reducing stress concentrations and greatly reducing the likelihood of delamination. The theoretical reasons for using this approach are described as well as field experience with the system. Case histories are cited along with pictures of the application process and of the finished boats.

History of the Problem

For many years builders have tried to cover planked wooden hulls with a membrane in order to eliminate leaking, atacks from marine organisms or to retard marine growth. Copper, canvas, and reinforced plastics have all been used with varying degrees of success. When reinforced plastics was introduced, it seemed the ideal material for this purpose. While the covering did not by itself retard marine growth, it did offer a continuous waterproof membrane which strengthened the hull, stopped leaking, and protected the wood from marine borers.

The process was relatively easy to do, particularly if the hull was turned upside down. The materials were not expensive relative to the benefits which were derived. The first boats which were covered were naturally small boats although it was clear that large vessels such as large fishing vessels would benefit if a workable process were available. There were attempts made to cover large vessels using the standard techniques but, unfortunately, although the process worked well on smaller boats, problems began to develop with the jobs done on the larger vessels. Some rules on successful sheathing practice began to emerge:

1. Small boats, particularly plywood boats and particularly fir plywood boats were relatively easy to cover and were usually very successfully done with the standard polyester resin and glass cloth technique.

2. A better bond could be developed between the wood and the glass reinforced plastics (GRP) covering if a chopped strand mat (CSM) were used first on the wood.

3. Oily woods and hardwoods were difficult to bond to with polyester resin and epoxy resin should be substituted for

a better bond.

4. Glued construction hulls, either double planked, cold molded, or strip planked were much better candidates for sheathing with GRP than were caulked carvel planked hulls.

5. Large carvel planked hulls (over 40' or so) were difficult to successfully sheath with GRP. Delamination of the GRP from the wood was likely to occur in a short time.

6. Any large vessel was difficult to sheath because of the problems of doing the overhead GRP work on the bottom of the hull. If the hull could be tipped to the side this problem was greatly aleviated, but this was difficult to do on larger boats.

The major problem with sheathing large caulked hulls remained that the sheathing was likely to delaminate from the wood. It is recognized that differential movement between the wood and the GRP is the cause of the stresses which develop and attempts to reduce these stresses have included using a more flexible reinforcement in the sheathing (Dynel and polypropylene fabrics) with epoxy resins. In the U.K., the Cascover process was developed. This process involves gluing a nylon fabric to the wood with Cascophen Glue.

Another technique to deal with the tendency of the GRP to delaminate from the wood has been developed by Mr. Alain Vaistes of Mattapoisett, Massachusetts, U.S. Mr. Vaistes does not attempt to develop a bond between the sheathing and the wood, but rather mechanically fastens the sheathing to the hull with nails or staples. These vessels are usually tipped to the side to facilitate the work. The author understands that Mr. Vaistes has done many successful jobs up to 60' vessels.

In late 1978 the author began to develop a process for sheathing carvel planked hulls which would address the problems of the delamination of the sheathing and the problem of working the wet GRP process overhead on the bottom of large wooden hulls without having to turn or tip the hull. It was recognized that the process, if successful, would be most useful in places where the kind of equipment necessary to tip a large hull was not available.

A full scale half section of a trawler hull was constructed and the first work was on the techniques to eliminate the necessity for tipping the hull. It was found that C-Flex Fiberglass Planking could easily be applied to the overhead if it was applied dry into an adhesive paste. C-Flex is a 12" wide unidirectionally reinforced material which has high strength (67000 psi tensile, 46000 psi compressive) in the long direction of the roll. The C-Flex was applied perpendicular to the run of the planking, that is from gunwale to keel. Applied in this manner, the high strength of the material is orientated in the proper direction to withstand the high loading of expansion of the planking. Also, it was found that the material was very easy to apply in this direction. Once the upper end of the strip was secured in place, it proved very easy to bend the rest of the plank into position, even working on the overhead. C-Flex has precured pultruded rods as part of its construction and it proved necessary to staple battens every 12" for so to hold the material in place properly bedded into the adhesive. The adhesive used in this first series of tests was a filled polyester paste.

Once the bedding adhesive had cured, the battens could be pulled off. The C-Flex was then saturated with resin and several layers of chopped strand were applied to finish the job. The chopped strand mat layers proved to be relatively easy to apply if the upper end was done first, then the rest carefully rolled down and under the hull.

This application technique seemed to solve the problem of being able to sheath a hull easily without tipping or turning the hull. Most of the material was applied dry, and the material which was applied wet was manageable.

The next step was testing of the full scale model by cycling it through wet-dry cycles. Unfortunately, the sheathing separated from the planking after the first cycle and close examination revealed that the bond between the wood and the GRP had not broken but that the wood fibers beneath the bond had failed.

What was observed was that there was considerable movement of the planking in a direction normal to the plank surface and this movement caused high peel forces which tore the sheathing from the wood. It was theorized that the sheathing, which was 1/4" thick in this case, had reached a degree of stiffness where it would resist bending beyond the capacity of the wood fiber tear-out strength normal to the plank surface. This explained why sheathing systems would work on smaller boats, where the sheathing was thin and not of sufficient stiffness to resist the tear out strength of the wood fibers. The author had sheathed a 36' sailboat in the late1950's where he had faired the hull to the point where no plank seams showed before launch, and he had observed, to his horror, every seam become obvious within two weeks of launch as the planking regained its moisture content and warped to a new shape.

It was obvious that there were two major kinds of movement of the planking that exerted high peel stresses on the bond between the sheathing and the wood.

1. Warpage fo the plank.

2. Slippage of the planks.

It was clear that no improvement in bond strength could overcome these problems, but that an entirely different approach was needed. After testing many types of adhesives, a polyurethane elastomeric adhesive was selected. This particular adhesive bonded tenaciously to every type of wood tested, including oak, teak, cypress, mahogany, and fir. It is a moisture cure product, and has the capacity fo bonding well to damp wood. Furthermore, and most important, it has high elasticity (up to 300% tensile elongation before break) and thus allows some movement between the sheathing and the wood without developing the high stress concentrations of a rigid adhesive. Wet strength retention was excellent and in most cases tested the adhesive system strength exceeded the strength of the substrate. In all cases the peel strength of the system was dramatically improved over a rigid adhesive.

The decision was made to go ahead with a full scale test of the system. The 55' ACF motor cruiser "Sally Forth" owned by Alton McIver of New Orleans was the first boat to be done. The boat had been built in 1929 and showed the effects of its almost 50 years: thinned planks from many sandings, loose fastenings, broken frames, a tendency to sag when being hauled, etc. The job was finished in spring 1979 and the boat is stiff, does not leak, has showed no signs of delamination or signs of rot caused by the sheathing system. Since that time many vessels have been sheathed, including many commercial fishing boats and, the largest to date, a 138' ex-minesweeper.

There have been no reports of delamination, even though some of the vessels have been in heavy commercial fishing service. Furthermore, many of the jobs have been done outside with no cover and, even under these bad conditions, satisfactory jobs have been possible.

The vessels after sheathing are stronger, do not leak and are able to perform to like new standards in commercial service.

Two examples of commercial fishing boats done to date are:

The 65' shrimp trawler "North Star" owned and sheathed by Bell Buoy Seafood, Edisto Beach, South Carolina. Before sheathing when she had been hauled with water in her bilges the weight of the water had pushed several planks off her. These planks were refastened, but no other refastening was done. She was sheathed only to the waterline. Last winter she went on a shoal near Frampton Inlet and pounded in breaking surf for several days. The rigging was bent from the force of the pounding, the rudder shaft was bent so that the shaft lay right up along side the hull. Yet when she was pulled off there was no hull structural damage except that the samson-post and stem were pulled out of her in the refloating operation. The hull only leaked slightly along the garboard where the sheathing had not been carried under the keel but had been stopped on each side of the keel. Delamination had occured in areas where bilge oil leaking from the hull had prevented a proper bond during the application of the sheathing. No other areas fo delamination have been found.

The 70' snapper and longline vessel "Friendship II" - Before sheathing this vessel had been given up for scrap and was destined for the artificial reef. Old refrigerators, etc had already been loaded aboard. Raffield Shipbuilding took on the job to sheath and to re-equip her and she has recently been appraised at $300,000.00

This process has proven itself well in short term testing. Only really long term testing will finally prove or disprove the worth of this system.

Application Procedures

Hull Preparation

In order to obtain a proper bond to the planking, all paint and foreign matter must be removed from the wood. The best technique to use to remove the bottom paint is to sandblast. Sandblasting is fast, will give the wood an excellent texture for the adhesive to grip to as well as to remove rotten wood. Unfortunately, sandblasting will not work very well on the topsides as the topsides enamel is usually harder than the wood and the sandblasting will eat the wood around patches of paint. The best way to remove topsides paint is to use a large disc grinder with coarse paper (16 or 24 grit). Grinder swirls are no problem but, of course, care should be taken to keep, or make, the surface as fair as possible.

Rotten planks should be replaced. Fits are not really important, although a workmanlike job should be done. Bad spots in the planking may be spot patched with pieces of wood glued in place with the same adhesive used to bond the sheathing in place. Again, fit is not really important as this adhesive is gap-filling. Seams need not be reefed but any loose material should be scraped away.

Thru-hull fittings should be removed, if feasible. Same for rudder fittings, struts, etc. In some cases where these fittings are extremely difficult to remove it has proven possible to work over them and to bond and mechanically fasten the sheathing to the outside of the fitting. The C-Bond adhesive will bond well to bronze and, particularly if the bond is backed up by mechanical fastenings, will adequately seal the joint.

Some re-fastening of the planking may be necessary but not as much as would be needed if the boat were not to be sheathed. One of the main purposes for planking fastening is to keep the planks from popping off the frames. The sheathing will perform this function once it is in place, so extensive re-fastening may not be necessary. Several boats have been done with no re-fastening at all, even though the fastenings were known to be bad, and no problems have developed.

Bad frames will have to be replaced or sistered on a case by case basis. The sheathing, with its 1/4"+ thickness and extremely high strength, will replace some of the function of the framing and make some of the framing redundant. This must be judged on a case by case basis. Some of the boats that have been sheathed still have cracked or bad frames in them and are holding up in heavy service with no problems to date. More experience will be needed before a recommendation can be made on this point.

Adhesive Application

The one-part polyurethane adhesive which has been used for this process is a viscous mastic. It can be made very workable by heating to 100 deg. F or so. It is applied with a 12" wide deep notched trowel at a rate of approx. 25 sq. ft. to the gallon. The pot life fo the mastic is at least 4 hours and one person applying the adhesive can easily keep ahaed of two people applying the C-Flex.

Applying the C-Flex

The first plank should be applied amidship and the work should proceed forward and aft from there. The planks are first stapled across the top, then held in place with wooden battens every foot or so. A pneumatic stapler is necessary for this part of the job. It is essential that the C-Flex be pressed completely into the adhesive. After a short instruction period the applicator will be able to visually determine if enough battens have been used by observing whether the adhesive has oozed through the C-FLEX to a sufficent extent. A good three man crew can apply approximately 75 sq. ft. of adhesive and planking per hour. The adhesive will take approximately 24 hours to cure and then the battens may be stripped off.

After the battens have been stripped off the next step is to saturate the C-FLEX with resin. Some care must be exercised at this point to ensure that the resin is catalyzed to cure slowly enough to allow time to fully saturate the C-Flex. The heavier grade of C-Flex contains 65 ounces of fiberglass reinforcement per sq. yd. so it takes a fair amount of resin to fully saturate. The proper procedure to follow to ensure complete saturation is to work three strips at once. Roll the resin onto the first strip, then the second, then the third. Then repeat the process twice before moving to the fourth strip. Each strip will thus be coated three times with some time between applications to allow for saturation. When the C-Flwex is fully saturated the dark brown color of the adhesive will be clearly visible.

While the resin is still wet additional staples are driven through the C-Flex to mechanically fasten the sheathing to the hull. This is usually done at a rate of approximately 12 staples per square foot. As we gain experience with this system we are questioning whether or not these staples are necessary. At least one boat has been done without these staples. The aircraft industry calls these "chicken fasteners". They don't really seem necessary, but we are too chicken to not use them.

Lay-Up of Laminate over the C-Flex

Once the C-Flex is cured, the surface is sanded to knock off high spots and then the rest of teh laminate is applied. Vessels up to 70' LOA will need approximately 3 ounces of CSM over the C-Flex, with additional layers of CSM and Woven Roving (WR) used to reinforce specific areas such as the garboard, stem and transom joints, etc. Typically these areas will be additionally reinforced with three plies of (1.5 oz. CSM + 24 oz. WR). This laminate is doubled under the keel and across the stem, giving a laminate thickness in these areas, including the C-Flex, of approximately 5/8".

Service experience seems to bear out that this is adequate for vessels up to 70' or so. A case in point is the 65' vessel "North Star". This vessel was sheathed with the laminate as suggested with the exception that the sheathing was stopped slightly above the waterline and it was not carried across under the keel as suggested, but stopped on each side of the keel. As mentioned previously, this vessel was grounded on an exposed bar last winter and pounded in heavy surf for four tides. She was washed 1/4 mile across the bar. The concensus of experienced fishermen who witnessed the event was that even a new wooden vessel would not have survived but would have broken up. The only place where she leaked when she was refloated was along the garboard where, according to the owner, the sheathing had not bonded well in some areas due to oil leaking through the garboard seam during the sheathing process. The vessel had not been refastened prior to sheathing and no other structural work had been done such as rebolting floors, etc.

Fairing and Finishing the Hull

This can be the most time consuming step in the process if a yacht type finish is required. The processes of filling and sanding, priming and topcoating are well known and are the same as are used to finish a custom GRP vessel. Commercial fishing boats do not require this type of finish and the author has recently developed a technique of reducing the time required to achieve an acceptable commercial finish.

This technique consists of laying a layer of aircraft peel-ply material over the last CSM as it is being applied. Peel-ply is a fine nylon cloth which, when wet with resin and a fine spray of styrene monomer, can be troweled with a wide steel trowel to a surprisingly smooth and fair finish. What really happens is that the nylon cloth acts as somewhat of a screed to allow the resin in the CSM to be redistributed to fill in the low spots and thin down the high spots. Once the resin has cured, the peel ply is pulled off, and the surface is ready for paint. The surface that results after peeling the peel ply away is a finely textured surface which will accept paint without any further work whatsoever. The peel ply is not expensive (approx. $.25 per sq. ft.) and the process seems to be easy to do, although no full scale vessels have been done with this procedure as yet.

Conclusions

The C-Flex Sheathing System has been in use for approximately five years. The original objectives of the system, to develop a system which would resist delamination and which would be easy to apply, seem to have been met. To date there have been no major problems with the system. An added plus seems to be that the sheathing adds a substantial amount of strength and stiffness to a hull. This has allowed older commercial fishing hulls to be put back into service on a "like new" basis. Only more time and experience will prove whether this system has solved all the problems of sheathing large carvel planked hulls. It is the author's hope that boatowners, shipyard operators and surveyors will appraise this system critically and report on problems and advantages as they see them.

Material cost breakdown per square foot, in U.S. Dollars: (1984)

Adhesive	$1.91
C/FLEX (CF-65)	1.58
Resin	.71
3 Oz. Chopped Strand Mat	.30
Catalyst	.03
Acetone	.08
Staples	.20
Miscellaneous	.08
TOTAL COST PER SQUARE FOOT	$4.89

This translates into a total material cost to cover a 55' fishing boat of approximately 1700 square feet to be about $8,313.00. Cost and material requirement will vary, depending upon size of vessel.

POTENTIAL FOR ADVANCED BRAYTON-CYCLE ENGINES FOR COMMERCIAL FISHING VESSELS

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ABSTRACT

A strong potential exists for the development of an engine that would be more efficient than the advanced Diesel engine both at full and part power, and that would also be lower in first cost, have a lower mass and volume, and require less maintenance.

Although this engine could be produced with conventional technology and materials, to realize its fullest potential it depends, as does the advanced Diesel, on the development of improved ceramics. There are at present several research and development programs, funded privately and by the U.S. and other governments, that are showing very promising results for the application of ceramics to both types of engines.

Ceramics or certain other nonmetals would enable high turbine-inlet temperatures to be used in small gas-turbine engines. Large gas-turbine engines such as aircraft jet engines employ air-cooled metal turbine blades to allow high temperatures to be used. Small engines (below 1 MW) cannot use air-cooled blades with economy. Yet high inlet temperatures must be used if the efficiency of small gas-turbine engines is to be improved. The use of ceramics would bring other advantages: the cost would be far lower; designers could use more turbine stages with lighter loading and increased efficiency; we would not be dependent on strategically scarce materials like cobalt and chromium; and wasteful use of compressed air for turbine-blade cooling would no longer be needed.

The engines we are proposing appear to have some advantages over current experimental gas-turbine engines using ceramics. We show in this paper that the use of highly effective ceramic heat exchangers enables the pressure ratio to be reduced from the frequently used range of 5-15 to about 3. The result is an engine in which stresses and speeds can be so reduced as to allow the compressor to be made from a commercial reinforced plastic, while giving outstanding efficiency and range of operation. The turbine-blade stresses would also be greatly reduced. The resulting engine is predicted to give 10 to 30 percent improvement in fuel consumption over the advanced Diesel engine at full and part power, while retaining its advantages of small size, high reliability, and potentially lower cost.

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Advanced engines

Background

Although the events that set off the "energy crisis" of a decade ago produced an unprecedented increase in the price of petroleum fuels, the inflation induced in most countries has since reduced the relative effect considerably. Nevertheless, there is still sufficient remaining differential to cause fuel costs to be a very significant component of the total costs of commercial fishing. The long-term, and perhaps the short-term, trends are for further increases in the costs of fuel relative to other costs. The current study was undertaken, therefore, to examine whether or not the principal engine used by fishermen today, the diesel engine, could be surpassed, at least in fuel efficiency, by turbine engines.

The diesel engine is already the most efficient of prime movers, exceeding in certain cases the efficiency of the largest steam-turbine plant. Moreover, the peak thermal efficiency of the most advanced diesel engines is being continually increased, reaching over 50 percent in some large marine units. In contrast, the efficiency of steam-turbine plant is decreasing because of the effects of legal requirements to reduce sulfur emissions.

The diesel engine has other virtues. Its idling fuel consumption is the lowest of any of the principal engines, and its part-load efficiency is also very good. Its operation is little affected by water, a very favorable attribute in comparison with the spark-ignition engine, and although a salt-laden atmosphere is not beneficial, it is also not crippling. The engine is extremely reliable so long as somewhat demanding maintenance schedules are observed.

The diesel engine would appear to be a paragon, difficult to improve upon. That we should have the temerity to suggest that a better engine for such a specialized duty as that demanded by US fishermen could be produced requires a thorough and convincing explanation. We do our best below. Full documentation of our arguments will have to await the publication of our final project report for Sea Grant, due in July, 1984. A large measure of the justification for our approach is given in a just-published text by one of the authors (1).

The emergence of the high-efficiency gas-turbine engine

Although the first gas-turbine engines were designed for industrial purposes, research and development were soon dominated by the particular requirements of military and commercial aircraft engines. All heat engines are endowed with improvements in both thermal efficiency and specific power (a measure of the power-to-weight ratio) if the maximum temperature of the "working fluid" - air in this case - is increased. (The diesel engine achieves its high efficiency principally because the short duration of its combustion processes allows very high gas temperatures to be used. Its specific power is low mainly because the combustion occupies so short a proportion of each cycle.)

Under the intense stimulus of the rewards bestowed on aircraft turbine engines by higher gas (air) temperatures at turbine inlet, extraordinary developments first in metallurgy and then in methods of cooling turbine blades have led to temperatures only a little below 3000 F (1650 C) to be in current use (figure 1). The thermodynamics of the cycle require that, for the full advantages of the higher temperatures to be obtained, the compressor pressure ratio must also be substantially increased (ref. 1). Modern jet engines have compressor pressure ratios between 20:1 and 40:1.

These high pressure ratios have in turn led to extraordinary developments. Compressors of high pressure ratio are extremely temperamental, and it has taken Herculean efforts to provide them with narrow but adequate working ranges of acceptable efficiency by either splitting them into several low-pressure-ratio compressors driven by separate turbines through complex concentric shaft arrangements, or by equally complex systems whereby about half the stationary blades of an engine compressor are pivoted and moved through precise angles at different points in the operating envelope. A very large proportion of the huge expense necessary to develop new aircraft engines goes to the cost of producing an acceptable high-pressure-ratio compressor.

To a large extent, aircraft-engine developments have dominated much of the commercial gas-turbine field. Many industrial gas-turbine engines are, in fact, simply jet engines in which the exhausts pass through large shaft-power turbines in place of the normal propelling nozzles. However, low-power gas-turbine engines have had to take a different approach. Their small physical size has made it impossible, for rather abstruse but definite fluid-mechanical reasons, to design compressors of high pressure ratio, and it is also impracticable to produce small turbine blades having tortuous cooling passages.

Designers of small turbine engines for, principally, automotive uses were forced to take a different approach: the heat-exchanger or "regenerative" cycle. In this a low-pressure-ratio compressor passes its flow through one "side" of a heat exchanger, through the other side of which flow the hot gases from the turbine exhaust (figure 2). The fuel flow required in the combustion chamber to produce the design turbine-inlet temperature can then be reduced, and an acceptable thermal efficiency can be obtained.

We are proposing a rather minor extrapolation of the heat-exchanger cycle that simply makes a virtue of the necessity of using both the heat exchanger and the low pressure ratio. Instead of aiming for somewhere near the maximum pressure ratio easily obtainable in small engines, an approach often used in the past, we are choosing to design for the maximum feasible heat-exchanger effectiveness coupled with a pressure ratio that will give the optimum set of characteristics for the engine duty specified. It has turned out that this approach has serendipitously brought several unsuspected advantages.

THE LOW-PRESSURE-RATIO CYCLE

The specification of the maximum feasible heat-exchanger effectiveness requires engineering and economic judgement. In the first place, the only feasible heat exchanger for a high-temperature low-pressure-ratio cycle is a ceramic rotary regenerator (sometimes called a "heat wheel" in air-conditioning applications) (figure 3). The reason is that the maximum possible inlet temperature for a metallic heat exchanger is currently below 760 C (1400 F). Turbine-inlet temperatures are currently over 930 C (1700 F) for uncooled turbines and up to 1540 C (2800 F) for cooled turbines (figure 1). The temperature drop through a low-pressure-ratio turbine expander may be as low as 250 degrees C (450 degrees F), so that metallic heat exchangers could be used only either for low-temperature (uncooled) gas turbines or for high-temperature turbines having a high pressure ratio. The size of engine that we have chosen as typical for fishing vessels in our study is 1120 kW (1500 hp), large enough to have cooled blades and thus to be categorized as potentially a high-temperature engine. If we opt, therefore, for a low-pressure-ratio cycle, the use of metallic heat exchangers must be ruled out.

In any case, ceramic heat exchangers have certain distinct advantages for this application, and this is where economic judgement must be used. The size of heat exchangers for a given duty is roughly proportional to the square of the passage (hydraulic) diameter. The passages in a rotary regenerator can be made far smaller than those in a metallic recuperator for two significant reasons. In a rotary regenerator (figure 3) the flow reverses every few seconds through every passage, tending to clear away accumulated particles. It has also become practicable to extrude regenerator cores having passages of extremely small hydraulic diameter without a large increase in cost per unit surface area (possibly, in fact, a decrease in unit cost). These are advantages over metal recuperators in which the flow is unidirectional, and small passages tend to become clogged, and in which the unit cost increases as passages are made small. The overall size of rotary regenerators of given effectiveness can be a small fraction of the size of a metallic recuperator in consequence of this ability to form and to use small passages.

A disadvantage of rotary regenerators over recuperators is leakage, which inevitably flows past the wiping seals around the four ducts that lead the two flows to and from the faces of the disk, and which also occurs through the carrying of the trapped gas in the matrix passages into the opposing stream. The effect of both of these leakages is reduced by reduced pressure ratio. A limiting pressure ratio of 6:1 is generally applied to rotary regenerators (ref. 2); the cycles we are recommending are well below this limit.

Therefore a strong case can be made for the specification of ceramic rotary regenerators. Judgement is required in the specification of the effectiveness - a characteristic that can be roughly translated to "heat-transfer efficiency". Maximum effectivenesses for gas-turbine heat exchangers has risen rapidly since the early problems with rotary regenerators have been solved, as shown in figure 4. The highest effectiveness heat exchanger we know of in a gas turbine is in the Allison GT 404, at just over 0.95. This engine, designed in the early 1970s (ref. 3), uses the common arrangement of twin ceramic disks taking the compressor and turbine flows split evenly between them, disposed on opposite sides of the shaft with their common axis of rotation intersecting the main turbine axis at right angles (figure 5). They are of moderate size. To increase the effectiveness to 0.975 - halving the thermal losses - could be done by doubling the thickness of the ceramic disks, which would keep the size within reasonable bounds and would improve the flow distribution to give further gains. We have therefore specified this value of heat-exchanger effectiveness in the engines we are proposing. It could well be economically justified to propose even higher levels of effectiveness - which would entail even lower optimum pressure ratios for the gas-turbine cycle - but our judgement is that we should proceed with some caution.

Advantages of low pressure ratios in gas-turbine engines

The combination of low compressor pressure ratio with high heat-exchanger effectiveness has some particular advantages for the type of high-efficiency engines needed for marine propulsion. The design-point thermal efficiency will be higher. The maximum possible thermal efficiency of a heat engine is set by the thermodynamic Carnot limit, which is $(1 - 1/T')$, where T' is the ratio of the (absolute) turbine-inlet temperature to the compressor-inlet temperature. For marine engines in the late 1980s, T' will be between its present value of about 5 to a future value which should be reached with ceramic turbine blades of at least 6. The Carnot efficiency limit is therefore between 0.80 and 0.83. The closeness with which actual gas turbines approach the thermodynamic limit will depend on the sum of the component losses. Compressor losses (roughly taken as (1 - polytropic efficiency) decrease with decreasing pressure ratio (figure 6), but seem to reach a limiting low value of 0.06. Heat exchangers, on the other hand, seem to have no theoretical lower limit of losses. In practice, the sum of the thermal and pressure-drop losses can be considerably less than 0.06, and far less than compressor losses would be for a high-pressure-ratio no-heat-exchanger cycle.

Heat exchangers also have efficiency advantages over compressors, especially those of high pressure ratio, at part load. Heat exchangers actually improve their performance at part load, while compressors deteriorate markedly (figure 7). We are working to substantiate these estimated variations with analytically derived characteristics, but there is no doubt as to the trend. We have shown one calculated curve of part-load efficiency (ref. 3) for a converted, and therefore compromised, helicopter engine, and have estimated a performance curve for a "clean-sheet" engine design in figure 8.

A major uncertainty with regard to future high-efficiency gas turbines at present is the possibility of using nonmetallic materials. The chief candidates are various forms of ceramics, all being extremely brittle, and tending to fail after a time of successful use in ways as yet imperfectly understood; and graphite fibers in a graphite matrix (called "carbon-carbon") which is a tough composite material actually increasing in strength up to at least 3500 F (2200 K), but requiring the absolute exclusion of oxygen by means of coatings. The ceramic materials seem closer to realization, but have experienced failures when incorporated into engines having very highly loaded turbines with high peripheral speeds and consequent high centrifugal stresses. The LPR cycle we are proposing here uses lightly loaded turbines of low peripheral speeds and low centrifugal stresses, and accordingly might be an ideal candidate for the introduction of ceramic blades, possibly strengthened with graphite fibers.

Being able to use ceramic or other nonmetallic blades in a gas-turbine engine could increase the efficiency at full- and part-load for three reasons.

1. There would be need for only a small amount of cooling air bled from the compressor discharge (figure 9). The compressor work required to provide this cooling air has a damaging effect on engine thermal efficiency, and therefore the smaller the quantity required the better the performance.

2. Turbine blade shapes would not need to be compromised to provide for internal cooling passages, and the discharge of cooling air would be eliminated, thus decreasing the fluid-mechanic losses in two ways.

3. Because ceramic (or other nonmetallic) blades are expected eventually to be relatively inexpensive to produce, there would no longer be a cost reason to keep the number of turbine stages to a minimum. The need to minimize the number of blades to be cooled by expensive compressed air would also disappear. Accordingly, in most nonaircraft situations it would be found to be economically optimum to choose to have the number of turbine stages above the minimum, and thus to gain the benefits of higher turbine efficiency that come with lower loading (figure 10).

The higher turbine efficiency that would result would have a snowball effect, because it would result in an optimum pressure ratio that would be lower than before, other things being equal. This lower pressure ratio would in turn require a compressor and turbine of lower loading and therefore higher efficiency, further lowering the optimum pressure ratio.

Using nonmetallic materials in place of alloys high in chromium and cobalt would have significant strategic advantages. There may, in addition, be reduced material attack from fuel constituents and from aerosols in the airflow in marine atmospheres.

The "LPR" approach increases the size of the turbomachinery, making it unsuitable for high-speed aircraft. For marine and other uses, the turbomachinery is still small in comparison with the size of alternative engines, as will be seen below. The shaft speed is considerably reduced compared with high-pressure-ratio gas turbines, which is an advantage.

PRELIMINARY DESIGN OF* A* BASELINE ENGINE

Our preliminary "clean-sheet" design is not yet completed, but we can give some details with fair confidence. We chose as our "baseline" fishing vessel a 36-meter (119-ft) stern trawler with a propulsion engine rated at 1.12 MW (1500 hp) on the recommendation of Jack Gilbert (ref. 5).

We chose 1555 K (2330 F) for the turbine-inlet temperature. This is typical of current naval gas turbines using metal blades (ref. 6) and would therefore allow the design to be produced with either metallic or nonmetallic blading. The temperature ratio T' is, with an ambient (compressor-inlet) temperature of 288 K (58 F), 5.4. Using a rotary regenerator of effectiveness 0.975 the thermal efficiency was found to be a slowly varying function of pressure ratio (figure 11), and we chose a value of 3:1.

With two alternative values of cooling-air requirements, a higher one for conventional metallic blades based on NASA work (ref. 7), and a lower proportion assumed for nonmetallic blades, we found that the metallic-bladed engine should have a gross thermal efficiency of 57 percent, and an engine with nonmetallic blades should have an efficiency two points higher. The net efficiency, by which we mean the brake thermal efficiency at the shaft when all the deductions for bearing friction, windage, fuel and oil pumps and control drives have been made, should be no more than two points lower. These translate to specific-fuel-consumption values of 0.243 - 0.252 lbm per bhp-hr.

The preliminary design of an engine to give 1500 hp (1.12 MW) shows that an axial compressor attaining a pressure ratio of 3:1 with a mass flow of 5.5 kg/s would have six stages with a rotor-blade-tip diameter of 300 mm (11.8 in) and an overall length for the compressor of under 500 mm (20 in) including the diffuser. With this design the blade speeds would be low enough for the compressor blading, and perhaps the disks and casings, to be made in fairly commonplace reinforced molding resins. Three such materials with outstanding high-temperature fatigue and creep resistance are polyphenylene sulfide (PPS), polyetheretherketone (PEEK) and polyethersulfone (PES), which could be reinforced with glass or carbon fibers (ref. 8). These would also convey excellent moisture and salt-spray resistance.

The axial-flow turbine required for this engine would have three stages (six rows of blades) with an outside diameter of about 450 mm (17.7 in). The shaft speed would be about 16,700 rev/min, giving very low turbine-blade stresses compared with conventional design, and therefore providing favorable conditions for the application of nonmetallic blades. A two-stage epicyclic reduction would probably be used if the engine is directly coupled to a controllable-reversible-pitch propeller.

The turbine exhaust would pass into a ceramic regenerator. The usual arrangement for the small engines so far equipped with this type of heat exchanger is to use two ceramic disks, one on each side of the turbine (figure 5). If this scheme were used for the 1500-hp engine, the disks would be 1.75 m (69 in) diameter and 136 mm (5.4 in) thick. Both the diameter and the thickness are apparently beyond the range of current production technology, at least in the US, but could possibly be manufactured overseas by extrusion or by building up a large disk from smaller sections. If the present US limits were to be adhered to, 12 double-thickness disks 0.71 mm (28 in) diameter and 76 mm (3 in) thick would be used. This arrangement would complicate the ducting but lower the production cost because a standard regenerator could be used. Each disk pair would be independently driven by a fractional-horsepower electric motor through a standard gear reduction and stainless-steel-belt drive. The cool exhaust gases would leave the opposite faces of the disks and be ducted up the stack, perhaps giving up further heat to a waste-heat boiler and/or an absorption chiller/freezer on the way.

The compressed air leaving the compressor would be ducted to sectors on the disks, pass through the matrix and be combined in the casing of a single combustor supplying the turbine. The combination of ducting, heat exchanger and combustor would probably be located above the turbomachinery line to allow easy access for servicing.

While all new machinery is customarily introduced with promises of very low maintenance requirements, promises not always borne out in practice, the gas turbine in several duties, including marine service with the US and Royal Navies (ref. 9) with highly rated aircaft-derivative units, has indeed required exceptionally low maintenance. In naval duty it is generally the practice to exchange whole engines when anything greater than minor maintenance is needed. The small size and low weight of turbine units make them fairly easy to remove and replace, for instance through the stack.

CONCLUSIONS

We have made a case that the low-pressure-ratio highly regenerated gas turbine has particular advantages for marine use. The low blade speeds required would enable nonmetallic materials to be used with advantage, although the virtues of the cycle are not dependent on the use of nonmetals. The design-point fuel consumption should be exceptionally good, and part-load consumption down to at least 25-percent power should be better than that of any competitor. Engines of this type could be produced today - indeed, it could be said that the industry is moving cautiously toward this type of design - but developments in nonmetallic materials, particularly in ceramics and ceramic-shielded graphite fibers, would, if initial good reports of the resistance of ceramic coatings to sulfidation attack are further confirmed, make the engine even more attractive for marine use.

A university group cannot do much more than point out advantages and disadvantages of different technologies. We are funded by public money to stimulate change. We hope that some engine manufacturers will study this apparently attractive engine and produce some version of it for US and overseas fishing fleets.

ACKNOWLEDGEMENTS

This project was funded in part by the US Department of Commerce, Office of Sea Grant. We have been given a great deal of help by an advisory committe with members from the fishing, marine-construction, and engine-manufacturing industries, and from academia. Our Sea-Grant contract monitors have given particular assistance. Our grateful thanks are due to all. Their names will be listed in the project final report, which should be available in August 1984 from the Office of Sea Grant, MIT room E38-302, CAMBRIDGE, MA 02139, or from the Department of Commerce in Washington, DC.

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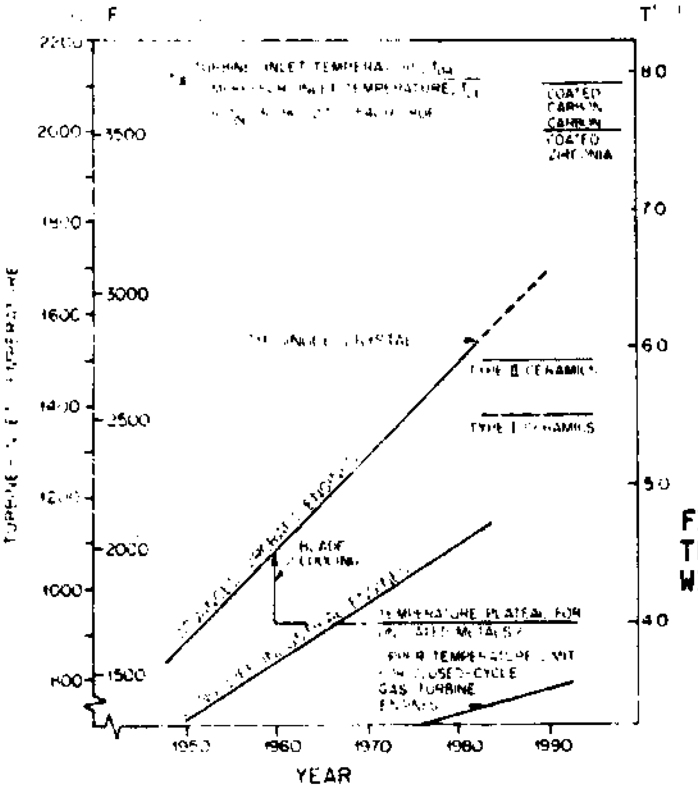

FIGURE 1 INCREASE OF TURBINE-INLET TEMPERATURE WITH TIME

from ref. 1

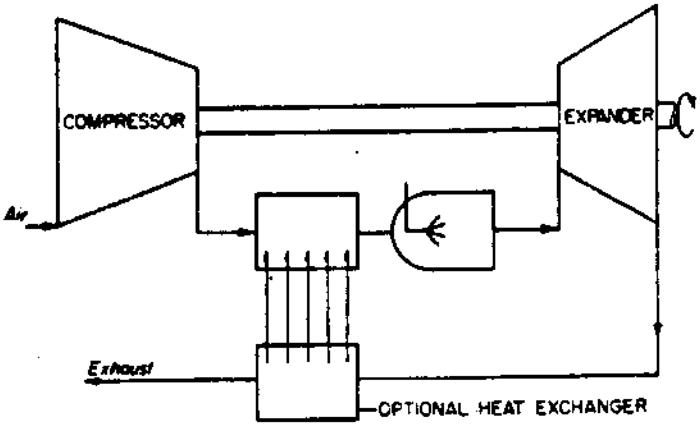

FIGURE 2 BLOCK DIAGRAM OF GAS-TURBINE ENGINE WITH HEAT EXCHANGER

from ref. 1

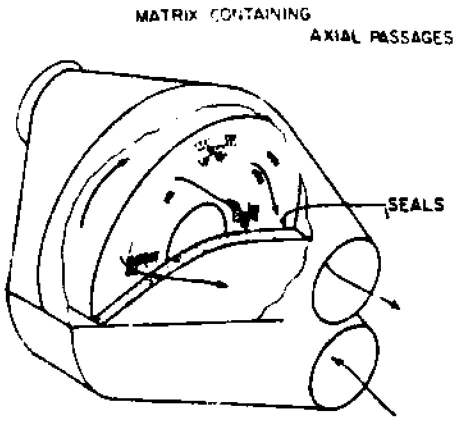

FIGURE 3 DISK-TYPE ROTARY REGENERATOR

from ref. 1

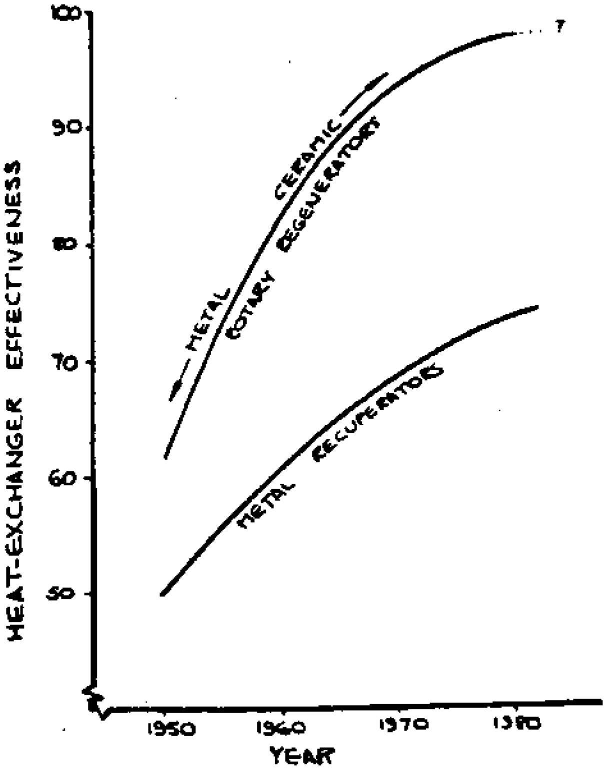

FIGURE 4 GROWTH OF PEAK EFFECTIVENESS USED IN GAS-TURBINE HEAT EXCHANGERS (ESTIMATED)

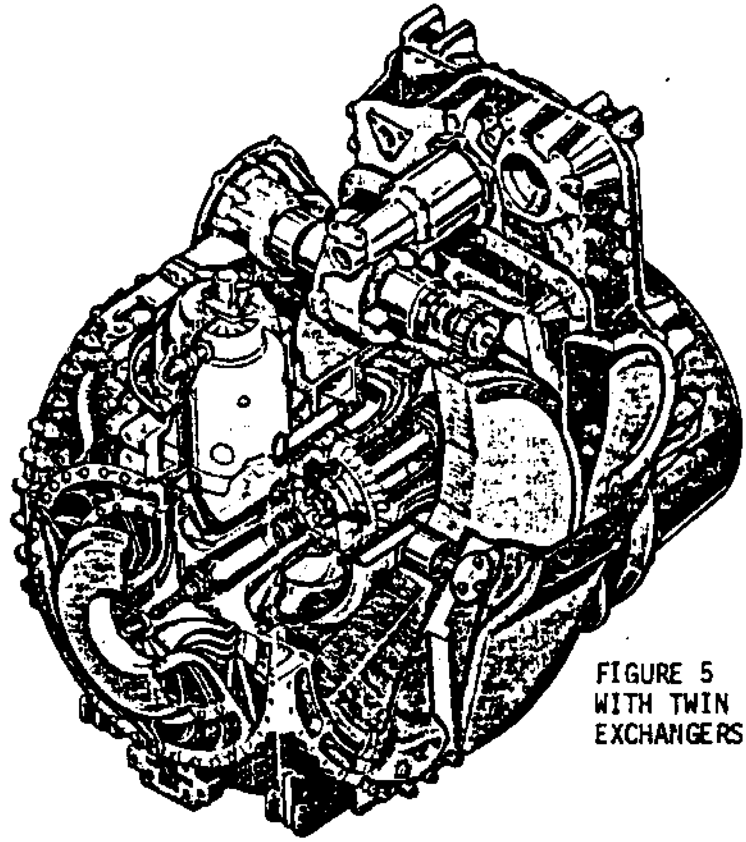

FIGURE 5 GAS-TURBINE ENGINE WITH TWIN REGENERATIVE HEAT EXCHANGERS (DDA GT 404)

from ref. 3

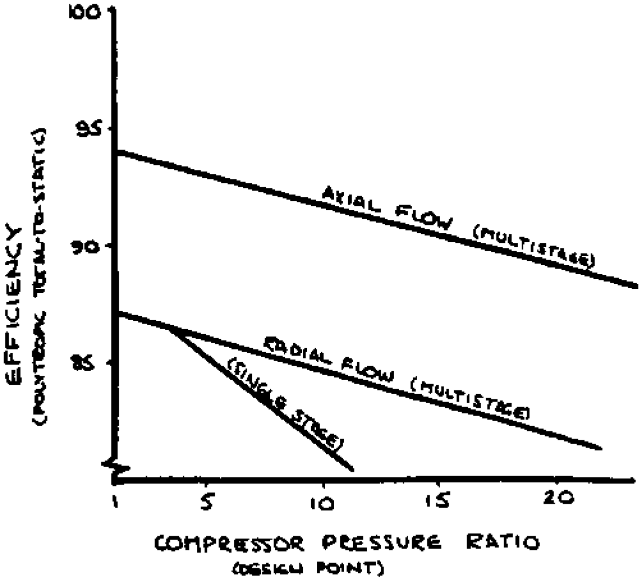

FIGURE 6 PEAK COMPRESSOR EFFICIENCIES
(ESTIMATED)

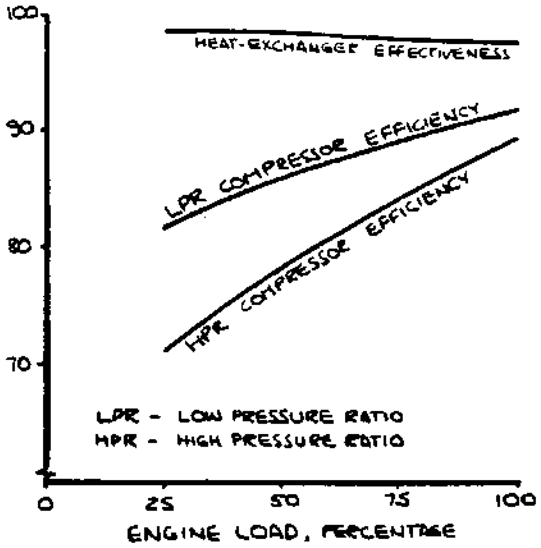

FIGURE 7 TYPICAL VARIATION OF COMPONENT EFFICIENCIES WITH ENGINE LOAD

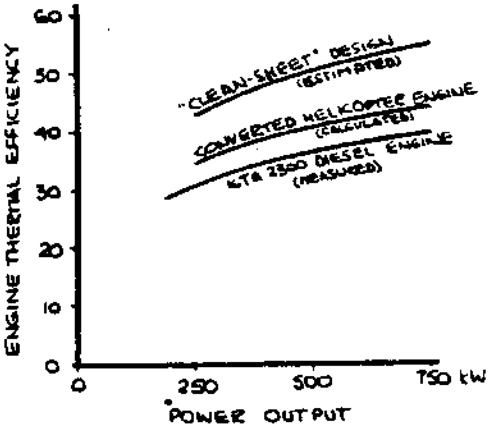

FIGURE 8 PART-LOAD PERFORMANCE OF ALTERNATIVE ENGINES

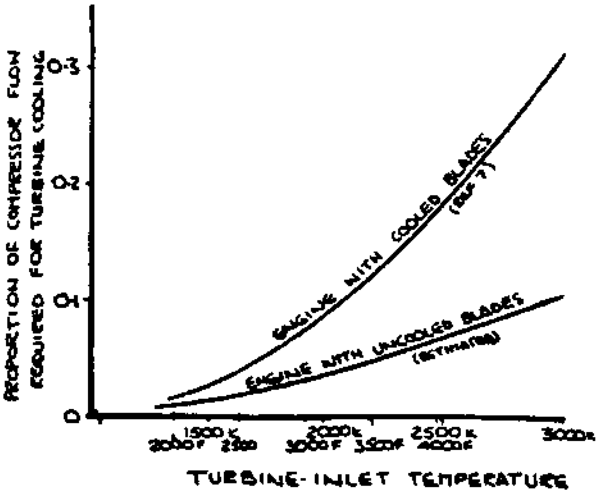

FIGURE 9 TURBINE COOLING-AIR REQUIREMENTS

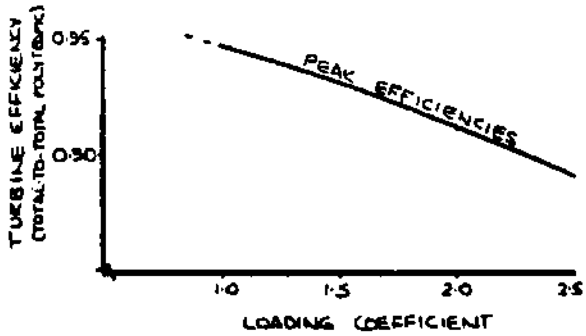

FIGURE 10 PEAK TURBINE EFFICIENCY VS AERODYNAMIC LOADING

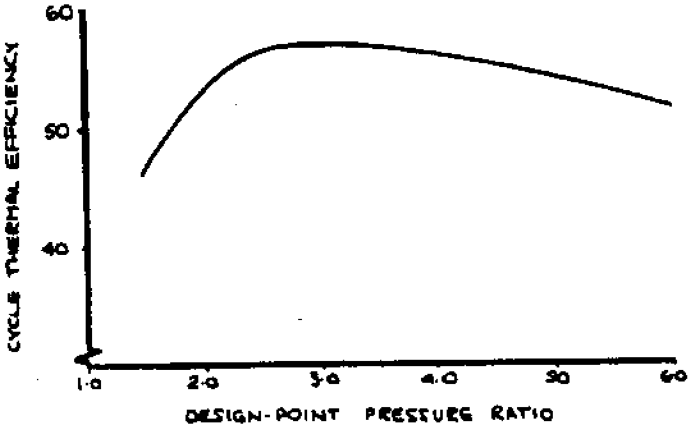

FIGURE 11 THERMAL EFFICIENCY VS DESIGN PRESSURE RATIO FOR THE "BASELINE" ENGINE

TRANSPORTATION EFFICIENCY DATA DEVELOPMENT USING A HIGH SENSITIVITY LORAN "C" RECEIVER OUTPUTTED TO A CPU BASED ENERGY CONVERSION INSTRUMENTATION AND INFORMATION REDUCTION SYSTEM

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(Editors' Note: We regret that due to the length of this paper (75 pages) it is not possible to present it in its entirety in these proceedings. Anyone wishing to review the 29 additional pages of the report along with the 37 pages of illustrations and data which had to be omitted may contact Dr. John Sainsbury, Department of Oceanography and Ocean Engineering, Florida Institute of Technology, Melbourne, FL 32901.)

ABSTRACT

Cargo transportation and bottom fishing are two basic types of operations where EFFICIENCY analysis based on speed over land is important. This information is usually integrated with data concerning energy utilization, such as gallons per hour of fossil fuel consumed, product data, variable costs concerning crew wages, vessel support, depreciation allowances and cost of lost opportunity. Manual methods of operating on "sampled" data is slow and inadequate for use in pseudocyclical markets wherein the market cycle is much greater than the cargo revenue period, company liquidity is low and the cost of money high. The method, therefore, must deal with the single cargo revenue period and multi-scenario projections. An iterative approach to speed made good toward destination and projections pertaining thereto fitted to a historical economic model, frequently updated, could be of prime importance, particularly since management goals and operator behavior MUST be tightly interlocked for servivability in a highly competitive market place with narrow profit margins. This paper explores some of the tools needed in reaching toward that ideal result.

The nature of the requisite sensor technology, data path and initial data reduction as developed in work with 5,000 to 12,000 hp. Inland Waterways tow boats is discussed. Early Fuel Management System equipment applications on fisheries research vessels are discussed, along with opportunities for further research utilizing the technology developed as a subject of this paper.

INTRODUCTION:

Fuel and Power Management Instrumentation (or PROPULSION MANAGEMENT SYSTEMS of varying classes depending on the degree of sensor complexity and vertical integration of information), as priority equipment on board commercial vessels, has grown significantly since 1978. Design life of the various measuring elements has been improving, as well as the scope of the measurement systems in their application to myriad types of vessels. Increases in fishing catches and reductions in operating costs with constant or improving revenue dollars has resulted in "word-of-mouth" publicity enlarging the sales of this type of equipment amongst fishermen. Fuel and power instrumentation is becomming increasingly important in the economic survival of marine commerce, as it gives a greater degree of precision by which the vessel operator may discriminate the relationship between what the vessel is doing vs. the financial goals that he is after.

The PROPULSION MANAGEMENT INSTRUMENT, known by product classes as "Fuel Management System", "Power Management System", "Performance Monitoring System" and other titles, is a clustering of instruments, read-out systems and data gathering systems. This process, to be a true management tool, must include a data reduction and aids in report writing, e.g., software.

PROPULSION MANAGEMENT SYSTEM instrumentation generally comprises several classes of sensors that input into calculating electronics. The inputing instrumentation deals with energy conversion, and hence deals with such simple elements as time, motion, volume, pressure and location. Propulsion management systems deal with the systematic gathering of various types of power conversion instrumentation, the presentation of that data and varying degrees of data reduction (supplier dependent).

The POWER MANAGEMENT SYSTEM used in starting this study was comprised of a three (3) step approach. This involved (1) instrumentation for use by the pilot and engineer, (2) data gathering, storage and its preliminary decoding and storage onto a magnetic "floppy" disc, and (3) the reduction of data for interpretation using a small desk top computer and software and "staged" software to facilitate modeling and projections development (A.D.R.E.S.tm).

Approximately 15 tow boat vessels (see photos 1, 2 and 3) have been equipped with instrumentation using a specially modified Trimble Navigation model 100 Loran "C", proximity type digital tachometers and M.S.E. recirculation type fuel flow measurement systems. The bulk of the vessels were equipped by:

American Performance Monitoring Systems
One Todd Plaza
Pass Christian, Mississippi 39571

(601) 452-7621

The equipment is rebranded MSE equipment, and A.P.M.S. leases these units to companies interested in fuel consumption research.

The companies could have A.P.M.S. and/or M.S.E. develop reports and software, or could option, as some did, purchase the equipment and do their analysis in-house (these companies had "main frame" and staff time available). The vast majority of the customer vessels are 6,000 to 12,000 horsepower tow boats operating on the Mississippi, Missouri, Ohio and Tennesse rivers, the Gulf of Mexico and the Caribbean.

The Fuel Management System computer (NAVSCANtm), Tachometer and Fuel sensors, Magnetic Tape data storage device, Loran "C" Data Discrimination Card and A.D.R.E.S.tm (Automated Data Retrieval and Evaluation System) are products of Marshall Scott Electronics.

Two important drawbacks had to be addressed at the start of the program:

1) Since the introduction of Loran "C" receivers with a land speed measuring capability, many tow boat operators have experimented with Loran units for speed over bottom measurements. Due to lofty claims by poorly informed sales people, the receivers were misapplied with disasterous results on the inland waterways. By word of mouth, everyone "knew" that Loran "C" couldn't be used for land location and speed information on the inland waterways. This was due to the difference in radio wave propogation speed over land and over water. "None" could produce accurate coordinates and speed data with over land measurements. It was up to us to develop a systematic approach and prove otherwise.

2) Instrumentation reliability was a must. Common ship board instrumentation, such as D.C. or "turns counter" digital tachometers were not reliable or accurate enough for baseline and ongoing measurements. On many work boats, M.T.B.F. (Mean Time Before Failure) for tachometer drives was less than 6 months, and had interpretation as well as absolute accuracy problems. "Water Meter" P.D. and turbine fuel flow sensors could give useable short term results when properly calibrated, but were uncertain over time due to measurement drift and sticking.

Although the above seem simple to overcome at first glance, much time and experimentation was required to put together a working system and shake-out the result sufficiently to begin a program and demonstrate the results. The vessel operators must be convinced, then the port engineer, and finally, management.

Of particular importance, in dealing with work boats, is that they are almost always moving cargo over long distances, and significant effort must be expended in developing, installing, servicing and gathering the data from the equipment. The boat is rarely where it is suppose to be at the time intended, and service must be accordingly planned. Usually, the work must be conducted while the vessel is moving, with great risk involved in getting on and off the boat with the powerful engines driving against the current. Getting off the boat would be at the convenience of the Captain, with his cargo and crew considerations, and finding a location at were some form of surface transportation is waiting is difficult (usually at fleeting and fueling locations near major river cities). Recent advances in Loran "C" technology employing "overland correction" (Millington's method) has produced automatic electronically corrected Latitude/ Longitude and Land Speed information adequate for both use over developed countries' coastal waters and Inland Waterways in the United States. It has advanced the Loran "C", in our research, as the most promising candidate for low cost iterative speed measurement (approx. 1/6 to 1/10th the cost of Doppler Shift speed log with greater position information) for incorporation into a computer system. This computer system, still under evolution, is capable of automated data descrimination, interpolation and integration with other power variables. Because of river water channel width and depth effects, knowing where the vessel was when the data was gathered has become as important to vessel managers as knowing how fast the vessel is going is to the vessel pilot. The data must be later statistically separated and reduced.

By having an understanding of the vessels' fuel consumption relative to engine r.p.m., knowing the vessels land speed, its location, time and the make up of the tow, a generalized mathematical relationship can be developed which, through operations research, can predict cost savings with different vessel operating scenarios. Unfortunately, the accuracy of the underlying equation is dependent on the amount of data to be explained, with confidence growing with the size and ordering of the data base.

M.S.E. is currently building an iterative approach to economic model development. Near term results of this effort will be the subject of later papers as authorized by the researching companies.

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3) "Phase of the Low Radio Frequency Ground Wave", J.R. Johler, W.J. Keller and L.C. Weltus (U.S. National Bureau of Standards Circular 73, 1956)

4) "Ground Wave Propogation Over an Inhomagenous Smooth Earth", G. Millington, J.R. Johler, J. Ralph and R.H. Doherty (Proc. Inst. Elect. Engrs., Pt. 3.96, Part 3, Vol. 96, p. 53, Jan. 1949)

5) "Ground Wave Propogation Over an Inhomagenous Smooth Earth" Part 2, Experimental Evidence and Practical Implementations. G. Millington (Proc. I.E.E., Part II, Vol. 97, p.209, July 1950)

6) "Optimization of Underway Efficiency Through Marine Power Conversion Instrumentation Placed at Critical Energy Transformation Points", H.W. Haynes et al (Collection published papers, Marshall Scott Electronics Corp., Seattle, Wa., August 1982)

7) "Low Cost Microprocessor Based Power Conversion Instrumentation Applicable to Sail Assisted Commercial Fishing Vessels", H.W. Haynes et al (International Symposium on Sail Assisted Fishing Vessels, Tarpon Springs, Fla., 18 May, 1983, proceedings pp. 311-361)

8) "TRIMBLE NAVIGATION on the Issues of Accuracy and Range", Brochure (Trimble Navigation, Mountain View, Ca. 94043, (415) 962-9893, (800) TRIMBLE, 1984)

9) "The World's Most Accurate and Comprehensive LORAN Computer", Brochure on Model 200 (Ibid, Trimble Nav.)

10) "Comprehensive Navigation and Performance Features of the Model 200 Computer", Brochure on Model 200 (Ibid, Trimble Nav.)

WRITTEN CONTRIBUTION

N. T. Riley

The following may be of interest in establishing a lower end point to Mr. Wilson's data:

We were required to design a trawler for the Gulf of Carpenteria prawn fishery in Northern Australia, but because of crewing regulations and government subsidery constraints the L.O.A. and hence design L.W.L. was restructed to 60.6 ft.

The relevant details of the vessel were as follows:

L.W.L.	60.6	ft.
Moulded beam	23	ft.
Mean draft	8	ft.
Displacement	150	tons
L/B	2.63	
B/d	3.20	
$/(0-01L)^3$	674	

The vessel was of double chine construction. When designing the lines we become concerned about the directional stability characteristics principally because of the high displacement length ratio, the low L/B ratio and also we had no comparative data for a vessel of these extreme proportions.

To ensure that we had no problems in this area we paid special attention to the design of the shape in the aft end and in particular the shape of the buttocks in the run.

In particular we made sure that the angle of the mean buttock did not exceed 15 degrees to the horizontal, the immersion of the transom was moderate and the skeg was carried well aft.

When trials were carried out the directional stability was found to be without problems.

WRITTEN CONTRIBUTION

Cliff Goudey

Mr. Tupper identifies the deterioration of trawl gear performance as the most frequent negative effect of vessel motions. While some improvements in vessel sea-keeping can be achieved through the use of bilge keels, steading sails, or paravanes, conventional hulls will always be hampered by adverse sea-states.

One method of maintaining trawl performance would be to isolate the effects of heave, pitch and surge from the trawl. Through the use of some form of constant tension system or a passive cable accumulator the trawl doors and net would experience a far more uniform pull.

PART IV

FISHING VESSEL SAFETY

ANALYSIS OF U.S. COMMERCIAL FISHING VESSEL LOSSES 1970-1982

LCDR TONY E. HART
U.S. COAST GUARD

MR. FRANK PERRINI
U.S. COAST GUARD

Abstract

From the viewpoint of the U.S. Coast Guard commercial vessel safety program, the U.S. fishing fleet is essentially unregulated. The one notable exception is the requirement similar to requirements imposed on all vessels to carry certain lifesaving and firefighting equipment onboard.

The safety record of the commercial fishing fleet leaves much to be desired. Analysis indicates that the fatality rate of commercial fishermen, considering deaths associated with a casualty and accidental deaths, is approximately 7 times that of the overall U.S. industry average, and although the loss rate of vessels showed some improvement during the 1970's, the rate began to increase again in 1981 and 1982.

Scope

This analysis examined casualties suffered by the U.S. commercial fishing fleet from 1970 through 1982. The analysis concentrated on casualties which involved the total loss of a fishing vessel documented by the Coast Guard, and all loss of life on documented fishing vessels. As additional information on the magnitude and nature of casualties being incurred by the fishing fleet, the number of vessels reporting damages are shown in Appendix A to this analysis.

Casualties involving state-numbered boats used in commercial fishing are not considered in the analysis. It is difficult to distinguish from casualty data those state-numbered boats used in commercial fishing from those used for recreational purposes. Further, there is no good measure of the total state-numbered boats used in commercial fishing. Also, a casualty incurred by one of these boats, which is often nothing more than a small, outboard-powered open craft, is difficult to compare with the foundering, capsizing or grounding of a larger, documented vessel.

Casualty data for this analysis was obtained from the Coast Guard vessel casualty file at Coast Guard Headquarters. This file has been compiled from reports of casualties and personnel accidents submitted to the Coast Guard by vessel owners and operators. These reports are required by 46 Code of Federal Regulations Part 4. Fishing vessel population data were obtained from Coast Guard vessel documentation files. Other sources of information regarding fishing activities included National Marine Fisheries Service (NMFS) annual publications, "Fisheries of the United States" and "Fisheries Statistics of the United States."

Analysis - Vessel Losses

It is generally acknowledged that commercial fishermen are engaged in one of the more hazardous occupations. This perception is reinforced by this examination of casualties which have been reported to the Coast Guard and which are maintained in the computerized vessel casualty file at Coast Guard Headquarters. This examination of casualties for the period, 1970-1982 has disclosed that the U.S. commercial fishing fleet incurred vessel loss rates on the order of 5 times that of U.S. ocean-going cargo ships and 3 times that of U.S. ocean-going tankships.

Fishing vessel losses for the period 1970-1982 are shown in Table 1, broken down by the nature of the casualty incurred. One can see that the single greatest contributer to losses is the category: foundering, flooding and capsizing. Fires and explosions are the next leading category, and within the past three years the total number of vessels lost to fire/explosion has increased. Throughout most of the period examined, the total number of vessels lost generally fell between 150-200. However, during 1981 and 1982, the number of vessels lost increased substantially.

TABLE 1: U.S. DOCUMENTED FISHING VESSELS TOTALLY LOST
BREAKDOWN BY NATURE OF CASUALTY

	1970	1971	1972	1973	1974	1975	1976	1977	1978	1979	1980	1981	1982
COLLISION	41	42	20	24	23	16	17	20	7	17	13	20	23
FIRES & EXPLOSION	43	41	25	31	34	25	35	26	14	35	52	57	74
GROUNDING	52	44	27	47	36	23	27	26	16	37	32	27	31
FOUNDER FLOODING CAPSIZE	19	8	13	18	79	103	92	72	51	112	95	125	119
HEAVY WX	6	0	2	3	0	2	1	0	0	0	0	0	3
MATERIAL FAILURE	49	42	54	47	12	0	5	7	0	7	4	21	20
OTHER	5	4	5	1	4	0	1	0	1	1	1	0	0

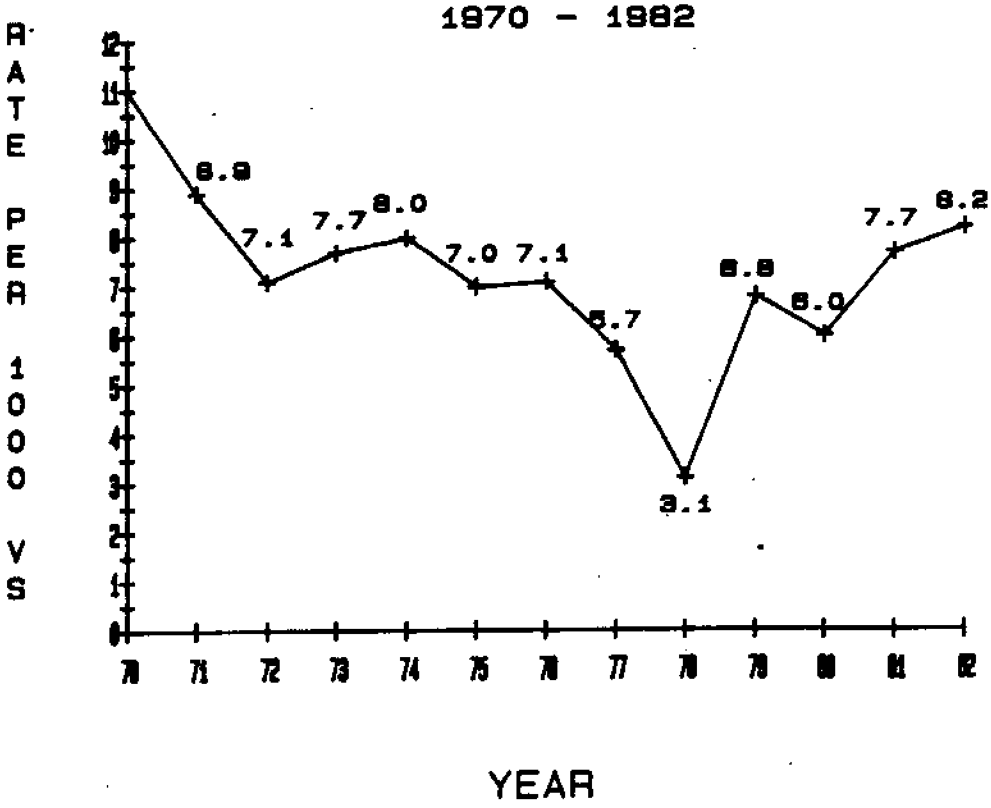
NOTE: Between 1970 and 1973 many of founderings, floodings and capsizings were coded as material failures.

Looking only at a change in the number of casualties without consideration of any changes in the number of vessels at risk may not present a complete picture of the fleet's safety record. While the number of annual fishing vessel losses remained relatively constant between 1970 and 1980, the fishing fleet itself experienced substantial growth. According to the Coast Guard Vessel Documentation File at Coast Guard Headquarters, the U.S. fleet numbered about 19,500 documented fishing vessels in 1970. By 1982, this had grown to approximately 33,000 vessels. Using vessel population as a basis for expressing vessel loss rates, the casualty rate for fishing vessels declined from 11 vessels lost per 1000 vessels in 1970 to 6.0 vessels lost per 1000 vessels in 1980. This improvement did not continue in 1981 and 1982, as the casualty rate increased to 7.7 vessels per 1000 in 1981 and up to 8.2 vessels in 1982. Table 2 shows the annual casualty rates for the period examined.

TABLE 2: U.S. DOCUMENTED FISHING VESSEL LOSSES - 1970-1982

Year	Number of Vessels Lost	Vessel Population (per 1000 vessels)	Casualty Rate
1970	215	19589	11.0
1971	181	20263	8.9
1972	146	20623	7.1
1973	171	22140	7.7
1974	188	23442	8.0
1975	169	24150	7.0
1976	178	24965	7.1
1977	151	26374	5.7
1978	89	28434	3.1
1979	209	30567	6.8
1980	197	32658	6.0
1981	250	32800 (est.)	7.7
1982	270	33000 (est.)	8.2

FIGURE 1: Loss Rate for Documented F/V
1970 - 1982



The analysis also examined losses in relation to certain vessel characteristics to determine what effect, if any, they could be having relative to the number of losses suffered. Characteristics chosen for the study included hull material, vessel length and vessel age.

In 1970, the initial point of the study, 86.8% of the fleet were wooden vessels with steel and fiberglass accounting for an additional 11.0% and 1.9% respectively. During the next twelve years, this composition changed considerably as the number of fiberglass vessels increased from 1.9% to approximately 22% by 1982. At the same time, wooden hulled vessels declined to approximately 60% of the total while steel hulled vessels increased slightly to about 16%. Wood, steel and fiberglass hulled vessels now make up approximately 98% of the commercial fishing fleet.

Loss rates for wood, steel and fiberglass hulled vessels are shown in Table 3. With the exception of the last two years, wood and steel vessels experienced approximately the same loss rates. Throughout the entire period examined the loss rate for fiberglass vessels has been substantially lower than those vessels constructed of wood or steel. However, this loss rate may be somewhat influenced because fiberglass vessels generally tend to be concentrated more toward the smaller segment of the fleet and therefore may not venture as far out to sea or carry as much topside equipment as wooden or steel vessels.

TABLE 3: U.S. DOCUMENTED FISHING VESSELS TOTALLY LOST CASUALTY RATES PER 1000 VESSELS BY HULL TYPE

	Wooden Hull Vessels	Steel Hull Vessels	Fiberglass Vessels
1970	11.6	6.5	0.0
1971	9.1	6.8	0.0
1972	7.1	7.4	0.0
1973	7.5	9.2	4.3
1974	8.4	6.8	3.6
1975	7.5	7.5	1.9
1976	7.3	7.0	4.0
1977	6.0	5.8	3.7
1978	3.2	4.0	1.6
1979	7.9	7.5	2.0
1980	5.8	6.1	2.9
1981	7.7	10.6	5.0
1982	8.4	10.5	5.8

Age of the vessel also appears to have an effect on the loss rate. As shown in Table 4, as age increases so does the loss rate, at least until some point at which the loss rate begins to decline. It is interesting to note that a similar trend; i.e., an increase in loss rate and then a decrease, has been observed for other types of merchant vessels. It is unclear why the rate begins to decrease at some point but one possibility includes a decrease in use of the "older" vessel which would decrease its exposure to risk. It is encouraging to note that, as a whole, the overall age of the fleet has decreased between 1970 and 1982. In 1970, 54% of the fleet was 20 years old or older. By 1982, the percentage of the fleet in this age group had decreased to 41%.

TABLE 4: U.S. DOCUMENTED FISHING VESSELS TOTALLY LOST CASUALTY RATES PER 1000 VESSELS

	Vessel less than 5 years old	Vessel 5 to less than 10 years	Vessel 10 to less than 15 years	Vessel 15 to less than 20 years	Vessel 20 and greater
1970	5.5	8.1	17.6	20.9	10.0
1975	5.1	3.6	6.6	9.2	8.2
1980	5.1	5.0	5.2	9.2	6.2
1982	6.0	7.2	8.0	7.8	6.8

Table 5 presents loss rates based on vessel length broken down into 3 separate ranges of vessel size. These ranges were dictated by constraints present in the casualty file. Until 1981, vessel length, and certain other vessel characteristics were entered into various coding groups in the casualty file. It is now possible starting with 1981 casualty data to extract the exact length of vessels involved in casualties.

Vessels less than 65 feet in length experienced the lowest loss rate during the entire time frame examined. One particularly distressing note is the tremendous increase in the loss rate during 1982 of vessels greater than 100 feet long. Between 1970 and 1981 an average of 6.75 vessels greater than 100 feet were lost. In 1982, 13 vessels in this size range were lost. This increase in the number of larger vessels lost also is depicted by casualty data published by Lloyd's Register of Shipping. Lloyd's compiles and publishes losses of self-propelled vessels of 100 gross tons and above. According to Lloyd's, 24 vessels greater than 100 gross tons were lost in 1982 compared to 14 vessels of that size during 1981, a 71% increase.

TABLE 5: U.S. DOCUMENTED FISHING VESSELS TOTALLY LOST CASUALTY RATES PER 1000 VESSELS

	Vessels 65 feet and under	Vessels greater than 65 feet and less than 100 feet	Vessels greater than 100 feet
1970	9.7	19.6	14.0
1971	7.3	17.8	14.3
1972	5.8	12.9	19.7
1973	6.1	18.8	5.6
1974	7.0	13.3	16.2
1975	6.3	11.8	9.0
1976	6.1	13.6	12.6
1977	5.2	8.1	8.1
1978	2.6	5.1	10.2
1979	6.2	10.4	9.6
1980	5.0	10.6	14.6
1981	6.4	15.7	7.7
1982	6.7	16.0	19.8

Included in the analysis was an effort to determine how the U.S. fishing fleet's safety record compares with foreign fleets. Exact comparisons are not possible. Each country's fleet is different in vessel characteristics, the fisheries engaged in, geographical areas and environmental conditions in which the fleet operates, etc. Additionally, in some countries, the fishing fleet is more highly regulated than in others. Because there are no standardized, internationally used casualty reporting requirements or casualty publication formats, each country's casualty data is different in content and format which further compounds the problem of comparisons. Comparisons made to date are still preliminary, but indications point toward common problems worldwide with fishing vessel safety.

Table 6A presents loss rates for United Kingdom fishing vessels between 1975 and 1982 alongside the U.S. rates for the same period. Notice that the rates correspond very closely.

TABLE 6A:

COMPARISON OF UNITED KINGDOM & UNITED STATES FISHING VESSEL LOSSES 1975-1982

	United Kingdom			United States
Year	Population	Losses	Rate (per 1000 vessels)	Loss Rate (per 1000 ves)
1975	6691	47	7.0	7.0
1976	6740	35	5.2	7.1
1977	6953	37	5.6	5.7
1978	7067	38	5.1	3.1
1979	7242	42	5.8	6.8
1980	6895	40	5.8	6.0
1981	7351	52	7.1	7.6
1982	6797	50	7.4	8.2

Note: Population data and losses for United Kingdom vessels obtained from the annual Casualties to Vessels and Accidents to Men, published by the Department of Trade, London, England.

One source for obtaining worldwide casualty information is Lloyd's Register of Shipping. For examining fishing vessel losses, the primary constraint with using data published in the Lloyd's Casualty Returns is that the minimum size of vessel included is 100 gross tons. This excludes the majority of fishing vessels in our fleet as well as most other countries. However, using casualty data from the quarterly Lloyd's Register of Shipping Casualty Returns, Table 6B has been prepared to depict the average loss rates for fishing vessels of selected countries during the period 1979-1982.

TABLE 6B: LOSS RATES FOR FISHING VESSELS OF 100 GROSS TONS AND ABOVE 1979-1982

Country	Average Loss Rate during 1979-1982 (per 1000 vessels)
Canada	6.4
Taiwan	30.3
Japan	3.0
South Korea	18.7
Norway	5.1
Spain	5.1
United Kingdom	3.1
United States	7.8

Note: Total losses taken from Lloyd's Casualty Returns. Vessel population taken from Table 13 of the annual Lloyd's Statistical Tables.

Cause Of Casualties

In the analysis of vessel losses it is often difficult to identify and isolate the specific cause of the casualty. However, an analysis of the causes of the casualties suggests that many could have been prevented or the severity of the incident diminished had a few precautions been taken. The installation of fire alarms in the engine spaces would have in many cases alerted the crew to a fire in its early stages. Similarily, a bilge alarm would have alerted the crew to the early ingress of water into the vessel. Generally, causes can fall into three broad categories: human failure, vessel related, or environmental, but very seldom is the cause the result of a single failure. Instead, the cause is usually the result of a chain of events which culminate in an accident.

Among the various causes noted in reviewing casualties, human failure stands out. These failures include:

a. Poor watchkeeping practices.
b. Navigational errors and rules of the road violations.
c. Lack of understanding of the various forces acting on the vessel, especially as concerns the stability; i.e., failure to load and operate the vessel according to its stability chart, modification of the vessel without consideration of possible change in stability characteristics, operation of the vessel in weather conditions which overwhelm the vessel, etc.

Furthermore, the human factor often plays a role in those casualties where the direct cause was the failure of some vessel component. Required or prudent maintenance may not have done or possibly the cleanliness of the vessel was not maintained which led to fire.

Faced with increasing fisheries conservations efforts and by greater competition for limited resources, operators are embarking on voyages in weather conditions which would otherwise dictate staying in port. With vessels routinely encountering conditions which severely tax their capabilities, losses are inevitable. It should be kept in mind that only 2% of the fishing fleet are larger than 100 feet. Therefore, seas which pose only a minimal threat to the larger segments of the merchant fleet can be life-threatening to the relatively small fishing vessel.

One particular causal area which is of increasing concern, and more so in certain fisheries than in others, is that of stability. Casualty data for the period 1970-1982 indicate that approximately 44% of all losses can be categorized as foundering, flooding or capsizing. Of these, about 13% can be attributed to capsizing. Some losses due to capsizing can be traced directly to an unstable vessel design. In one case a vessel capsized, was placed back into service, and capsized again resulting in a fatality. However, not all capsizings are due to unstable design. A vessel's intact stability condition is influenced by three basic factors:

a. Inherent stability.
b. Operations.
c. Forces of nature.

The naval architect and/or builder has the greatest control over the original design and consequently, the initial stability of the vessel. If the purchaser makes modifications, as often happens, the original stability can be adversely affected. A vessel can also become an unstable platform if operated beyond its capabilities. There appears to be little appreciation on the part of many vessel operators of the number of hazards confronted at sea which can seriously reduce a vessel's stability. Among these are: .

a. Excess deck weights (crab pots, catch, nets, etc.).
b. Free surface (water on deck or within the hull).
c. Operations in following seas (reduction in waterplane as a vessel momentarily sits on a wave).
d. Beam seas/winds.
e. Towing.
f. Off center loading or unsecured catch on deck.

Again, the human element is present. A number of casualty reports have been reviewed in which the vessel was built to International Maritime Organization recommended stability criteria and had stability loading charts but the operator neglected to load or operate the vessel according to these charts.

Analysis - Personnel Deaths

In addition to vessels, commercial fishing also exacts a heavy toll in human lives. Between 1970 and 1980, an average of 103 fishermen died from accidents onboard vessels, from natural causes, or as the result of a vessel being lost or damaged. In 1981 and 1982, the average declined to 69 deaths per year but much of this decline was due to a change in reporting requirements. Starting with 1981, deaths resulting from natural causes, including suicide, were no longer required to be investigated. The average for these two years does appear to be somewhat below the average for earlier years, if one excludes those deaths resulting from natural causes. The expected average should be between 80-85 per year. The total number of deaths onboard documented fishing vessels from all causes between 1970-1982 are shown in Figure 2. These deaths are then broken down in Table 7 as to whether the death resulted from the total loss of a vessel, a vessel suffering damages, or from accidental causes.

FIGURE 2: Fatalities on Documented F/Vs 1970 - 1982

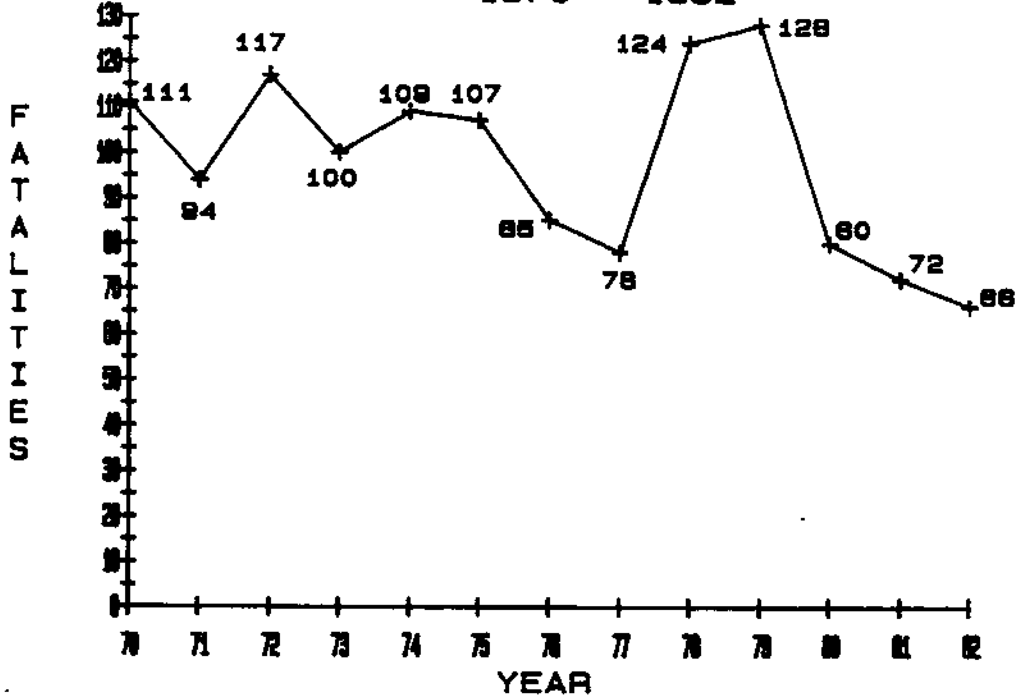


TABLE 7: DEATHS ONBOARD DOCUMENTED FISHING VESSELS 1970 -1982

Year	Total Loss of Vessel	Vessel Damaged	Accident Onboard Vessel
1970	59	6	46
1971	40	9	45
1972	59	11	47
1973	30	21	49
1974	44	14	51
1975	53	23	31
1976	37	10	38
1977	38	6	34
1978	52	21	51
1979	50	18	60
1980	46	5	29
1981	25	22	25
1982	31	3	32

The greatest loss of life occurs when a vessel is totally lost through foundering or capsizing. An average of 30.2 fishermen died during these occurrences each year, almost two-thirds of the deaths associated with the total loss of a vessel. Similarly, an average of 6 deaths, or about one-half of the deaths resulting from a damaged vessel, occurred when flooding or capsizing of a vessel took place. By far the greatest cause of accidental deaths was from falling overboard. An average of 20.7 persons died each year from these falls.

In order to develop fatality rates, statistics on the number of persons engaged in commercial fishing are needed. Unfortunately these statistics are lacking in sufficient detail. In this analysis, fatality rates for 1981 and 1982 were developed by estimating the total number of fishermen using casualty reports received by the Coast Guard. The number of persons onboard the vessel at the time of the casualty was noted along with the size of the vessel involved. Using this method, the number of fishermen onboard documented vessels was estimated to be approximately 83,000 for 1981 and 1982. From this, it is estimated that the fatality rate for fishermen was approximately 86 per 100,000 persons in 1981 and 83 per 100,000 in 1982. This is about 7 times greater than that of the national average for all types of workplace safety as shown in Table 8 below.

TABLE 8: U.S. WORKPLACE FATALITY RATES 1981 & 1982: Rates per 100,000 Workers

Industry Group	1981	1982
All Industries	12	11
Mining, Quarrying	55	55
Construction	40	40
Government	10	10
COMMERCIAL FISHING	86	83

Note: Fatality data for workplace safety (except commercial fishing) extracted from ACCIDENT FACTS, 1982 and 1983 editions, published by the National Safety Council.

These rates do not appear to be out of line with foreign experience. A study by Hans Peder Pederson entitled, "Fatal Accidents On Norwegian Fishing Vessels During The Years 1961-1975" states:

> "Comparing work onboard the fishing vessels against shore industries in Norway the level of risk for loss of life, fishing is 10-20 times higher than in the industries ashore. The risk level in the fisheries is about 13 fatalities/10,000 man year."

Summary

This analysis presents a general overview of the safety record of the fishing fleet between 1970-1982. The analysis does not cover all aspects of fishing vessel safety which should be examined, nor does it address in detail the question of why certain things are taking place. Instead, it provides a foundation for measuring the fleet's safety record in future years and a starting place for additional analysis. Work in other areas needs to be done such as the identification of losses based on geographic areas and specific fisheries. A more accurate means of determining the number of fishermen that work on documented fishing vessels needs to be developed in order to generate more accurate fatality rates. Also, a more accurate means of developing vessel loss rates would be useful since rates based on vessel population does not take into account the varying degrees of vessel utilization. There should also be an analysis done to determine why the vessel loss rate has started to increase again. Finally, a program, or programs, aimed at reducing the losses should be developed and implemented.

APPENDIX A: U.S. DOCUMENTED FISHING VESSELS DAMAGED BREAKDOWN BY NATURE OF CASUALTY

	1970	1971	1972	1973	1974	1975	1976	1977	1978	1979	1980	1981	1982
COLLISION	165	109	121	178	182	149	168	194	326	226	257	92	103
FIRES& EXPLOSIONS	20	24	19	32	36	33	37	33	60	38	49	45	37
GROUNDING	130	85	114	119	197	180	174	231	283	262	233	140	108
FOUNDERING, FLOODING, CAPSIZING	10	9	24	39	70	88	81	109	152	139	109	91	111
HEAVY WX	21	0	2	7	9	0	5	2	0	0	2	4	4
MATERIAL FAILURE	63	23	196	194	174	53	90	146	356	456	455	519	357
OTHER	6	1	10	27	33	11	17	18	24	46	56	0	5
TOTAL	415	251	486	596	701	604	572	733	1201	1167	1161	891	725

Note: Effective January 1981, one of the reporting criteria for defining a reportable marine casualty was revised. Whereas the reporting criteria for being a reportable casualty was property damage in excess of $1500., the revised criteria for property damaged was raised to $25,000.

A REVIEW OF SOME RECENT STABILITY CASUALTIES INVOLVING PACIFIC NORTHWEST FISHING VESSELS

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Abstract

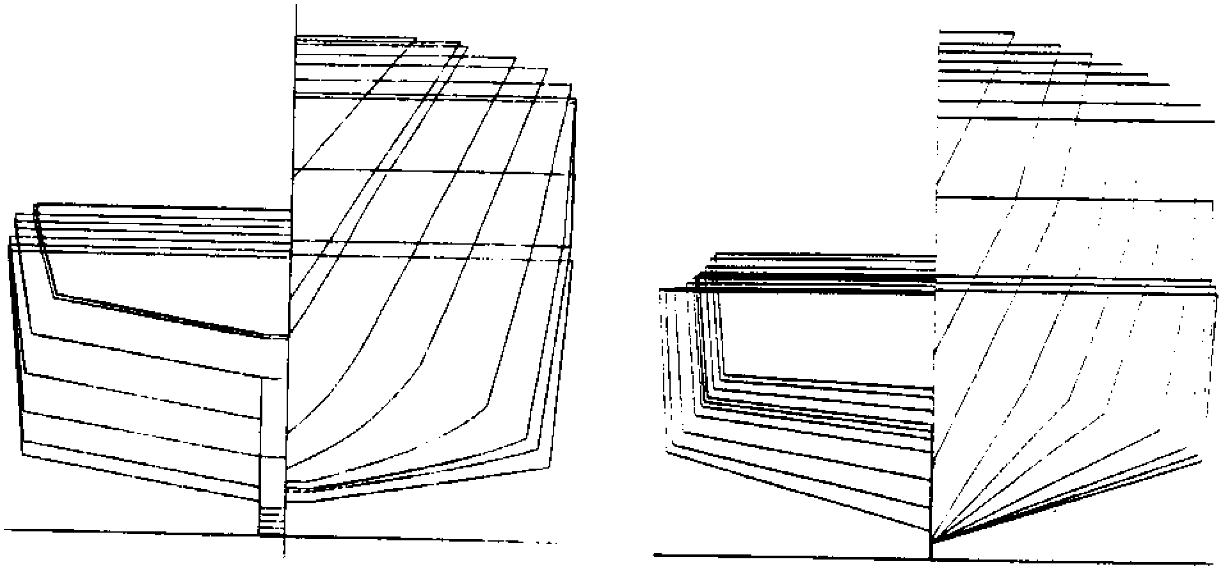
Fishing vessels engaged in the Alaska king-crab fishery are subject to large variations in loading resulting from variable fuel loading, the need to carry the catch in large circulating seawater tanks and the number of heavy crab pots carried on deck. In the past year, losses of vessels and lives have been particularly heavy due to capsizing. Five capsizing cases are reviewed, including the construction of probable loading conditions at the time of capsizing and an examination of other possible contributing factors. Recommendations are provided which may assist operators and naval architects in reducing the tragic toll of losses.

Introduction

Over the years, the Alaska king-crab fishery has proven to be very hazardous, with large losses of vessels and lives. The 1982-83 season seemed extraordinary in this respect. Major losses were suffered as a result of capsizings. Five of these capsizing casualties are described in this paper.

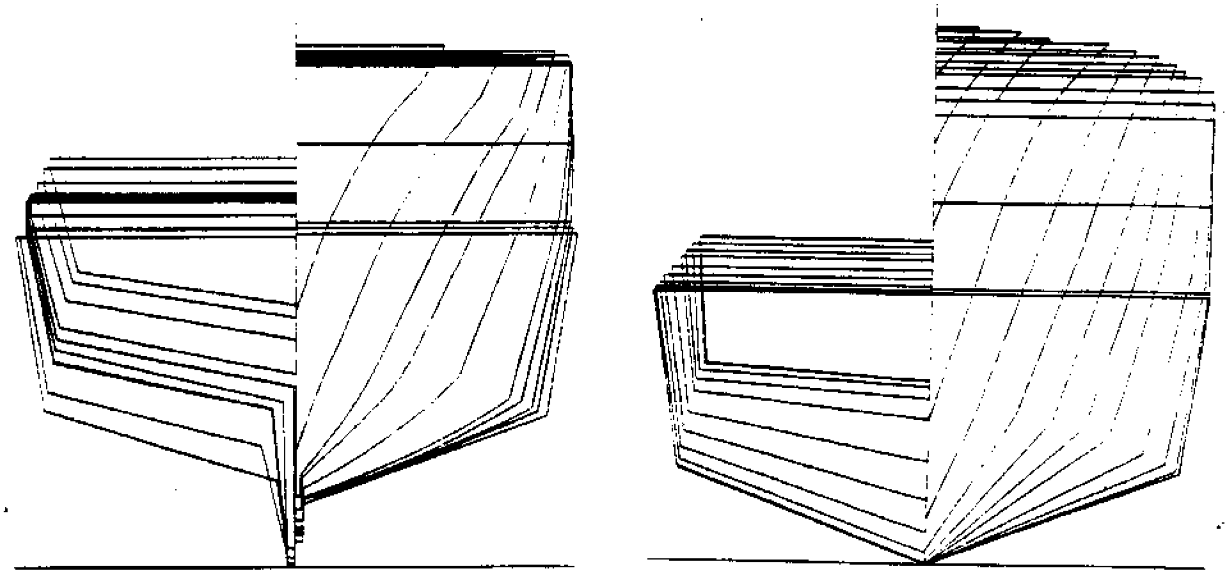
Most of the vessels engaged in the king-crab fishery have very similar characteristics. They are hard chine vessels with a long open deck and low freeboard aft. Forward, the vessels generally have considerable flare, and a high focsle deck and deckhouse. Crab tanks are located below the main deck aft. When in use they are filled with circulating seawater and live crab. The engine room is located forward of the crab tanks. Above the engine room are the galley and crew's quarters. Because of the significant price differential for fuel between Alaska and the Northwest states, the vessels are often capable of carrying relatively large amounts of fuel. Body plans for the vessels discussed in this paper are shown in Figure 1.

The crab are captured in large pots weighing about 700 pounds each which must be stacked on the deck of the vessel as it travels to and from the fishing grounds. When the fishing begins, the pots are baited and set in the water. After some time, the pots are recovered and the crab loaded into the tanks. Once one of the circulating seawater tanks is filled, there is little change in the weight of the vessel no matter how many crab are caught. Major changes in loading take place as the pot load on deck changes.



AMERICUS/ALTAIR

OCEAN GRACE



GOLDEN VIKING

ARCTIC DREAMER

Figure 1. Body plans of four Alaska king crab vessels.

In the past few years, rapidly declining stocks of crab have led to changes which tend to reduce the stability margin for many of these vessels. Many of the crabbing areas near the major ports have been depleted and fishing emphasis has shifted to more distant and more hostile locations. Because of the short openings of the seasons, as much as gear as possible must be employed and moved into position. The tendency is to make a minimum number of trips to move pots. Consequently, the vessels often are heavily loaded.

The decline in the crab fishery has also led many owners to add trawling gear enabling vessels to be used for a greater portion of the year. These conversions add significant weight above the main deck and reduce the stability of the converted vessel.

When assessing the statical stability of larger fishing vessels, the International Maritime Organization (IMO) stability guidelines [1] are almost universally applied in the United States. Section 5.1 of these guidelines establishes the following stability criteria:

(a) The area under the righting lever curve should not be less than:

  (1) 0.055 meter-radians (10.339 feet-degrees) up to 30 degrees

  (2) 0.09 meter-radians (16.918 feet-degrees) up to 40 degrees

  (3) 0.03 meter-radians (5.639 feet-degrees) between 30 and 40 degrees.

(b) The righting lever should be at least 0.20 meters (0.631 feet) at an angle of heel equal to or greater than 30 degrees.

(c) The maximum righting lever should occur at an angle of heel preferably exceeding 30 degrees but not less than 25 degrees.

(d) The initial metacentric height should not be less than 0.35 meters (1.148 feet).

This assumes that the angle of downflooding is greater than 40 degrees.

Stability curves are calculated for the subject vessels under various possible loading conditions using the constant trim moment method of calculation. In most of these conditions, the free surface effect is small and is included in the analysis by using an apparent position of the vertical center of gravity which includes a "maximum" free surface correction rather than the formula contained in the IMO stability guidelines.

Icing does not appear to have played a major role in the cases considered, and it has not been included in the calculations.

The reader is also asked to view the results presented with some caution. While every effort has been made to accurately reconstruct the loading condition of the vessel, many assumptions were necessary in each case. In the cases of the Americus and Altair there were no survivors, so there were no statements from the crew which could be used in determining the exact loading.

The purpose of this paper is not to publicize specific casualties or embarrass any individuals, but to help the fishing community, including naval architects and operators, learn from the tragedies which others have suffered.

Americus/Altair

The F.V. Americus and the F.V. Altair were two of seven sister ships built for king crabbing. These vessels were 123.5 feet in overall length with a beam of 32.0 feet at the main deck. The Americus was delivered in 1978 and the Altair in 1980. Over the period from December 1981 until January 1983, the vessels were substantially modified for use in trawling as well as crabbing. The modifications added an estimated 35.2 tons to the original light ship displacement of 289.3 tons.

A stability booklet had been prepared for each of these vessels at the time they were delivered. No modifications of the stability booklets had been made to include the subsequent additions to the vessels. All of the the stability calculations were based on an inclining of the F.V. Antares, the first of the class. There is no indication that a deadweight survey had been performed on the later vessels in the class to compare with the Antares.

Following work in a shipyard in January, the vessels left Anacortes, Washington for Dutch Harbor, Alaska in early February 1983. Before their departure the vessels took on fuel to bring their total to about 73,000 gallons each. When they arrived in Dutch Harbor the vessels discharged about 28,000 gallons of fuel each, loaded crab pots and prepared to enter the Pribilof Islands crab fishery.

The Altair left Dutch Harbor about 2 a.m., followed by the Americus at about 8:30 a.m. on 14 February 1983. Both vessels were almost identically loaded. Another fishing vessel heading for Dutch Harbor passed the Altair about 4 a.m. around 20 miles northwest of Dutch Harbor. No messages were ever received from either boat. The next contact with the vessels was made by a passing freighter at 3:20 p.m. on 14 February when the Americus was reported floating upside down about 30 miles northwest of Dutch Harbor. The hull did not appear to have been damaged and continued to float until 11:30 a.m. on 16 February. Thirty-two days after the casualty an inflatable raft identified as coming from the Altair was found about 5 miles from where the Americus was first sighted floating upside down. All 14 hands are presumed lost with the vessels.

At the time of the casualty the weather conditions were relatively calm for the area. Winds reported by vessels in the vicinity were between 10 and 20 knots from the east. There was a swell of about 1 meter in height and seas of about 0.5 to 1 meter. Both vessels should have been in beam seas on the course from Dutch Harbor to the Pribilof Islands.

In attempting to determine the reasons for the capsizing of the Americus, the investigation initially followed two paths. The first was to examine the existing calculations found in the stability booklet. Figure 2, Curve 1 is the stability curve for the Americus for loading condition 5 (heavily loaded) described in the stability booklet. The vessel easily meets the IMO stability criteria in this condition. Curve 1 is based on the original light ship weight estimate which must be modified to account for the addition of the trawling gear. If the light ship weight is modified, then stability Curve 2 in Figure 2 is the result. This loading is detailed in Table 1 for reference. The vessel in this loading condition would not meet all the IMO stability criteria.

The second path which was followed was to reconstruct the actual loading condition as the vessels departed Dutch Harbor. Based on the evidence, two possible loading conditions were developed. The only difference between these was in the distribution of the estimated fuel on board. Figures 3 and 4, Curve 1 represent the stability curves for the two possible loading conditions, with an assumed light ship weight including the modifications. With the double bottom fuel tanks full (Figure 3, Curve 1) all IMO stability criteria are satisfied except (c) because the peak of the curve occurs at about 21 degrees. With the double bottom fuel tanks empty, stability criteria (b) and (c) are not satisfied.

Although the calculations show the vessels would not meet IMO stability criteria under these possible loading conditions, they do not indicate grossly unstable vessels in a condition where both would be likely to capsize.

The next step was to verify the applicability of the results of the inclining experiment performed on the Antares to the Americus and Altair at the time of their loss. An opportunity arose in July 1983 to perform an inclining test on the F.V. Morning Star, another sister ship which had also been converted to a combination crabber/trawler. The inclining experiment revealed a startling difference. After the conversion weight was accounted for, the Morning Star was about 56 tons heavier than an estimate based on the Antares inclining indicated.

Using the data for the Morning Star, a new estimate was made for the Americus/Altair light ship condition. Tables 2 and 3 are estimates of two possible loading conditions as the vessels left Dutch Harbor based on the Morning Star inclining. The stability curves for these conditions are plotted in Figures 3 and 4, Curve 2. The load condition from the stability booklet was also recalculated and is plotted in Figure 2, Curve 3.

Particularly with the double bottom fuel tanks empty, these vessels as they were loaded would be vulnerable to capsizing.

Because of the magnitude of the difference in the light ship weight, additional corroborating evidence was sought. A deadweight survey on the F.V. Viking Explorer, another sister vessel, (Viking Explorer had not been converted to a combination vessel at the time of the deadweight survey) revealed a difference in light ship weight of almost the same 60-ton magnitude. Another vessel, the F.V. Aleyeska, has subseqeently been reinclined. Although this is not a sister vessel, it is almost the same except for a shallower depth. In this case there also was an increase of about 60 tons in light ship weight over and above the changes which resulted from conversion when the initial and recent inclining results were compared.

At the Coast Guard inquiry into the loss of these two vessels, testimony was introduced from which an approximate waterline at the time they left Dutch Harbor may be calculated. The displacement for this waterline is about 20 tons above the displacement estimated based on the Morning Star inclining as the possible loading condition at the time the vessels were lost.

Ocean Grace

The F.V. Ocean Grace was delivered in 1980 for use in the Alaska king-crab fishery. It had an overall length of 107 feet and a beam at the main deck of 27.17 feet.

In August 1983 the vessel was preparing to leave Dutch Harbor to enter the St. Matthew crab fishery. Some problems were discovered with the steering system and at least temporary repairs were made which would not delay the vessel's departure. The vessel loaded fuel, water, and pots in the Dutch Harbor area and departed at about 2 p.m. on 14 August 1983. About an hour after leaving, the captain did not like the way the vessel was handling and decided to fill the aft crab tank with seawater. The tank was filled successfully but the engineer was concerned with the very heavy load the vessel was carrying. He reported that the guard on the port side was underwater from a little ahead of midship all the way to the stern except for the last 10 feet at the stern. Soon after

> she layed over a little bit, ..., on a normal swell, but she just stopped there. She didn't come back. On the next one she went a lot farther. Well, that got everybody startled, and everybody came piling out of their staterooms, ... I didn't say anything, I was just trying to feel the boat, ..., thinking, well, okay, she should start coming back, and she never did. [2]

At the time the vessel capsized, conditions were moderate and getting worse. The wind was about 30-35 knots, and the vessel was travelling in 6- to 7-foot following seas. There was one survivor out of a crew of 5.

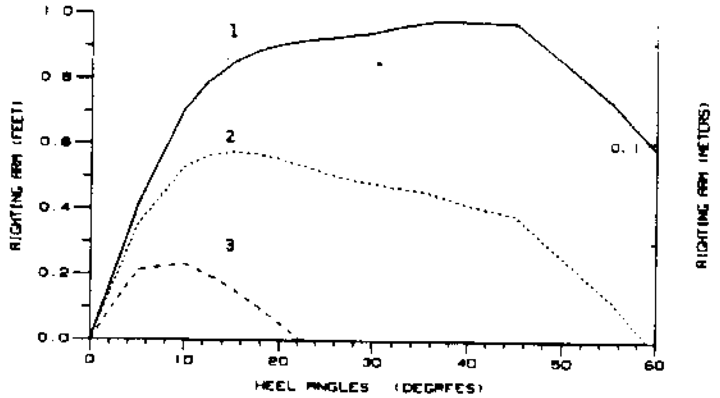


Figure 2. Stability curves for F.V. Americus for loading condition No. 5 (see Table 1) in stability booklet.

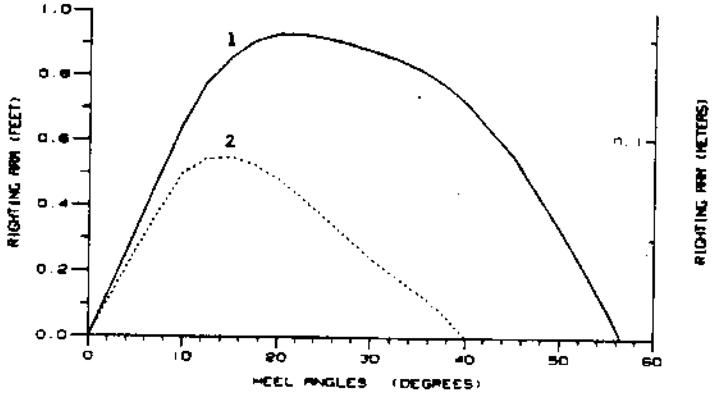


Figure 3. Stability curves for F.V. Americus for estimated loading as vessel left Dutch Harbor (double bottom fuel tanks full, see Table 2).

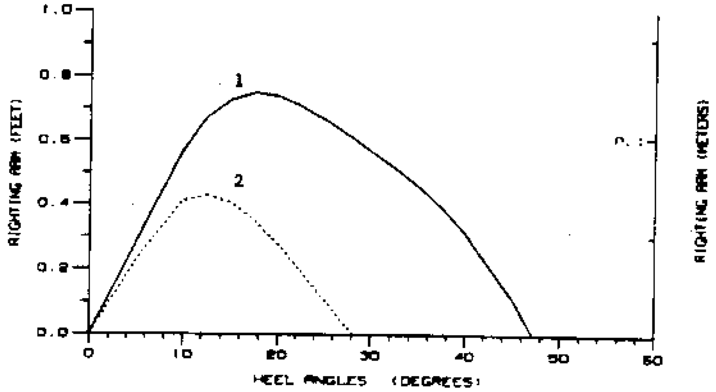


Figure 4. Stability curves for F.V. Americus for estimated loading as vessel left Dutch Harbor (double bottom fuel tanks empty, see Table 3).

Table 1. Loading of F.V. Americus as described in stability booklet (see Figure 2, curve 2).

VESSEL: AMERICUS/ALTAIR
CONDITION : LOADING CONDITION NO. 5 FROM STABILITY BOOKLET
LIGHT SHIP BASED ON ANTARES INCLINING WITH ADDITION OF CONVERSION WEIGHT
LCG EXPRESSED IN FEET AFT OF FORWARD PERPENDICULAR
VCG EXPRESSED IN FEET ABOVE THE BASELINE
WEIGHT IN LONG TONS
CRAB POTS ARE 7 X 7 X 2.67 FT @ 700 LB EACH

DESCRIPTION	%	F.S.	WEIGHT	VCG	V MOM	LCG	L MOM
CREW, PROVISIONS & STORES	0.	0.0	2.00	22.50	45.00	15.00	30.00
REFERIGERATED STORES	0.	0.0	5.00	20.50	102.50	34.20	171.00
SHIP'S STORES FR 33-36	0.	0.0	2.00	12.50	25.00	88.20	176.40
NO. 1 CARGO TANK P/S	100.	0.0	137.50	10.75	1478.13	50.00	6875.00
NO. 2 CARGO TANK P/S	100.	0.0	133.60	11.00	1469.60	70.00	9352.00
F. W. NO. 1 WING TANK P/S	50.	2.7	18.70	6.50	121.55	30.12	563.24
F. O. DAY TANK FR 4-6	100.	66.5	18.10	13.16	238.20	10.60	191.86
F. O. NO. 1 DBTM TANK P/S	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. NO. 2 DBTM TANK P/S	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. NO. 2 WING TANK P/S	100.	3.2	33.00	10.53	347.49	50.27	1658.91
F. O. NO. 3 WING TANK P/S	100.	0.0	31.50	10.58	333.27	69.66	2194.29
F. O. NO. 4 WING TANK P/S	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. DEEP TANK C. L.	100.	0.0	47.20	11.97	564.98	93.66	4420.75
HYDRAULIC OIL FR 6-8 S	95.	1.7	4.30	14.10	60.63	15.13	65.06
LUBE OIL TANK	95.	1.7	4.30	14.10	60.63	15.13	65.06
SEWAGE TANK	10.	0.9	0.30	6.35	1.91	15.86	4.76
CONT. OIL TANK FR 6-10	0.	0.0	0.00	0.00	0.00	0.00	0.00
CRAB POTS 1ST TIER (94)	0.	0.0	29.40	20.50	602.70	80.00	2352.00
CRAB POTS 2ND TIER (30)	0.	0.0	9.40	25.33	238.10	80.00	752.00
CRAB POTS 3RD TIER (30)	0.	0.0	9.40	28.00	263.20	80.00	752.00
CRAB POTS 4TH TIER (30)	0.	0.0	9.40	30.67	288.30	80.00	752.00
TOTALS		76.7	495.10	12.61	6241.18	61.35	30376.33
LIGHT SHIP			324.46	15.80	5126.47	51.79	16803.78
LOADED DISPLACEMENT		76.7	819.56	13.87	11367.65	57.57	47180.12

TRANSVERSE METACENTER ABOVE BASELINE	18.04
TRANSVERSE METACENTRIC HEIGHT	4.17
FREE SURFACE CORRECTION	0.09
APPARENT TRANSVERSE METACENTRIC HEIGHT	4.08
APPARENT VERTICAL CENTER OF GRAVITY	13.96

Table 2. Possible loading of F.V. Americus as vessel left Dutch Harbor (see Figure 3, curve 2).

VESSEL: AMERICUS/ALTAIR
CONDITION : POSSIBLE LOADING WITH DOUBLE BOTTOM TANKS FILLED
LIGHT SHIP BASED ON MORNING STAR INCLINING
LCG EXPRESSED IN FEET AFT OF FORWARD PERPENDICULAR
VCG EXPRESSED IN FEET ABOVE THE BASELINE
WEIGHT IN LONG TONS
CRAB POTS ARE 7 X 7 X 2.67 FT @ 700 LB EACH

DESCRIPTION	%	F.S.	WEIGHT	VCG	V MOM	LCG	L MOM
CREW, PROVISIONS & STORES	0.	0.0	5.00	22.50	112.50	15.00	75.00
REFERIGERATED STORES	0.	0.0	5.00	20.50	102.50	34.20	171.00
SHIP'S STORES FR 33-36	0.	0.0	5.00	12.50	62.50	88.20	441.00
NO. 1 CARGO TANK P	100.	0.0	65.84	10.75	707.78	50.00	3292.00
NO. 2 CARGO TANK S	100.	0.0	63.29	11.00	696.19	70.00	4430.30
F. W. NO. 1 WING TANK P/S	35.	15.7	13.07	5.79	75.68	30.12	393.67
F. O. DAY TANK FR 4-6	85.	66.5	15.37	12.30	189.05	10.60	162.92
F. O. NO. 1 DBTM TANK P/S	100.	0.0	26.02	4.31	112.15	50.34	1309.85
F. O. NO. 2 DBTM TANK P/S	100.	0.0	23.82	4.70	111.95	69.00	1643.58
F. O. NO. 2 WING TANK P/S	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. NO. 3 WING TANK P/S	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. NO. 4 WING TANK P/S	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. DEEP TANK C. L.	100.	0.0	47.20	11.97	564.98	93.66	4420.75
HYDRAULIC OIL FR 6-8 S	100.	0.0	4.65	14.10	65.57	15.13	70.35
LUBE OIL TANK	100.	0.0	4.65	14.10	65.57	15.13	70.35
SEWAGE TANK	0.	0.0	0.00	0.00	0.00	0.00	0.00
CONT. OIL TANK FR 6-10	0.	0.0	0.00	0.00	0.00	0.00	0.00
CRAB POTS 1ST TIER (88)	0.	0.0	27.50	20.50	563.75	75.80	2084.50
CRAB POTS 2ND TIER (28)	0.	0.0	8.75	25.33	221.64	75.80	663.25
CRAB POTS 3RD TIER (28)	0.	0.0	8.75	28.00	245.00	75.80	663.25
CRAB POTS 4TH TIER (28)	0.	0.0	8.75	30.67	268.36	75.80	663.25
CRAB POTS 5TH TIER (28)	0.	0.0	8.75	33.33	291.64	75.80	663.25
CRAB POTS 6TH TIER (28)	0.	0.0	8.75	36.00	315.00	75.80	663.25
TOTALS		82.2	350.16	13.63	4771.80	62.49	21881.53
LIGHT SHIP			380.00	16.57	6296.60	54.26	20618.80
LOADED DISPLACEMENT		82.2	730.16	15.16	11068.40	58.21	42500.33

TRANSVERSE METACENTER ABOVE BASELINE	18.27
TRANSVERSE METACENTRIC HEIGHT	3.11
FREE SURFACE CORRECTION	0.11
APPARENT TRANSVERSE METACENTRIC HEIGHT	3.00
APPARENT VERTICAL CENTER OF GRAVITY	15.27

Table 3. Possible loading of F.V. Americus as vessel left Dutch Harbor (see Figure 4, curve 2).

VESSEL: AMERICUS/ALTAIR
CONDITION : POSSIBLE LOADING WITH DOUBLE BOTTOM TANKS EMPTY
LIGHT SHIP BASED ON MORNING STAR INCLINING
LCG EXPRESSED IN FEET AFT OF FORWARD PERPENDICULAR
VCG EXPRESSED IN FEET ABOVE THE BASELINE
WEIGHT IN LONG TONS
CRAB POTS ARE 7 X 7 X 2.67 FT @ 700 LB EACH

DESCRIPTION	%	F.S.	WEIGHT	VCG	V MOM	LCG	L MOM
CREW, PROVISIONS & STORES	0.	0.0	5.00	22.50	112.50	15.00	75.00
REFERIGERATED STORES	0.	0.0	5.00	20.50	102.50	34.20	171.00
SHIP'S STORES FR 33-36	0.	0.0	5.00	12.50	62.50	88.20	441.00
NO. 1 CARGO TANK P	100.	0.0	65.84	10.75	707.78	50.00	3292.00
NO. 2 CARGO TANK S	100.	0.0	63.29	11.00	696.19	70.00	4430.30
F. W. NO. 1 WING TANK P/S	35.	15.7	13.07	5.79	75.68	30.12	393.67
F. O. DAY TANK FR 4-6	85.	66.5	15.37	12.30	189.05	10.60	162.92
F. O. NO. 1 DBTM TANK P/S	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. NO. 2 DBTM TANK P/S	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. NO. 2 WING TANK P/S	25.	1.4	8.24	5.76	47.46	50.27	414.22
F. O. NO. 3 WING TANK P/S	100.	0.0	31.50	10.58	333.27	69.66	2194.29
F. O. NO. 4 WING TANK P/S	100.	0.0	59.80	12.59	752.88	92.41	5526.12
F. O. DEEP TANK C. L.	0.	0.0	0.00	0.00	0.00	0.00	0.00
HYDRAULIC OIL FR 6-8 S	100.	0.0	4.65	14.10	65.57	15.13	70.35
LUBE OIL TANK	100.	0.0	4.65	14.10	65.57	15.13	70.35
SEWAGE TANK	0.	0.0	0.00	0.00	0.00	0.00	0.00
CONT. OIL TANK FR 6-10	0.	0.0	0.00	0.00	0.00	0.00	0.00
CRAB POTS 1ST TIER (88)	0.	0.0	27.50	20.50	563.75	75.80	2084.50
CRAB POTS 2ND TIER (28)	0.	0.0	8.75	25.33	221.64	75.80	663.25
CRAB POTS 3RD TIER (28)	0.	0.0	8.75	28.00	245.00	75.80	663.25
CRAB POTS 4TH TIER (28)	0.	0.0	8.75	30.67	268.36	75.80	663.25
CRAB POTS 5TH TIER (28)	0.	0.0	8.75	33.33	291.64	75.80	663.25
CRAB POTS 6TH TIER (28)	0.	0.0	8.75	36.00	315.00	75.80	663.25
TOTALS		83.6	352.66	14.51	5116.33	64.20	22641.98
LIGHT SHIP			380.00	16.57	6296.60	54.26	20618.80
LOADED DISPLACEMENT		83.6	732.66	15.58	11412.93	59.05	43260.79

TRANSVERSE METACENTER ABOVE BASELINE	18.26
TRANSVERSE METACENTRIC HEIGHT	2.68
FREE SURFACE CORRECTION	0.11
APPARENT TRANSVERSE METACENTRIC HEIGHT	2.57
APPARENT VERTICAL CENTER OF GRAVITY	15.69

Table 4. Maximum pot loading of F.V. Ocean Grace fom stability booklet (see Figure 5).

VESSEL: OCEAN GRACE
CONDITION : LOADING CONDITION FROM STABILITY BOOK FOR MAXIMUM POTS
LCG EXPRESSED IN FEET AFT OF FORWARD PERPENDICULAR
VCG EXPRESSED IN FEET ABOVE THE BASELINE
WEIGHT IN LONG TONS
CRAB POTS ARE 7 X 7 FT @ 700 LB EACH

DESCRIPTION	%	F.S.	WEIGHT	VCG	V MOM	LCG	L MOM
CREW & EFFECTS	0.	0.0	0.70	18.57	13.00	19.62	13.73
BOSUN'S STORES	0.	0.0	0.50	10.00	5.00	87.62	43.81
ENGINEER'S STORES	0.	0.0	0.50	8.00	4.00	17.62	8.81
CONSUMABLE STORES	0.	0.0	0.50	16.00	8.00	31.62	15.81
NO. 1 CARGO TANK (FWD)	100.	0.0	88.60	9.30	823.98	56.80	5032.48
NO. 2 CARGO TANK (AFT)	100.	0.0	50.90	10.10	514.09	85.95	4374.85
FRESH WATER	50.	4.0	9.00	6.78	61.02	43.07	387.63
FUEL OIL	50.	99.0	34.10	7.10	242.11	24.90	849.09
LUBE OIL TANK	64.	1.0	1.40	3.57	5.00	25.34	35.48
HYDRAULIC OIL	64.	1.0	1.40	3.57	5.00	25.34	35.48
SEWAGE TANK	0.	0.0	0.00	0.00	0.00	0.00	0.00
CONT. OIL TANK	0.	0.0	0.00	0.00	0.00	0.00	0.00
CRAB POTS ON DECK (68)	0.	0.0	21.30	17.18	365.93	70.85	1509.10
TOTALS		105.0	208.90	9.80	2047.13	58.91	12306.27
LIGHT SHIP			184.00	14.92	2745.28	42.06	7739.04
LOADED DISPLACEMENT		105.0	392.90	12.20	4792.41	51.02	20045.31

TRANSVERSE METACENTER ABOVE BASELINE	15.12
TRANSVERSE METACENTRIC HEIGHT	2.92
FREE SURFACE CORRECTION	0.27
APPARENT TRANSVERSE METACENTRIC HEIGHT	2.66
APPARENT VERTICAL CENTER OF GRAVITY	12.46

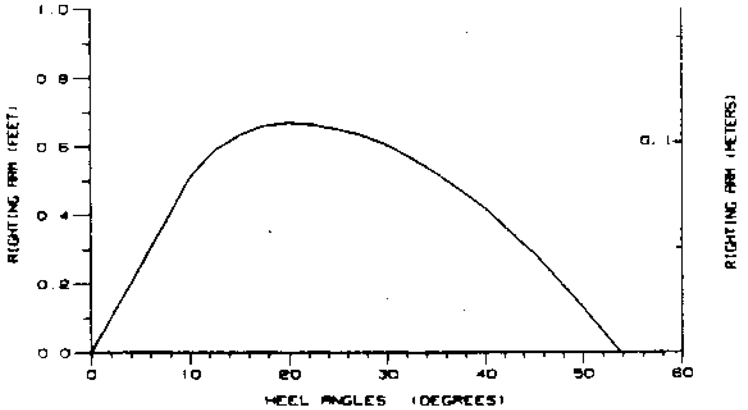


Figure 5. Stability curve for F.V. Ocean Grace for maximum pot loading from stability booklet.

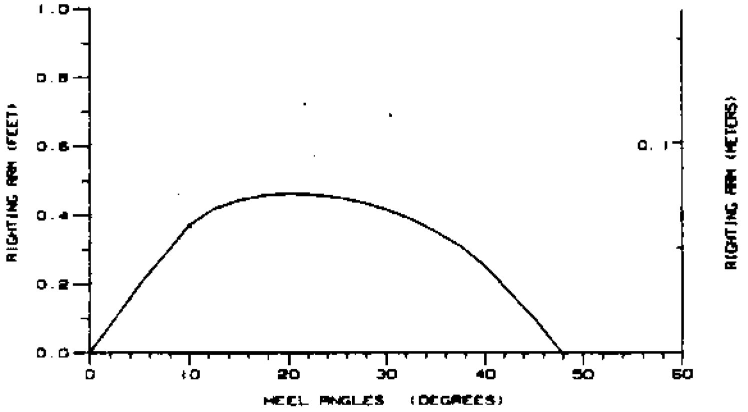


Figure 6. Stability curve for F.V. Ocean Grace for assumed loading as the vessel left Dutch Harbor.

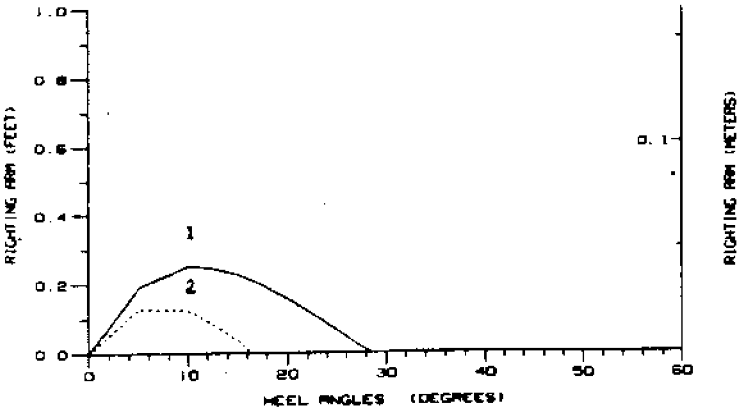


Figure 7. Stability curves for F.V. Ocean Grace for assumed loading with aft tank flooded.

Table 5. Possible loading of F.V. Ocean Grace as it left Dutch Harbor (see Figure 6).

VESSEL: OCEAN GRACE
CONDITION : LOADING CONDITION DESCRIBED BY ENGINEER/RELIEF SKIPPER
LCG EXPRESSED IN FEET AFT OF FORWARD PERPENDICULAR
VCG EXPRESSED IN FEET ABOVE THE BASELINE
WEIGHT IN LONG TONS
CRAB POTS ARE 7 X 7 X 2.83 FT @ 700 LB EACH

DESCRIPTION	%	F.S.	WEIGHT	VCG	V MOM	LCG	L MOM
CREW & EFFECTS	0.	0.0	1.50	19.00	28.50	18.00	27.00
REFERIGERATED STORES	0.	0.0	4.00	18.00	72.00	30.00	120.00
SHIP'S STORES	0.	0.0	1.00	20.00	20.00	4.00	4.00
NO. 1 CARGO TANK (FWD)	100.	0.0	88.60	9.30	823.98	56.80	5032.48
NO. 2 CARGO TANK (AFT)	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. W. TANK P/S	100.	0.0	28.98	8.90	257.92	41.20	1193.98
F. O. DBTM. TANK	100.	0.0	16.50	5.56	91.74	18.78	309.87
F. O. NO. 1 WING TANK P	100.	0.0	13.17	10.10	133.02	30.68	404.06
F. O. NO. 1 WING TANK S	100.	0.0	10.78	9.80	105.64	29.90	322.32
F. O. NO. 2 WING TANK P/S	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. NO. 2 C. L. TANK	100.	0.0	10.12	10.98	111.12	95.00	961.40
LUBE OIL TANK	64.	1.0	1.03	2.82	2.90	25.12	25.87
HYDRAULIC OIL	64.	1.0	1.03	2.82	2.90	25.12	25.87
SEWAGE TANK	0.	0.0	0.00	0.00	0.00	0.00	0.00
CONT. OIL TANK	0.	0.0	0.00	0.00	0.00	0.00	0.00
CRAB POTS AFT TANK (11)	0.	0.0	3.44	8.25	28.38	56.00	192.64
CRAB POTS 1ST TIER (54)	0.	0.0	16.88	17.30	292.02	71.50	1206.92
CRAB POTS 2ND TIER (21)	0.	0.0	6.56	22.22	145.76	75.00	492.00
CRAB POTS 3RD TIER (21)	0.	0.0	6.56	25.04	164.26	75.00	492.00
CRAB POTS 4TH TIER (16)	0.	0.0	5.00	27.88	139.40	75.00	375.00
TOTALS		2.0	215.15	11.25	2419.56	51.99	11185.41
LIGHT SHIP			184.00	14.92	2745.28	42.06	7739.04
LOADED DISPLACEMENT		2.0	399.15	12.94	5164.84	47.41	18924.45

TRANSVERSE METACENTER ABOVE BASELINE	15.26
TRANSVERSE METACENTRIC HEIGHT	2.32
FREE SURFACE CORRECTION	0.01
APPARENT TRANSVERSE METACENTRIC HEIGHT	2.32
APPARENT VERTICAL CENTER OF GRAVITY	12.94

Table 6. Possible loading of F.V. Ocean Grace with aft crab tank flooded (see Figure 7, curve 1).

VESSEL: OCEAN GRACE
CONDITION : LOADING CONDITION DESCRIBED BY ENGINEER/RELIEF SKIPPER
AFT CRAB TANK FLOODED
LCG EXPRESSED IN FEET AFT OF FORWARD PERPENDICULAR
VCG EXPRESSED IN FEET ABOVE THE BASELINE
WEIGHT IN LONG TONS
CRAB POTS ARE 7 X 7 X 2.83 FT @ 700 LB EACH

DESCRIPTION	%	F.S.	WEIGHT	VCG	V MOM	LCG	L MOM
CREW & EFFECTS	0.	0.0	1.50	19.00	28.50	18.00	27.00
REFERIGERATED STORES	0.	0.0	4.00	18.00	72.00	30.00	120.00
SHIP'S STORES	0.	0.0	1.00	20.00	20.00	4.00	4.00
NO. 1 CARGO TANK (FWD)	100.	0.0	88.60	9.30	823.98	56.80	5032.48
NO. 2 CARGO TANK (AFT)	100.	0.0	50.86	10.10	513.69	75.00	3814.50
F. W. TANK P/S	100.	0.0	28.98	8.90	257.92	41.20	1193.98
F. O. DBTM. TANK	100.	0.0	16.50	5.56	91.74	18.78	309.87
F. O. NO. 1 WING TANK P	100.	0.0	13.17	10.10	133.02	30.68	404.06
F. O. NO. 1 WING TANK S	100.	0.0	10.78	9.80	105.64	29.90	322.32
F. O. NO. 2 WING TANK P/S	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. NO. 2 C. L. TANK	100.	0.0	10.12	10.98	111.12	95.00	961.40
LUBE OIL TANK	64.	1.0	1.03	2.82	2.90	25.12	25.87
HYDRAULIC OIL	64.	1.0	1.03	2.82	2.90	25.12	25.87
SEWAGE TANK	0.	0.0	0.00	0.00	0.00	0.00	0.00
CONT. OIL TANK	0.	0.0	0.00	0.00	0.00	0.00	0.00
CRAB POTS AFT TANK (11)	0.	0.0	3.44	8.25	28.38	56.00	192.64
CRAB POTS 1ST TIER (54)	0.	0.0	16.88	17.30	292.02	71.50	1206.92
CRAB POTS 2ND TIER (21)	0.	0.0	6.56	22.22	145.76	75.00	492.00
CRAB POTS 3RD TIER (21)	0.	0.0	6.56	25.04	164.26	75.00	492.00
CRAB POTS 4TH TIER (16)	0.	0.0	5.00	27.88	139.40	75.00	375.00
TOTALS		2.0	266.01	11.03	2933.25	56.39	14999.91
LIGHT SHIP			184.00	14.92	2745.28	42.06	7739.04
LOADED DISPLACEMENT		2.0	450.01	12.62	5678.53	50.53	22738.95

TRANSVERSE METACENTER ABOVE BASELINE	14.89
TRANSVERSE METACENTRIC HEIGHT	2.27
FREE SURFACE CORRECTION	0.00
APPARENT TRANSVERSE METACENTRIC HEIGHT	2.27
APPARENT VERTICAL CENTER OF GRAVITY	12.62

According to the stability booklet, the maximum number of pots recommended for the Ocean Grace was 68. A complete listing of the loading for these conditions is given in Table 4 and the stability curve is plotted in Figure 5. Under the calculation scheme used in this paper to compute the stability curve, the vessel does not meet the stability criteria in this loading condition. There is not sufficient area under the curve between 30 and 40 degrees. The maximum righting lever occurs at about 20 degrees and the righting lever is insufficient at 30 degrees or greater.

An estimate of the loading condition for the Ocean Grace as it left Dutch Harbor was reconstructed on the basis of the survivor's statement. This is described in Table 5 and the stability curve for this loading condition is plotted in Figure 6. Under this loading condition the vessel fails to meet all of the stability criteria except (d), the initial metacentric height requirement.

Before the vessel capsized, the aft tank was filled. This apparently was completed while the vessel was still in protected water. The loading in this condition is given in Table 6 and the stability curve is plotted in Figure 7 as Curve 1. Looking at this figure leaves little doubt that the vessel had very little static stability. Curve 2 in Figure 7 is for the same loading condition with a maximum free surface moment for the aft crab tank also included.

Golden Viking

The F.V. Golden Viking was constructed in 1975 for king crabbing. The vessel had an overall length of 84.15 feet and a beam at the main deck of 24.5 feet. In early 1983 the vessel was "stretched" by adding 13 feet to the stern, bringing the overall length to 97.33 feet after the modification. At the time a new inclining experiment was performed and a new stability booklet prepared.

Following the lengthening, the vessel returned to fish successfully in the remaining seasons early in 1983. The capsizing occurred on 1 September 1983 near St. Matthew Island. The vessel had loaded crab pots from its storage area and was proceeding toward St. Matthew Island to begin fishing. The vessel seemed to develop a port list which had caused concern for the captain and engineer. Checks were initiated for slack fluid in the tanks, but none was found, so some fuel was transferred to starboard. As fuel was being transferred, the vessel took a starboard list more suddenly than the captain expected and he ordered some fuel transferred back to the port tanks. The vessel then proceeded to the fishing grounds in a quartering sea on the starboard bow. When they reached the fishing area, the captain began a turn to port and the vessel was hit by a wave which crested over the starboard rail. The water on deck caused a severe starboard list, breaking loose the fresh bait on deck, which contributed to the starboard list. The vessel seemed to hold a starboard list. To correct this, the captain steered hard to starboard and applied full power in an attempt to right the boat. The attempt was unsuccessful and the vessel and two of the crew of six were lost.

The weather reported at the time the vessel capsized was winds of 25-30 knots, an 8 to 10-foot swell and about a 4-foot chop.

There are several phenomena which occurred in this case that should be carefully noted by operators. The first is the description of the rapid shift in list from port to starboard as fuel was transferred. This is a sign of low stability. The second is the large angle of heel assumed by the vessel when the water swept on deck. This appears to be related to the description of a vessel reaching a "pseudo-static" angle of heel. This is a well-documented phenomenon for vessels with low metacentric height. When water is trapped on deck these vessels will take an immediate list of 20-30 degrees into the waves and will appear to have reached equilibrium at this angle of heel until the water is sufficiently cleared from the deck or the vessel capsizes. Finally, there is the qaestion of what can be done to save a vessel in grave danger of capsizing. Tests of a radio-controlled model indicate that no sudden actions should be taken. Power should be reduced to zero and the rudder should be held close to the center position, which allows the boat to slowly right itself (this presumes no downflooding is occurring).

The application of maximum power and hard over rudder may do two things to decrease the chances of saving the vessel. Since the propeller shaft is usually below the center of gravity, the increase in power results in trim by the stern. Increased water velocity in the region of the propeller also results in lower pressure on the stern further increasing trim by the stern. In this type of vessel this is detrimental to stability.

If a vessel is heeled to starboard and a hard starboard turn is initiated, the immediate effect is to cause further heel to starboard. It is not until the turn is underway that centrifugal force will cause the vessel to heel to port.

The maximum pot loading recommended for the Golden Viking in its stability booklet allows 61 pots on deck. The loading is given in Table 7 and the stability curve is plotted in Figure 8. In this loading condition the vessel meets all the stability criteria.

There is some discrepancy between the captain's and the engineer's description of the loading at the time the vessel capsized. The captain claims the vessel was carrying 95 crab pots and the engineer 103. The captain's description is given in Table 8 and the associated stability curve is plotted in Figure 9, Curve 1. The stability curve based on the engineer's statement of loading is given in Figure 9, Curve 2. In both cases the vessel would have had insufficient stability.

In this case it is interesting to examine how the crab pot loading would have to be reduced in order for the vessel to meet the stability criteria. Figure 10 illustrates a variety of pot reductions. Curve 1 is based on the captain's statement but with a reduction of 1 tier of pots, leaving 2 tiers with a total of 84 pots. Curve 2 is for 1 tier of 60 pots and Curve 3 is with no pots on deck. To meet the stability

Table 7. Maximum pot loading of F. V. Golden Viking from stability booklet (see Figure 8).

```
     VESSEL: GOLDEN VIKING
     CONDITION : LOADING CONDITION X FROM STABILITY BOOKLET
          LCG EXPRESSED IN FEET AFT OF FORWARD PERPENDICULAR
          VCG EXPRESSED IN FEET ABOVE THE BASELINE
          WEIGHT IN LONG TONS
          CRAB POTS ARE 6.5 X 6.5 X 2.83 FT @ 700 LB EACH

DESCRIPTION                 % F.S.  WEIGHT    VCG     V MOM     LCG     L MOM
------------------------------------------------------------------------------
CREW & EFFECTS             0.  0.0    1.50  18.25     27.38   23.12     34.68
BAIT FREEZER               0.  0.0    2.50  12.50     31.25   63.88    159.70
NO. 1 CARGO TANK          98.  0.0   86.71  10.60    919.13   40.12   3478.81
NO. 2 CARGO TANK           0.  0.0    0.00   0.00      0.00    0.00      0.00
F. W. DBTM. TANK          98.  0.0    2.98   6.65     19.82    9.18     27.36
F. W. PEAK TANK            0.  0.0    0.00   0.00      0.00    0.00      0.00
F. O. DAY TANK P/S        98.  1.3    5.44  10.85     59.02   30.58    166.36
F. O. NO. 1 TANK P/S      98.  2.9   13.88  10.90    151.29   37.52    520.78
F. O. NO. 2 TANK P/S      98.  2.9   12.94  11.00    142.34   46.88    606.63
F. O. NO. 3 TANK P/S      10.  2.4    1.10   8.20      9.02   54.98     60.48
F. O. NO. 4 TANK P/S       0.  0.0    0.00   0.00      0.00    0.00      0.00
F. O. NO. 5 TANK P/S       0.  0.0    0.00   0.00      0.00    0.00      0.00
F. O. DBTM. TANK P/S      98.  0.0   10.42   5.35     55.75   37.08    386.37
LUBE OIL TANK              0.  0.0    0.00   0.00      0.00    0.00      0.00
SEWAGE TANK                0.  0.0    0.00   0.00      0.00    0.00      0.00
CONT. OIL TANK             0.  0.0    0.00   0.00      0.00    0.00      0.00
CRAB POTS 1ST TIER (58)    0.  0.0   18.13  19.70    357.16   60.92   1104.48
CRAB POTS 2ND TIER (13)    0.  0.0    4.06  24.55     99.67   65.72    266.82
------------------------------------------------------------------------------
                 TOTALS        9.5  159.66  11.72   1871.82   42.67   6812.46
------------------------------------------------------------------------------
LIGHT SHIP                          190.33  14.83   2822.59   36.92   7026.98
------------------------------------------------------------------------------
LOADED DISPLACEMENT            9.5  349.99  13.41   4694.42   39.54  13839.44
------------------------------------------------------------------------------

     TRANSVERSE METACENTER ABOVE BASELINE       16.45
     TRANSVERSE METACENTRIC HEIGHT               3.04
     FREE SURFACE CORRECTION                     0.03
     APPARENT TRANSVERSE METACENTRIC HEIGHT      3.01
     APPARENT VERTICAL CENTER OF GRAVITY        13.44
```

Table 8. Possible loading (Captain's description) of F.V. Golden Viking at the time the vessel capsized (see Figure 9, curve 1).

VESSEL: GOLDEN VIKING
CONDITION : LOADING CONDITION DESCRIBED BY CAPTAIN
LCG EXPRESSED IN FEET AFT OF FORWARD PERPENDICULAR
VCG EXPRESSED IN FEET ABOVE THE BASELINE
WEIGHT IN LONG TONS
CRAB POTS ARE 6.5 X 6.5 X 2.83 FT @ 700 LB EACH

DESCRIPTION	%	F.S.	WEIGHT	VCG	V MOM	LCG	L MOM
CREW & EFFECTS	0.	0.0	1.50	18.25	27.38	23.12	34.68
REFERIGERATED STORES	0.	0.0	3.00	20.50	61.50	30.00	90.00
SHIP'S STORES	0.	0.0	1.00	18.00	18.00	0.00	0.00
BAIT LOCKER (AFT)	0.	0.0	2.68	11.00	29.48	65.00	174.20
BAIT BOXES (DECKHOUSE)	0.	0.0	2.68	24.00	64.32	32.50	87.10
NO. 1 CARGO TANK	98.	0.0	86.71	10.60	919.13	40.12	3478.81
NO. 2 CARGO TANK	0.	0.0	0.00	11.15	0.00	40.12	0.00
F. W. DBTM. TANK	98.	0.0	2.98	6.65	19.82	9.18	27.36
F. W. PEAK TANK	98.	0.0	4.43	12.80	56.70	3.02	13.38
F. O. DAY TANK P/S	98.	0.0	5.44	10.85	59.02	30.58	166.36
F. O. NO. 1 TANK P	5.	1.5	0.32	6.55	2.10	37.52	12.01
F. O. NO. 1 TANK S	33.	1.5	2.27	7.85	17.82	37.52	85.17
F. O. NO. 2 TANK P	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. NO. 2 TANK S	25.	1.4	1.62	7.81	12.65	46.48	75.30
F. O. NO. 3 TANK P	98.	0.0	5.37	11.85	63.63	56.38	302.76
F. O. NO. 3 TANK S	0.	0.0	0.00	0.00	0.00	0.00	0.00
F. O. NO. 4 TANK P/S	98.	0.0	9.68	13.35	129.23	66.02	639.07
F. O. NO. 5 TANK P/S	98.	0.0	18.12	14.45	261.83	77.72	1408.29
F. O. DBTM. TANK P/S	0.	0.0	0.00	0.00	0.00	0.00	0.00
LUBE OIL TANK	0.	0.0	0.00	11.70	0.00	29.22	0.00
SEWAGE TANK	0.	0.0	0.00	0.00	0.00	0.00	0.00
CONT. OIL TANK	0.	0.0	0.00	0.00	0.00	0.00	0.00
CRAB POTS 1ST TIER (60)	0.	0.0	18.75	19.70	369.38	60.92	1142.25
CRAB POTS 2ND TIER (24)	0.	0.0	7.50	24.36	182.70	60.92	456.90
CRAB POTS 3RD TIER (11)	0.	0.0	3.44	27.19	93.53	57.00	196.08
TOTALS		4.4	177.49	13.46	2388.22	47.27	8389.70
LIGHT SHIP			190.33	14.83	2822.59	36.92	7026.98
LOADED DISPLACEMENT		4.4	367.82	14.17	5210.81	41.91	15416.68

TRANSVERSE METACENTER ABOVE BASELINE	16.47
TRANSVERSE METACENTRIC HEIGHT	2.30
FREE SURFACE CORRECTION	0.01
APPARENT TRANSVERSE METACENTRIC HEIGHT	2.29
APPARENT VERTICAL CENTER OF GRAVITY	14.18

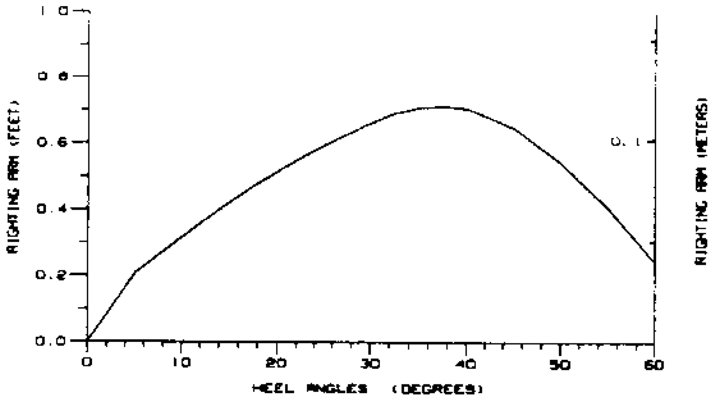


Figure 8. Stability curve for F.V. Golden Viking for maximum pot loading from stability booklet.

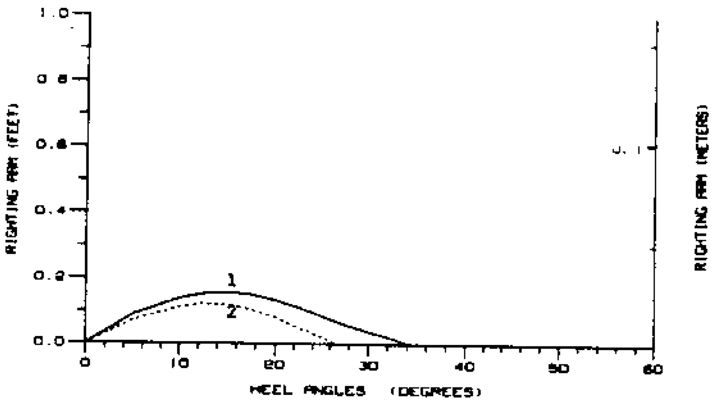


Figure 9. Stability curves for F.V. Golden Viking for possible loading at the time of capsizing.

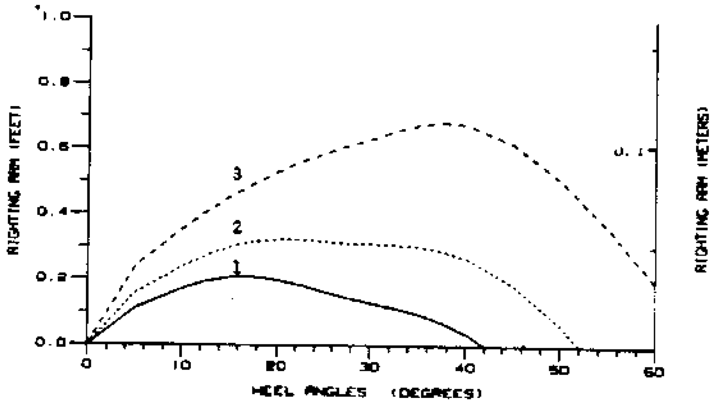


Figure 10. Stability curves for F.V. Golden Viking for possible loading with reduced numbers of crab pots.

criteria, the vessel would be able to carry only a few pots.

Arctic Dreamer

The Arctic Dreamer was built in 1980 and was a little over 98 feet in overall length with a 24-foot beam at the main deck. This design is based on an earlier 86-foot vessel to which an additional 12 feet were added. The 12 feet were added aft of the rudder, extending the stern and providing more deck area aft.

At the time of the casualty the vessel was proceeding from Akutan to Dutch Harbor. The forward crab tank was loaded. The vessel was full of fresh water and had between 6 and 7 thousand gallons of fuel in addition to full wing tanks which were not used. Between 68 and 70 pots were carried on deck.

The skipper was outside the bridge on the port wing. He looked aft and noticed that the stern had started to settle under the water and that there was water on deck. He ran to the steering station and switched from auto pilot to jog and applied full throttle in an attempt to raise the stern. The vessel dipped the starboard bulwark and water came onto the main deck. He then turned the rudder hard to starboard but the vessel did not recover. All the crew members were rescued.

The loss of the Arctic Dreamer is a very interesting case because several similar vessels have also been lost as a result of capsizing. Unfortunately, the available information is insufficient to perform accurate stability calculations. There seem to be considerable differences in the tankage described by the captain and similar ships and the light ship data from the inclining experiment performed on a sister ship are not contained in the stability booklet. However, it should be noted that the stability booklet recommends a maximum of 50 pots when one of the crab tanks is flooded.

Conclusions

When preparing a stability booklet for a vessel, consideration should be given to performing an inclining test for every vessel, including sister ships. At a minimum, a careful deadweight survey is needed to insure the applicability of a previous inclining.

These casualties also raise questions about the changes in light ship weight experienced by larger fishing vessels over time. This needs to be monitored on a regular basis with a reinclining required should significant changes occur.

The conversion to combination crabber/trawler vessels should also be followed by a reinclining. In preparation for a conversion, a deadweight survey should certainly be performed to insure that a previous inclining is still applicable. If this is not done, we may wind up with converted vessels which are not well suited to either fishery.

In conducting the inclining experiment on the Morning Star, it was necessary to weigh most of the items on board. This revealed that the estimates widely used for crew effects, stores and spare parts are low. Because of the price differential these vessels will fill with fuel and stores in Washington or Oregon before they travel north to Alaska. Consequently, at the beginning of the season the vessels may be heavily loaded with these items.

Meeting the IMO stability guidelines appears to be a good way to reduce the possibility of capsizing. In designing a vessel which may be converted in the future, a margin of safety above the guidelines also appears to be in order. The light ship weight increase which vessels seem to experience also indicates the need for a factor of safety.

On the operational side there are many recommendations. To begin with, the dangers of stern quartering and following seas must be understood. It is much more difficult to capsize a vessel in head or bow quartering seas.

Double bottom fuel tanks should be considered ballast tanks and only used when there is no more fuel left.

Everyone should be aware of the effect of modifications and additions on vessel stability. If the captain kept track of the drafts for a vessel under specific loading conditions, changes in these drafts over time would indicate changes in the vessel's stability and a new stability booklet could be requested.

In attempting to save a vessel that is on the verge of capsizing, the application of full power and hard over rudder are not recommended. It is better to reduce power, close any openings which can be closed to prevent downflooding, and very slowly turn into the waves. While heading at low speed into the waves, any possible corrective action (pumping out a flooded hold, dumping pots, etc.) should be taken.

The most important stability problem in the king-crab fishery is overloading. We must recognize that freeboard is critical to maintaining vessel stability.

Acknowledgement

The support of the Washington Sea Grant Program under a grant from the National Oceanic and Atmospheric Administration is gratefully acknowledged. Their support of the Fishing Vessel Safety Center at the University of Washington has made this paper possible. The cooperation of both the 13th and 17th Coast Guard Districts has been invaluable. In addition, several naval architects made significant contributions to this effort.

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LEGAL ASPECTS OF FISHING VESSEL SAFETY*

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This paper presents a brief overview of certain legal aspects of fishing vessel safety. The legal standards of liability as they effect the builder, designer, seller and repairmen are examined. Special emphasis is placed on the legal treatment of a vessel and its components as products. One critical aspect is whether the absence of stability would render a vessel defective as a product. Using certain technical considerations by a typical stability casualty as a starting point, the legal ramifications of imposing greater stability standards are then reviewed. It is pointed out that the demands of the legal and engineering disciplines sometimes conflict. The concluding section offers certain guidelines for promoting fishing vessel safety within this context.

Introduction

The commercial fishing industry has one of the worst safety records of any domestic industry.[1] Studies have shown that stability related casualties,[2] although among the least frequent form of loss, are associated with the greatest loss of life and vessel.[3] Yet the operational fact of life is that few fishermen are formally trained mariners and most have only an intuitive notion of whether their craft is stable. As a consequence, as we examine the question of how to promote fishing vessel safety and specifically how to reduce stability casualties, it is natural to focus on the inherent stability characteristics of the vessel, rather than on operational considerations.

Many of us are familiar with the hallmarks of a typical stability casualty. A loaded vessel is returning at night to port after several days at sea. The vessel may be overloaded with the extra catch stored outside of the holds, often in a manner that affects the center of gravity. The crew, except for the member at the helm, are asleep or otherwise occupied. The weather and sea, while not calm, do not appear to be hazardous, and so no special precautions are taken. The door to the lazarette may be open. Running with trailing or quarter seas, the aft portion of the deck becomes partially immersed, creating an entrapped water problem. Sea water begins to fill the fish holds or enter

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[*] The views expressed herein are solely those of the author.

the lazarette space. Perhaps at that point, the crew is alerted to the impending problem. Emergency actions often compound the loss of stability, and within minutes the vessel has capsized.

Setting aside the operational decisions of the crew, this scenario suggests the enormity of the technical problems involved: how to assess the effects of reduced freeboard, overloading, changes in center of gravity, loss of righting arm due to wave crests, entrapped water, etc., on a vessel's stability. Indeed, one author has noted that there are over thirty parameters that may have to be considered.[4] It is thus encouraging to note that the technical aspects of fishing vessel safety, especially as they relate to stability casualties, are receiving increasing attention in the technical literature.[5] One must remember, however, that the era of sophisticated stability analysis was ushered in some 45 years ago with Rahola's seminal analysis of fishing vessel stability.[6] As is evident from the continuing number of stability casualties, we remain far away from reaching a comprehensive solution to the problem of ensuring greater fishing vessel safety through improved design.

Similarly, on the legal front, the recent attempts by attorneys and others to articulate the legal standards applicable to the marine industry at large are also encouraging.[7] This is an equally difficult and important task, for it defines the framework in which the technical improvements often will be evaluated. It is my intention in presenting this paper to make the first step toward bridging the technical and legal gap in the context of promoting greater fishing vessel safety.

Legal Framework

A. Application of Admiralty Law

In understanding how the law effects the liability of builders, designers, sellers and others involved in the construction and outfitting of fishing vessels, two important considerations must be kept in mind. The first of these is derived from our federal constitutional mandate which dictates that the power of the federal courts shall extend to cases of admiralty and maritime jurisdiction.[8] Congress, since the beginning of this nation, has seen fit to endow the federal courts with jurisdiction over admiralty matters, although always reserving to litigants their rights to bring common law claims in state court.[9] The result has been the development of a distinct body of substantive law referred to as the general maritime or admiralty law.

With rare exception, Congress has avoided legislating substantive maritime law.[10] This has meant that the federal courts traditionally have been the primary source for the development of admiralty law. While these courts have labored to formulate a uniform law in this area, this evolution has not been without its problems, and numerous issues remain unresolved.[11]

The second consideration is more specific and is really an example of the practical implications of the first principle. In looking at the nature of substantive maritime law, one obvious and fundamental question is whether a contract to construct a ship is maritime in nature and thus within the court's maritime jurisdiction. Federal courts have consistently held that a contract to construct a vessel is not a maritime contract and, therefore, not subject to federal admiralty law.[12] This exception has been described as "the curious anomaly" of admiralty law. This does not mean that a contract to build a vessel or supply component parts would not wind up in federal court as an admiralty matter. Rather, it simply means that the various contractual relationships will be governed by the appropriate state law, rather than the general maritime law. In comparison, contracts to repair vessels have been deemed to be maritime contracts and thus cognizable in admiralty.[13] As a result, even though the nature of repair work is similar to new construction, the legal consequences may vary.

It is this second consideration that in recent years has had to bear the brunt of the infusion of products liability law into admiralty. At one time, problems relating to the construction of a vessel essentially were limited to contract disputes. With the evolution of products liability law in this century and its gradual incorporation into admiralty law, that statement is no longer true. What has happened is that federal courts have recognized that tortious conduct arising out of construction contracts is properly an admiralty matter. The gradual result has been to make such tortious conduct the focus of the litigation with the contract questions assuming a secondary role.

The introduction of products liability law into admiralty is generally acknowledged to have been initiated by the Court of Appeals for the Third Circuit's 1945 decision in Sieracki v. Seas Shipping Co.[14] Sieracki was a longshoreman injured by a falling boom and tackle while loading cargo after the shackle supporting the boom had failed. The trial court determined that the shackle had broken as a result of a forging defect. The appellate court, although concluding that the shipyard and its subcontractor had not been negligent, indicated that a maritime tort would be evaluated according to the standards of land based products liability law. The United States Supreme Court affirmed the appellate court's decision although the language of that decision did not deal directly with the products liability issue. Sieracki's precedential significance to products liability law has been to legitimize the perception that such action could be maintained in admiralty. This notion subsequently has been reinforced by numerous appellate court decisions.

The second seminal decision affecting products liability law in the maritime context was the United States Supreme Court's 1972 decision in Executive Jet Aviation, Inc. v. City of Cleveland.[15] This case arose out of an airplane crash, and the issue was whether the claim could be brought as an admiralty case in federal court. The traditional rule in the case of a maritime tort looked only to the locality of the injury. In Executive Jet, the Court held that in order to have a maritime tort it was necessary that (1) the injury occur on navigable waters (the locality requirement), and (2) the wrong bear some relationship to traditional maritime activities (the nexus requirement). While this decision ostensibly sought to reduce maritime tort case loads by eliminating certain tort actions such as airplane crashes, it also affirmed the rule that, for purposes of determining admiralty jurisdiction, the presence of a maritime tort was determined by the point of its occurrence, rather than by the conduct causing the tort. It thus mattered not that the underlying tortious conduct occurred on land and had little or no connection to maritime activities.

When viewed in conjunction with the gradual incorporation of products liability law into admiralty, the effect of the Executive Jet rule has been to bring nearly all phases of the design, construction, and repair of fishing vessels within the federal courts admiralty jurisdiction. It is, for example, settled law that an action in negligent design or construction can be brought by an owner against a shipyard concurrently with a contract action.[16] But the rule has been extended far beyond this. Thus, for instance, it has been held that the manufacturer of a defective steering system has a product liability claim against its sellers in admiralty for manufacturing allegedly defective component parts.[17] Significantly, the court there determined it was immaterial whether the subcontractor had manufactured the components specifically for use in marine systems; it was sufficient merely that the product had found its way into a maritime application.[18] Such a holding clearly exposes a supplier or component parts manufacturer to suit in admiralty by an owner/operator or by other users, even though there is no contractual privity.

B. What is a Products Liability Action?

I have already alluded to the term "products liability" in discussing the Executive Jet decision. While I do not want to duplicate what other authors have done, a brief introduction into the nature of a products liability action is necessary because it is the focus of this paper. One authority has defined it in the following terms:

> the liability of a manufacturer, processor, or nonmanufacturing seller for injury to the person or property of a buyer or third party caused by a product which has been sold.[19]

Generally, recovery under a products liability theory does not include what the law refers to as intentional torts, such as fraud or deceit. The traditional assessment of a products liability action focuses on three components: (1) contract warranties, (2) negligence, and (3) strict liability. There is, in addition, a fourth theory which is taking on increasing importance: misrepresentation.

This is not to say that every claim arising from a fishing vessel casualty is properly a "products liability" claim. For example, the traditional rule applicable to a designer or naval architect is that his professional services are not to be regarded as products.[20] This is true even though he may furnish plans or drawings as part of his service.[21] Similarly, a repairyard may be subject to liability for breaching its warranty of workmanlike service where it has selected unsafe or defective equipment.[22] Yet, with certain exceptions, the legal theories encompassed within a products liability claim are generally applicable across the board to the activities of builders, designers, etc., and will form the basis of the following remarks.

The contract claim looks to the nature of the agreement between the parties. This breaks down into roughly two concerns. The first is whether the parties have performed the contract in accordance with the terms of the agreement. In effect, this concern largely ends when the product is delivered. The second, in contrast to the first, is whether the warranties contained in the contract have been breached. The emphasis here is on the use of the product.

It should be noted that such warranties could be either express or implied. Express warranties are identifiable from the terms of the contract. As an example, a builder may expressly warrant that his vessel conforms to certain regulatory standards. Implied warranties arise from the relationship of the parties. Thus, as to an owner/purchaser, the builder would be under an implied warranty that his product is suitable for its intended use. In an analogous fashion, a repairer would be under an implied duty to perform its services in a workmanlike manner. The hallmark of the warranty action is that plaintiff's burden of proof is simplified. As explained by one court, the plaintiff need only show that the warranty existed, that the product did not conform to the warranty and that the deviation proximately caused the injury or damage.[23] This contrasts significantly with the tort theories of recovery which will now be reviewed.

An action in negligence may coexist with an action in contract or exist in the absence of a contractual claim. It is important to understand that the presence of a contractual relationship between the parties does not automatically displace a negligence claim. The negligence claim is generally said to have four elements: (1) the existence of a duty or obligation recognized by law; (2) a failure to conform to that standard; (3) a causal connection between the breach of the duty and the resulting harm; and (4) some actual loss or damage.[24] It is evident from this definition that the focus is on the conduct of a liable party, as opposed to the terms of the agreement in the contract context.

The negligence action depends on the standard of care imposed by the court on the parties. This evaluation is subject to an assessment of the relative skills of the parties, the nature and extent of the work and the kind of judgment exercised by the parties. For example, the traditional rule is that a marine designer must exercise reasonable care in developing its design, but this duty is not limited to the party with whom the designer is in contractual privity. Thus, in one of the first post-Sieracki cases, the representatives of deceased crew members of a trawler lost after a fishing vessel capsized were allowed to bring their claims sounding in negligent design against both the shipbuilder and naval architect.[25] The vessel was found to lack sufficient stability because of faulty design. The implication of this decision is that the designer's duty to use due diligence in his design extends to the ultimate users of the vessel.

It is also a requirement of the negligence action that the act or omission be, in legal jargon, the proximate cause of the injury. An example of this latter limitation in practice is found in a recent case a designer was exonerated from liability because his alleged failure to perform certain calculations was found not to have been the cause of the casualty.[26] An important variation of the negligence action is the concept of negligence per se which is present in the case of a failure to meet statutory requirements. This rule traditionally has had the legal effect of shifting the burden of proof as to the negligence onto the offender. In fact, admiralty has an especially stringent version of this standard applicable in collision cases, called the Pennsylvania Rule, which requires that the party which violated the statute establish that it could not have been at fault for the casualty.[27] The converse proposition, that compliance with a statute would exonerate a party from negligence, is not axiomatically true. Nor should it be the case, for that would fail to promote a higher standard of care.

The third element of the typical products liability action is strict liability in tort. Here, the admiralty courts have uniformly, although not unanimously, adopted the definition given in the Restatement (Second) of Torts §402A. Under this theory of recovery, liability depends on whether the product was in a defective condition unreasonably dangerous to the user or consumer or to his property. As the term implies, showing fault is not required. Thus, it is not the nature of the conduct, but rather the nature of the item that is at issue. In theory, this claim does away with any requirement for plaintiff to establish negligent conduct and thus eases that burden of proof. Implicit in this formulation is that a "product" is involved and that the liable party is a "seller" as defined by that standard. One reason for the seller requirement as a basis of imposing liability is to ensure that the risk of increased costs can be distributed along the commercial chain. A related reason is that the costs of the improved product can be absorbed because of its mass production.

As courts have refined the concept of strict products liability under §402A, it has taken on several aspects. It has been recognized that a product can be defective because of defects in its manufacture, design or marketing.[28] The differences in those three types of defects can be illustrated by considering the typical winch used on a fishing vessel. If the winch were equipped with a shield which had broken, a manufacturing defect could be involved. If the winch did not have a guard such that it could become dangerous during operation, this would be design defect. If there were no warning about reaching under the shield, there may be a marketing defect. Any of these examples would render the product "defective" within the meaning of §402A.

As previously noted, an action based on strict liability would be cognizable in admiralty against sellers and component subcontractors, even in the absence of their specific knowledge of the product's ultimate use in a marine environment. As to the builder, inasmuch as his vessel may incorporate such defective components, he too would be liable as a "seller" under this theory. Typically, such a suit would name as defendants the builder of the vessel, its dealer and the manufacturer of the defective component product, as each of these would qualify as a seller within the meaning of §402A.

The threshold question, and, I believe this remains unresolved, is whether a vessel as a whole can be deemed to be a "product" within the meaning of §402A. There is at least one reported decision that appears to espouse that approach.[29] In that case, the owner of a fishing vessel brought an action against the shipyard which had designed, manufactured and sold the vessel. The vessel sank approximately two years after the date of sale. Although the decision does not state the cause of the sinking, the tenor of the court's decision is that the vessel itself, as opposed to a particular component, was treated as a "product" within the scope of §402A. In any event, unlike most other decisions dealing with this issue, this case did not specifically point to a defective component. The significance of this legal perception is that it in effect creates a new cause of action against a builder related to the inherent characteristics of the vessel, even though those characteristics are "design" related and no specific components were involved. In other words, the builder may be liable without a showing that he was negligent in the design or construction of the vessel.

While this decision has had limited precedential effect, it suggests that admiralty courts may eventually recognize that a strict liability action based on defective design will encompass questions of vessel stability. Stated in another way, the builder may be subject to liability under §402A because his vessel lacks stability. On the one hand, such a conclusion would seem inconsistent with the recognized principle that the liability of a designer or naval architect is limited to his negligent conduct. Yet it is the logical extension of the products liability analysis. We are all, for example, familiar with a vessel's the shoreside counterpart, the automobile, which repeatedly has been found to be defective because of its design. However, it must be conceded that a fishing vessel is not a "product" in the same sense as a car may be. The element of mass production is clearly absent. Moreover, vessels are unique in their construction and often built in accordance with the owner's guidelines.

Because of its implication in terms of the liability of a vessel builder, this is an especially critical aspect of the introduction of products liability law into maritime law. At least one appellate court has persuasively acknowledged that there are inherent limitations in the application of the strict liability doctrine to admiralty questions.[30] Resolution of these conflicting perceptions will be one of the major themes in admiralty law over the next decade. The outcome could have a significant impact on the liability of the builder of fishing vessels although this would not extend to manufacturers of component parts or systems.

There is one additional standard of liability whose significance seems destined to grow in this decade. I think this emerging theory may be especially important to the manufacturers of component equipment for use on fishing vessels. This is the standard of misrepresentation of a product as articulated in Restatement (Second) of Torts §402B:

> §402B. Misrepresentation by Seller of Chattels to Consumer
>
> One engaged in the business of selling chattels who, by advertising, labels, or otherwise, makes to the public a misrepresentation of a material fact concerning the character or quality of a chattel sold by him is subject to liability for physical harm to a consumer of the chattel caused by justifiable reliance upon the misrepresentation, even though
>
> (a) it is not made fraudulently or negligently, and
>
> (b) the consumer has not bought the chattel from or entered into any contractual relation with the seller.

The thrust of this tort is that the seller of the product is liable for physical harm to the consumer where a misrepresentation of a material fact is made to the public. It must be emphasized that the misrepresentation need not be fraudulent or require evidence of negligence; it can be made "innocently". This tort has some of the flavor of the warranty theory of contract law but is more properly regarded as a species of strict liability.

While the case law in this area is relatively sparse, this theory of liability fills a particular gap left by §402A by addressing misrepresentations relating to the character or quality of certain products. A frequently cited example is an advertisement representing that certain automobile glass is "shatterproof". A driver/owner injured by the shattering of such glass from an accident would have an action under §402B against a manufacturer, even though the glass is not defective. I suggest that manufacturers of component equipment for fishing vessels would do well to consider the implication of this theory and attempt to be circumscribed or precise about their product representations.

The questions presented by these theories of law raise complicated and difficult legal questions in the context of promoting fishing vessel safety. Who is entitled to claim the benefit of a contractual warranty? What is the appropriate standard of care in a particular set of circumstances? Is absence of stability a defect or a fishing vessel a product within the meaning of §402A? When does an advertisement become a misrepresentation? Such questions defy answering in the abstract. While addressing such questions in detail is beyond the scope of this paper, you should understand that there is extensive case law illustrating how courts apply these standards of liability to builders, designers, sellers and repairmen of fishing vessels.

Engineering Model

In the preceding section of this paper, I have discussed the legal framework confronting those engaged in the design, construction and repair of fishing vessels and their components. I now would like to focus on a second concern: how would our effort to promote greater fishing vessel safety through improved technology effect legal liability. Seemingly, we should want our legal system to promote greater safety by encouraging higher standards and by reducing exposure to liability for those who develop and maintain higher standards. Yet the realization of this objective is not so certain.

One point I think we can agree upon is that we do not want to permit the legal mechanism alone to impose more stringent standards. First, the uncertainty of this approach makes it undesirable. This reflects my earlier point that the law in this area is largely judicially fashioned, but it also follows from the inherent difficulty of the task. Second, this approach would be largely evolutionary, so it would have no immediate impact. There is, however, a third consideration which makes this inquiry germane to this conference. That is the question of how technical improvements dovetail with the demands of a society which believes in resolving disputes through litigation. In discussing this question, I will focus primarily on the problem of realizing greater vessel stability as a means of promoting safety by returning to some of the problems suggested in my earlier casualty scenario.

On one level, it would seem that law and engineering can function in a compatible fashion. For example, in considering a fishing vessel, it is readily apparent that a truly watertight bulkhead dividing the engine room from the fish hold will enhance a vessel's damage stability characteristics. The term watertight, however, may have several meanings, depending upon the perspective of the listener. To a lawyer, this means that the bulkhead would prevent the transfer of liquids or sea water. An engineer intuitively would agree, but his assessment implicitly contains certain engineering assumptions, as for example the appropriate design hydrostatic pressure. Thus, beyond that precisely defined standard, the bulkhead would not be watertight to an engineer. In most instances, the legal inquiry into whether a vessel is seaworthy[31] because of its watertight bulkhead arrangement translates into a fairly manageable engineering question. The tension between the two disciplines arises where the respective standards come into conflict.

Turning our attention toward the area of stability, we can see this problem more clearly. The question of vessel stability, whether or not raised in the context of a products liability action, has received limited judicial consideration. In a case involving a trawler casualty in which all lives were lost, the court found that the captain was negligent in attempting to return to port during a storm.[32] It also held that the vessel was unseaworthy. The vessel had been constructed from a basic plan but then modified during construction at the request of the owner/captain. The modifications consisted of lengthening the stem and raising the freeboard. A second modification was completed at another yard, also under the direction of the captain, and this resulted in the addition of a plywood enclosure extending from the deck house to the stern.

In discussing the unseaworthiness issue, the court noted that no naval architect or marine expert had been consulted regarding either of these changes. The court then proceeded to give perhaps the most thorough analysis of the appropriate standard for vessel stability appearing in the reported decisions:

> On the basis of expert testimony of naval architects at the trial,...I find that it is possible prior to construction of a vessel, once her plans and specifications have been completed, to make a computation of the metacentric height and metacentric radius of the vessel and to plot this information on a graph with other data and thus to determine her essential stability and consequently her seaworthiness. This can be done by making calculations based on weight estimates of the engine, hull, machinery, gear, cargo and stores. After the construction of a vessel it is possible to determine actual, as distinguished from theoretical, metacentric height, and hence actual as distinguished from theoretical stability of the vessel, by performing inclining tests with reference to the vessel. This was not done after construction of the MIDNIGHT SUN, and at no time prior to her last voyage were hydrostatic curves of the vessel computed, nor was any determination of her metacentric height ever made.
>
> Expert witnesses called by both sides to this controversy testified that they calculated the metacentric height of the MIDNIGHT SUN from her basic plans to be about 1.8 ft., which I find to be "moderate" absent any "degrading factors," i.e., special conditions tending to decrease metacentric height.[33]

The court then found that several such degrading factors were present: (1) seawater entrapped by the plywood shelter, (2) seawater entrapped by closed scuppers, (3) an absence of effective longitudinal partitioning in the fish hold and (4) presence of two uncovered dories on top of the pilot house. Therefore, reasoned the court, "these 'degrading factors' eradicated the moderate 1.8 ft. metacentric height of the MIDNIGHT SUN and produced a negative metacentric height during the storm, thereby rendering the vessel unstable and unseaworthy."[34]

The decision is noteworthy in that the court's determination that the vessel was unseaworthy is akin to a finding that it was defective in a strict liability sense: it does away with a requirement to prove negligent conduct or fault. The court concluded that the vessel was unseaworthy because it had insufficient metacentric height ("GM"). This result is in marked contrast to other decisions discussing GM because it amounts to a finding that the vessel was rendered defective because of the lack of GM.

As this analysis illustrates, reference to an initial GM has been the established legal standard for evaluating stability. In fact, that same standard has been recognized for years as the primary stability index by the industry.[35] Moreover, reliance on initial GM standard is understandable because it is a mathematically concise standard. Plus, it is intuitively appealing, for the absence of positive GM shows up immediately in terms of a ship's response to external forces, a permanent list being one example. Thus, the GM standard has an appeal both as a legal and engineering standard.

Yet, in another respect, this standard doesn't makes sense, for at least in the case of fishing vessels, the inadequacy of GM alone as an indicator of stability has been known since Rahola's historic study in 1939. It seems settled law that a builder would be held negligent for building a fishing vessel with inadequate or negative initial GM. Suppose, however, that the typical fishing vessel has adequate initial GM, but lacks sufficient positive righting arm at higher angles of heel or simply fails to comply with the more elaborate stability criteria contained in the IMO Resolution A.168. Under the present state of the law, it is not clear that such an absence of stability could establish negligence on the part of the owner. Arguably, however, it could render the vessel a defectively designed product under a strict liability standard. Thus, a shipyard could be held liable for constructing a defective vessel, even in the absence of any showing of negligence.

The problem becomes more acute as we attempt to assimilate more complicated technical matters into the present legal framework. In an engineering sense, saying that a fishing vessel meets this IMO standard resolves only part of the underlying question, for implicit in that conclusion is the fact that compliance is based on certain engineering assumptions. For example, it is now clear that the method of calculating cross-curves is critical to evaluating compliance with that stability standard. Use of the constant trim method, probably feasible on a hand-held calculator, may produce seriously incorrect results in the case of a fishing vessel.[36] Computation by means of the constant trim moment method, however, may require use of a sophisticated computer program. Performing such calculations thus may place a serious economic burden on the typical small yard building fishing vessels. It may, however, be a cost justified calculation because of its significant impact on assessing stability and, therefore, on creating a safer "product." A similar analysis could be applied to the selection of downflooding points for use in a stability calculation. The point is that the technical conclusion only has meaning in light of the underlying assumptions.

Recently, several technical papers have focused on more complex and mathematically elaborate questions as loss of stability due to wave crests or the effects of entrapped water. As the MIDNIGHT SUN decision indicates, these factors may well be appropriate for consideration by a court in determining liability. Yet, it is clear that the costs related to developing engineering solutions for such problems may render their treatment inaccessible in an economic sense to the typical shipyard building fishing vessels. Furthermore, such solutions only have utility in a legal sense insofar as they reasonably represent or reconstruct the situation under scrutiny.

The tension between the engineering and legal models arises because of the relative demands of the two disciplines. To an engineer, a stability criterion is essentially a hypothesis which states that compliance with those requirements will reduce the probability of a vessel capsize. It has been suggested, and I think correctly, suggested that a single all-purpose mathematical formula simply will not assure satisfactory vessel behavior under all anticipated named circumstances.[37] Even when couched in terms of probability, a legal inquiry can go astray unless the assumptions underlying the engineering answer are completely revealed. In effect, the legal question of whether a vessel is unseaworthy or defective becomes more elusive when encompassed within the framework of more technically refined engineering models.

What we ultimately have then is an inherent tension between the perspectives of engineers and attorneys, as they relate to assessing responsibility for a casualty. Advanced technical concepts do not find direct legal counterparts.

Courts have noted this problem in the context of expert witnesses testifying at trial, but the problem is really more acute as we attempt to formulate ways of designing a safer fishing vessel. My real concern is to ensure, as we introduce more sophisticated notions into our attempt to engineer sound and stable fishing vessels, that the legal system will not act in a contrary manner by exposing those people so engaged to greater liability. If I have a single response to this question, it is that all technical improvements be undertaken with the greatest degree of care.

One area that particularly troubles me is the prospect of greater computer utilizations. This will be a difficult transition for the typical small yard engaged largely in building fishing vessels. It may be necessary either to hire a naval architect of your own or run the risk of increasing costs by retaining a consultant. In a depressed market, these costs will be difficult to recover. Additionally, shipyards will have to exercise some degree of review responsibility over those independent contractors. This seems especially true when confronted with the prospect of a defectively designed vessel constituting a product within the meaning of products liability law, thus exposing you as a builder to liability even though the designer may only be measured by a negligence standard. While legal niceties such as indemnification may afford some relief and loss reallocation, your exposure may be significant. The fact of the matter is that whether builder or designer, it will be necessary to proceed on this course of action only by exercising the greatest possible care.

Conclusion

I would like to conclude by expressing several thoughts regarding the treatment of the problems which have been described.

First, the adoption of uniform regulatory or classification requirements would seem the preferrable way of setting minimum standards applicable across the board. It is not necessary, however, for these to be as stringent as is set forth in IMO Resolution A.168.[38] As an alternative, insurers could take a more active role by insisting that certain stability requirements be met as a pre-condition for issuing insurance or reducing premiums for vessels meeting such requirements.

Second, it is recommended that designers and builders should on their own initiative undertake a more sophisticated analysis of vessel stability as part of the design and manufacturing of a commercial fishing vessel. At the moment, this would seem to entail a computer modeling of your vessel from the point of view of stability; performance of an inclining test; and ensuring that certain bulkheads are watertight. Because of the sensitive nature of these vessels to stability at higher angles of heel, shipyards will be obligated to proceed with such an analysis beforehand to have a reasonable assessment of whether their product will meet those requirements. Shipyards and repairyards must appreciate that the law imposes certain implied obligations and that they, in undertaking to provide these services, also assume an obligation to fulfill those services at a certain standard.

Third, component suppliers and subcontractors must realize that they are liable for defective products and, in addition, for those products which fail to perform as specified. Product manufacturers and suppliers should be especially careful regarding their representations for their products.

Fourth, marine designers and naval architects should convey to the purchasers of their services the precise nature of their review and the underlying assumptions. This duty would extend to advising of the failure to meet the selected criteria.

Fifth, researchers should be encouraged to develop formulae that offer reasonable solutions to difficult technical problems which can be communicated or translated into cost effective methods for shipyards.

Sixth, and perhaps most importantly, the law itself must remain flexible enough to respond to evolving technology.

Admittedly, in the context of the harsh economic surrounding the construction industry, some of these comments may be unwelcome. Unlike consumer oriented industries, the fishing vessel industry is not in the position to absorb additional costs by virtue of its product line. Here the motivation is more direct: producing a safer product, especially in terms of a more stable vessel, may ultimately reduce your direct exposure to litigation. While economic demands are a definite factor, they should be balanced against these competing objectives. If the ultimate goal is to promote fishing vessel safety, we must be prepared to absorb the costs, in the hope of ultimately fostering a safer product.

REFERENCES

1. A 1980 study concluded that the death rate for fishermen was twice that of miners and 10 times the rate of all other workers.

2. A "stability related casualty" may represent a capsizing (loss of transverse stability), foundering (overwhelming by natural forces), or flooding (ingress of water from below waterline), although the focus will be on losses resulting from capsizing.

3. W.J. Ecker, *Report on Fishing Vessel Accidents* for 66th NSCE (October 1978) indicating that there is better than a 25% chance of a death occurring in the case of a stability related casualty.

4. W.A. Cleary, *Subdivision, Stability Liability*, 19 Marine Technology 228, 234 (July 1982).

5. Recent examples are: I. Caglayan and R.L. Storch, *Stability of Fishing Vessels with Water on Deck: A Review*, 26 Journal of Ship Research 106 (June 1982); J. Dillingham, *Motion Studies of a Vessel with Water on Deck*, 18 Marine Technology 38 (January 1981); R.L. Storch, *Alaskan King Crab Boat Casualties*, 15 Marine Technology 75 (January 1978).

6. J. Rahola, *Judging of Stability of Ships* (1939). The significance of Rahola's doctoral thesis is that it established that initial GM is an inadequate indicator of stability for certain types of vessels such as fishing vessels. His work has been incorporated into numerous stability standards, including the IMCO (now IMO) Recommendation on Intact Stability of Fishing Vessels (Resolution A.168). This standard appears in the U.S. Coast Guard NVIC 3-76, entitled "Stability of Fishing Vessels."

7. W.N. France, *The Professional Liability of Marine Designers and Constructors*, 18 Marine Technology 149 (April 1981); G.F. Chandler, III, *Professional Liability as it Relates to the Naval Architect/Marine Engineer (and Other Maritime Professionals*, 16 Marine Technology 4 (October 1979).

8. U.S. Const., Art. III, §9.

9. The Judiciary Act of 1789, §9.

10. One notable example is the Limitation of Liability Act, 46 U.S.C. §181-189, which in certain instances might permit an owner of a fishing vessel to reduce his liability arising from a casualty.

11. For instance, the federal appellate courts have reached differing conclusions on the question of whether asbestos litigation relating to ship construction is an admiralty matter. *Compare*, *Owens-Illinois, Inc. v. United States District Court*, 698 F.2d 967 (9th Cir. 1983), with *White v. Johns-Manville Corp.*, 662 F.2d 234 (4th Cir. 1981), *cert. denied*, 454 U.S. 1163 (1982).

12. *Kossick v. United Fruit Co.*, 365 U.S. 731 (1961); *The FRANCIS MC DONALD*, 254 U.S. 242 (1920); *People's Ferry Co. v. Beers*, 61 U.S. (20 How.) 129 (1860).

13. *New Bedford Dry Dock Co. v. Purdy*, 258 U.S. 96 (1922); *Point Adams Packing Co. v. Astoria Marine Construction Co.*, 594 F.2d 763 (9th Cir. 1979).

14. *Sieracki v. Seas Shipping Co.*, 149 F.2d 98 (3d Cir. 1945), *aff'd*, 328 U.S. 85 (1946) ("*Sieracki*").

15. *Executive Jet Aviation, Inc. v. City of Cleveland*, 409 U.S. 249 (1972) ("*Executive Jet*").

16. *Jig the Third Corp. v. Puritan Marine Insurance Underwriters Corp.*, 519 F.2d 171 (5th Cir. 1975), *cert. denied*, 424 U.S. 954 (1976).

17. *Sperry Rand Corp. v. Radio Corp. of America*, 618 F.2d 319 (5th Cir. 1980).

18. 618 F.2d at 321. *See* *Austin v. Unarco Industries, Inc.*, 705 F.2d 1, 10 (1st Cir. 1983).

19. R.D. Hursch and H.J. Bailey, *American Law of Products Liability 2d*, §1:1 (1974).

20. *LaRosa v. Scientific Design Co.*, 402 F.2d 937, 942-43 (3d Cir. 1968) (non-admiralty); *Avondale Shipyards, Inc. v. Vessel THOMAS E. CUFFE*, 434 F.Supp. 920, 929 (E.D. La. 1977).

21. This rule has its limits. *See* *Saloomey v. Jeppesen & Co.*, 707 F.2d 671 (2d Cir. 1983), holding that mass-produced navigational charts constitute "products" within the scope of products liability law.

22. *Little Beaver Enterprises v. Humphreys Railways*, 719 F.2d 75, 78 (4th Cir. 1983).

23. *Fischbach & Moore International Corp. v. Crane Barge R-14*, 632 F.2d 1123, 1125 (4th Cir. 1980).

24. W.L. Prosser, *Law of Torts*, §30 (4th ed. 1971).

25. Dunn v. Wheeler Shipbuilding Corp., 86 F.Supp. 659 (E.D.N.Y. 1949).

26. Dillingham Tug & Barge Corp. v. Collier Carbon & Chemical Corp., 707 F.2d 1086, 1092 (9th Cir. 1983).

27. The PENNSYLVANIA, 86 U.S. (19 Wall.) 125 (1873).

28. Pavlides v. Galveston Yacht Basin, Inc., 727 F.2d 339 (5th Cir. 1984).

29. Kelly, Inc. v. Bruno & Stillman Yacht Co., Inc.,1981 A.M.C. 104 (D.N.H. 1980).

30. McKay v. Rockwell International Corp., 704 F.2d 444, 447 (9th Cir. 1983), cert. denied, ___ U.S. ___, 104 S.Ct. 711.

31. "Seaworthiness" in a legal sense means that the vessel was sufficiently strong, staunch, and appropriately equipped to undertake its intended voyage or use. Mitchell v. Trawler Racer, Inc., 362 U.S. 539 (1960); Nygaand v. Peter Pan Seafoods, Inc., 508 F.Supp. 151 (W.D. Wash. 1981).

32. Petition of Risdal and Anderson, Inc. (the MIDNIGHT SUN), 248 F.Supp. 928 (D. Mass. 1966).

33. 248 F.Supp. at 931.

34. 248 F.Supp. at 932.

35. Cleary makes reference to this problem in his cited paper. See Reference 4, supra, at 233.

36. Storch gives a lucid analysis of the difference in those two methods in one of his cited papers. See Reference 5, supra, 15 Marine Technology at 77.

37. See Reference 4, supra, at 234.

38. An example of such a reduced standard appears in IMO Resolution A.207, also available as part of U.S. Coast Guard NVIC 3-76.

FISHING VESSEL SAFETY IN THE U.S.A.

WHAT ARE WE DOING?

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NATIONAL SEA GRANT COLLEGE PROGRAM

The National Sea Grant College Program Office hosted a national conference on Fishing Vessel Safety in November 1983. Its primary purpose was to bring together people who were working on Fishing Vessel Safety: and from this group, determine what needs could be addressed.

In attendance were Sea Grant College participants in Fishing Vessel Safety, primarily University of Rhode Island, University of Washington and University of Florida; U.S. Coast Guard; National Council of Fishing Vessel Safety; National Transportation Safety Board's Accident Division; Staff Members of the House Merchant Marine and Fisheries Subcommittee on Coast Guard and others.

Tasks that need to be addressed are:

(A) Review of current research, including that of the Coast Guard, the Society of Naval Architects and Marine Engineers, the Institute of Naval Architects, the Marine Index Bureau, NOAA through Sea Grant, other federal agency and foreign sources, where appropriate to determine what is being done and what remains to be done.

(B) Broadening the distribution of casualty reports, getting these to fishermen in a form that will be informative and useful.

(C) Investigating the possibility of equipping principal areas throughout the country with common visual aids, stability models and other devices that can make the fisherman aware of the potential safety problems and how to deal with them.

Active work by the National Council of Fishing Vessel Safety and Insurance, NFI is enlarging the knowledge base of fishing vessel safety.

Additionally, Coast Guard efforts to put its large depository of Fishing Vessel Casualty data to work is a heartening sign that productive effort to realize the need for education and knowledge of the need is forthcoming.

This conference is further addressing this very important issue.

The National Sea Grant Program has for over 10 years been involved in Fishing Vessel Safety via its Research and Extension arm. Research has been accomplished in several areas:

(Under operating conditions and - - - - - - - - - Vessel Stability
extreme sea conditions)
(Study of severity of ship motions - - - - - - Vessel Seakindlinees
in a specified sea spectrum)
Vessel Fuel Efficiency

Additionally, education of the user in vessel safety is being accomplished by Marine Extension Agents in several parts of the country.

Sea Grant support in this very important area has led to establishment of what are called Fishing Vessel Safety Centers. Two are in existance and a third is being formed; one is at the University of Rhode Island, Leader: Professor Tad Kowalski.

A feasibility study has been completed to establish a Fishing Vessel Safety Center at the University of Rhode Island for the New England area fsihermen. Ground rules, techniques for obtaining stability criteria, and an operating philosophy have been established. The Center has investigated four problem areas: vessel stability; vessel seakindlinees; vessel fuel efficiency: and, education in vessel safety. Through activities that provide much needed help to local fishermen, it is hoped the Center will achieve increased safety for fishing operations.

The second Fishing Vessel Safety Center is located in the Sea Grant Office at the University of Washington. Under Dr. Bruce Adee's capable management since 1978, a program of Research and Marine Extension activity has existed. This center has a traveling stability show and has traversed as far north as Alaska. Research in the Sea Grant Program as the University has been in stability and hull data. It is a very active enterprise on a very small budget.

The third center of interest lies here at FIT in Florida. Dr. Sainsbury has been funded some research in stability from Sea Grant and has started up an Advisory Committee to help determine where the needs of the Safety Center can best be applied.

The word center can be somewhat misleading. One could conjur up the image of a building with a sign outside saying Fishing Vessel Safety Center; walking in the door, a pretty receptionist would direct you to a manager who would answer all your questions about safety issues, show you demonstrations of stability, seakindliness, etc. with a "hot" line number.

This is not the case: rather three Sea Grant Regions are banding together their resources and knowledge to further the base of information on hand and needed to better educate the fishermen in this important area. One direct offshoot of this center concept is visibility of the need at least regionally.

The University of Florida has research in this area, also, and is developing the same concept in the Southeast.

Several other states are involved, from a marine extension viewpoint, in Fishing Vessel Safety i.e., Alaska, South Carolina, Maine, Texas, and Oregon.

The National Office position at this stage is to endorse the need, continue to endorse research that is competitive with other local and regional Sea Grant needs; motivate marine extension personnel to help their constituency to be sensitive to and aware of safety requirements, and join nationally with those leaders who view this problem as very important to further explore ways to educate and reduce the tragic accidents.

A vehicle presently envisioned is a committee of interested parties including Sea Grant, Coast Guard, National Council of Fishing Vessel Safety and Insurance and other interested parties.

We have called ourselves the Fishing Vessel Safety Activities Organization. There is no formal organizational set-up, but if the interest is great enough, I would certainly appreciate your inputs.

I don't think that we can pass this opportunity by. Lives are at stake and there are enough interested professionals around to band together and help reduce unnecessary hazards at sea.

Thank you for this opportunity to disucss my thoughts and count me in as one of your active players.

FISHING VESSEL INTACT STABILITY CRITERIA AND COMPLIANCE DUE TO VARIATION IN VESSEL DIMENSIONS

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Abstract

Fishing vessel safety is a problem of international concern. The intact transverse stability of fishing vessels has been an area of research. The present study examines the influence of small changes in a 90 ft. crab boat dimensions on the the lightship stability index. The index being taken as the GMT and GZ-θ curve of the intact vessel. Four variants are developed from the parent hull for the study. The stability calculations are summarized for these five hulls. Comparisons are made using three sets of GMT criteria and three sets of GZ-θ criteria.

1. Introduction to Problem

The safety of ships at sea is based on three factors:

I. The vessel design and compliance with safety standards.

II. The environmental conditions of sea waves, wind, icing, etc.

III. The judgment of the officers and crew operating the vessel.

The seasonal and regional nature of fishing has established numerous designs capable of operating in varied environments. The expansion of the fishing vessel fleet has caused a renewed interest on improvement of fishing vessel safety standards and their possible improvement /1/-/6/.

In the beginning of the 1983 tanner-crab fishing season, five crab boats capsized and 16 deaths were recorded /7/-/12/. These occurred to vessels operating in the Bering Sea out of Dutch Harbor, Alaska (Fig. 1). The problem of crab boat capsizing is a complex one due to the influence of the large number of crab pots carried on the deck, icing, as well as the free surface in the boat's tanks. In many cases these influences are controlled by factors II and III. Nevertheless, it is appropriate to examine factor I by studying a crab boat hull stability and compliance with available stability criteria.

Originally the study was to examine both a 60 ft. and 90 ft. vessel. However, it was necessary to limit the study to a 90 ft. vessel and four variants to finish the study as a class project /14/ in NA 402, Small Commercial Vessel Design, taught by the second author at the University of Michigan.

2. Nomenclature

B, B'	Molded beam
BM	Distance from center of buoyancy to metercenter
CB	Block coefficient CB = ∇/LBT
D	Molded depth to main deck
E	Restoring energy area under GZ-θ curve
FBD	Freeboard
GMT	Transverse metacentric height
GZ	Righting arm length
KB	Distance from keel to center of buoyancy
KG	Distance from keel to center of gravity
L, LBP	Length between perpendiculars
L FOCS	Length of Focs'le
LWL	Length of design waterline
T	Draft
Δ	Displacement
∇	Volume displacement
γ	Specific Gravity
θ	Heel angle

3. Background

Fishing vessel hull designs are evolved from general requirements of vessel capacity, crew size, operating time and speed. These translate into a total weight Δ of equivalent vessel displacement ∇. Using the block coefficient C_B vessel size and dimensions are related by

$$\Delta = \gamma C_B LBT \tag{1}$$

The influence of small changes in the vessel size can be estimated by rewriting equation (1):

$$\log \Delta = \log \gamma + \log C_B + \log L + \text{lob } B + \log T \tag{2}$$

and under the assumption γ = constant, C_B = constant, we obtain after differentiating:

$$\frac{d\Delta}{\Delta} = \frac{dL}{L} + \frac{dB}{B} + \frac{dT}{T} \tag{3}$$

For the case of constant displacement $d\Delta = 0$, equation (3) can be used to estimate the required changes in L, B and T /15/.

This type of redesign is easily done using a specially developed computer aided design system such as SPIRAL /16/. This or similar computer aided design systems, makes it relatively straight forward to examine the effect that small changes in a vessel's principal dimensions have on its stability. The stability being expressed in terms of static transverse stability GMT:

$$GMT = BM + KB - KG \tag{4}$$

and the righting arm GZ versus heel angle θ curves.

It becomes possible to illustrate qualitatively and quantitatively the impact on a fishing vessel's stability of small changes in its principal dimensions. This is done by using several sets of published stability criteria and examinig the degree of compliance of the parent and variant hull forms.

The vessel examined was a 90 foot crabbing vessel. The first author was very fortunate in being able to obtain lines and offsets for an existing vessel from a boatbuilder experienced in the design and construction of all sorts of fishing vessels. The parent hull was designed and built for operation in the harsh environment of the North Pacific Ocean. The stability of crab boats became the focus of attention when in early 1983, five similar vessels built by other boat builders were lost in the Bering Sea. In two of these accidents, there were no survivors. Fortunately, the full crew of the third boat escaped alive /7/-/12/. These tragedies remind us that the subject of ship's stability remains a problem.

4. Stability Calculation

4.1 Development of Four Variant Crab Boat Designs

The procedure used in this study is basically as follows: offsets for the parent crab boat hull were entered into the computer database. The preliminary design was then modified as follows:

1) Mod. A. Draft and depth were increased 10% while length was decreased 10%, for approximately constant displacement. This condition is called 1.1T in all figures and tables.

2) Mod. B. Draft and depth were decreased 10% while length was increased 10%. This condition is called 0.9T in all figures and tables.

3) Mod. C. Beam was increased 10% while length was decreased 10%. This condition is referred to as 1.1B in all figures and tables.

4) Mod. D. Beam was decreased 10% while length was increased 10%. This condition is called 0.9B in all figures and tables.

The parent crab boat hull offsets were modified using the ALTER Module of the SPIRAL computer system /16/. This input required for the ALTER Module included the parent hull offsets, dimensions, (L, B, T, D), and the desired dimensions of the variant. This resulted in five files containing the parent hull and four variant hull offsets. These five files were then processed into input data for the stability calculations using the VERIFY module of the SPIRAL system.

In all the calculations, the vessel displacement was assumed to be constant. This was found to be a reasonable assumption following equation (3) and the results of the displacement calculations made by the computer using the modified hull offsets. These computer calculations gave displacements within 5% of the parent hull displacement. The particulars of the parent and modified crab boat hulls are summarized in Table 1.

TABLE 1. VESSEL PARTICULARS

Crab Boat Study		Parent	Mod A 1.1xT	Mod B 0.9xT	Mod C 1.1xB	Mod D 0.9xB
Length LBP	ft	87.875	79.088	96.663	79.088	96.663
Beam B	ft	27.093	27.093	27.093	29.802	24.384
Draft T	ft	13.147	14.462	11.832	13.147	13.147
Depth D	ft	18.417	20.259	16.575	18.417	18.417
Displ.	tons	333.79	333.79	333.79	333.79	333.79
KG KG	ft	12.28	13.51	11.05	12.28	12.28

4.2 Calculation of GMT

The value of the parent and each variant GMT was estimated by the following formula developed from fishing vessel design data by the U. S. Maritime Administration /7/:

$$GMT' = B' \left(\frac{-LWL'}{400} + 0.185 \right) \qquad (5)$$

where GMT', B' and LWL' are in meters. In this and subsequent calculations LWL and T are taken at the design lightship conditions for the parent hull and corresponding variant conditions.

4.3 Righting Arm GZ - Angle of Heel θ Calculations

The GZ-θ curves for the parent and the four variant hulls were made using the SPIRAL computer system. These calculations were made using the STABLE module of the SPIRAL system. The input for the STABLE included the hull offsets, the data calculated from the VERIFY module, the desired waterline(s) and the angles of heel θ. Also the vessel's VCG obtained from boatbuilder. Here, the design waterline for the light ship condition was used.

Figure 2 shows the GZ righting arm curves for the parent and the four offspring hulls. As expected, when the beam is increased, or when draft is decreased, the righting arm, GZ, is larger than the parent hull GZ. This observation is valid up to angles of heel less than 40°. As the angle of heel approaches 90° the GZ values drop off sharply for the case of increased beam (or decreased length (Mod. C)). Another expected result was the loss in stability when draft is increased, and when beam is decreased. An unexpected result concerns the shape of the curves. In the conditions where the general dynamic stability is decreased, namely, when the beam is decreased (Mod D) and draft is increased (Mod A) the angle θ where the righting arm GZ is maximum is larger than the other curves. These GZ-θ curves appear to be more symmetrical than the other curves.

5. Stability Criteria

In order to get an idea of the sufficiency of the calculated stability the GMT and GZ-θ curves for the parent and four variants were compared to three sets of GMT criteria and three sets of GZ-θ criteria. More than one set of criteria was used to illustrate comparative differences. These differences mean that a vessel which has sufficient stability according to one set of criteria, could be deficient according to another criteria.

5.1 GMT Criteria

The following is a summary of the three static transverse stability GMT criteria:

1) Simplified IMCO Criterion. GMT must be greater than:

$$GMT = 1.7388 + (2B(GM1 + GM2)) \qquad (6)$$

where: $GM1 = 0.75 - (\frac{(0.37(FBD)}{B} + (0.82)\left(\frac{FBD}{B}\right)^2)$ and

$GM2 = -0.014(B/D) - 0.032\,(LFOCS/LWL)$

2) Japanese Criteria. GMT must be greater than the larger of:

$$GMT = \left(\frac{(B/3.2808) - 7}{12} + 0.4\right)(3.2808)\text{ ft.} \qquad (7)$$

and

$$GMT = \left(\frac{(L/3.2808) - 4.2}{72} + 0.4\right)(3.2808)\text{ ft.} \qquad (8)$$

3) Simplified Polish Criterion: GMT must be greater than:

$$GMT = D\left(0.105 - \frac{(0.706)(FBD)}{B} + (0.083)\left(\frac{B}{D}\right)\right).\ \text{ft.} \qquad (9)$$

5.2 GZ-θ Criteria

The following is a summary of the three sets of stability criteria applied to the vessel GZ-θ curve.

1. Rahola's Criteria.

I. The maximum righting arm must be greater than 0.656 ft.

II. The area under the righting arm curve to maximum GZ must be larger than 15.04 feet-degrees.

III. The angle where the righting arm is maximum must be larger than 30°.

2. U. S. Criteria.

I. The minimum area under the righting arm curve to 40° is 16.9 feet-degrees.

II. The minimum area under the righting arm curve from 30° to 40° is 5.6 feet-degrees.

III. The angle where the righting arm is maximum must be larger than 25°.

IV. The righting arm must be positive at a heel of 30°.

3. IMCO Criteria.

I. The minimum area under the righting arm curve to 30° is 10.3 feet-degrees.

II. The minimum area under the righting arm curve to 40° is 16.9 feet-degrees.

III. The minimum area under the righting arm curve between 30° and 40° is 5.6 feet-degrees.

IV. The angle where the righting arm is maximum must be larger than 25°.

V. The minimum righting arm at any angle over 30° is 0.656 ft.

6. Discussion of Calculated Stability Parameters and Stability Criteria

The comparisons of the calculated stability parameters and the various stability criteria described in Sections 5.1 and 5.2 are summarized in Tables 2, 3, 4 and 5.

6.1 GMT Comparisons

Table 2 shows the results of the calculations of actual metacentric height GMT and the GMT required by each criteria. The results for the parent design and the four variants are summarized in this Table. The changes in GMT for the different modifications are

TABLE 2. COMPARISON OF REQUIRED AND CALCULATED GMT VALUES

Crab Boat Study	Parent	Mod A 1.1xT	Mod B 0.9xT	Mod C 1.1xB	Mod D 2.6 ft
Static Stability GMT Calculated	3.16 ft	3.35 ft	2.98 ft	3.68 ft	2.68 ft
GMT Required by IMCO, ft	1.52	1.32	1.68	1.79	1.53
Satisfied?	Yes	Yes	Yes	Yes	Yes
GMT Required by Japanese Regulations, ft.	1.48	1.48	1.48	1.59	1.37
Satisfied?	Yes	Yes	Yes	Yes	Yes
GMT Required by Polish Regulations, ft.	2.68	2.78	2.92	3.24	2.57
Satisfied?	Yes	Yes	Yes	Yes	Yes

TABLE 3. COMPARISON OF GZ-θ CALCULATION WITH RAHOLA'S STABILITY CRITERIA

Crab Boat Study	Parent	Mod A 1.1xT	Mod B 0.9xT	Mod C 1.1xB	Mod D 0.9xB
I) GZ_{max} Calc. Ft.	2.11	1.68	2.18	2.15	1.57
GZ_{max} 0.656 ft. Satisfied?	Yes	Yes	Yes	Yes	Yes
II) E ft.-deg. to θ at GZ_{max}	53.53	46.83	63.73	59.37	43.91
E 15.04 ft-deg?	Yes	Yes	Yes	Yes	Yes
III) θ at GZ_{max} deg	46^o	49^o	40^o	40^o	48^o
θ 30°?	Yes	Yes	Yes	Yes	Yes

TABLE 4. COMPARISON OF GZ-θ CALCULATION WITH U.S. STABILITY CRITERIA

Crab Boat Study	Parent	Mod A 1.1xT	Mod B 0.9xT	Mod C 1.1xB	Mod D 0.9xB
I) E to $\theta=40^{\circ}$, ft. - deg.	53.5	42.57	63.70	59.4	39.4
$E \geq 16.9$ ft deg?	Yes	Yes	Yes	Yes	Yes
II) θ at GZ_{max} deg	46	49	40	40	40
$\theta > 25$ deg?	Yes	Yes	Yes	Yes	Yes
III) E between $30 < \theta < 40^{\circ}$, ft - deg	18.3	14.6	21.2	33.6	13.9
$E_{40^{\circ}}-E_{30^{\circ}} \geq$ 5.6?	Yes	Yes	Yes	Yes	Yes
IV) GZ at $\theta=60^{\circ}$ ft	1.88	1.54	1.89	1.67	1.48
GZ 0 ft	Yes	Yes	Yes	Yes	Yes

TABLE 5. COMPARISON OF GZ-θ CALCULATION WITH IMCO STABILITY CRITERIA

Crab Boat Study	Parent	Mod A 1x1xT	Mod B 0.9xT	Mod C 1.1xB	Mod D 0.9xB
I) E to $\theta=30^{\circ}$ ft deg	35.2	27.9	42.5	25.8	25.5
$E \geq 10.3$ ft deg?	Yes	Yes	Yes	Yes	Yes
II) E to $\theta=40^{\circ}$ ftdeg	53.53	42.57	63.7	59.4	39.4
$E \geq 16.9$ ft deg?	Yes	Yes	Yes	Yes	Yes
III) E between $30^{\circ} \leq \theta \leq 40^{\circ}$ ftdeg	18.3	14.6	21.2	33.6	13.9
$E_{40^{\circ}}-E_{30^{\circ}} \geq 5.6$?	Yes	Yes	Yes	Yes	Yes
IV) θ at GZ_{max} deg	46	49	40	40	48
$\theta > 25^{\circ}$?	Yes	Yes	Yes	Yes	Yes
V) For $\theta \geq 30^{\circ}$ $GZ \geq 0.657$ ft?	Yes	Yes	Yes	Yes	Yes

interesting. As intuitively expected, GMT varies with beam changes. It ranges from 2.68 feet for the smallest beam (Mod D) to 3.68 feet for the largest beam (Mod C). Thus a 20% variation in beam is responsible for a 1 foot increase in GMT. As for the differences in the required GMT, in every case, the Japanese and the IMCO Simplified Criteria yield similar results. The Simplified Polish Criteria is consistently stricter with required GMT values almost twice those of other criteria. With the parent vessel at lightship condition, all GMT requirements are met. However, in two of the offspring cases, the Polish GMT Criterion is met by less than a 0.25 ft. margin.

6.2 GZ-θ Comparisons

Tables 3, 4 and 5 present the comparisons for the GZ-θ calculations and GZ-θ criteria described in Section 5.2. In the criteria the energy E represented by the area under the GZ-θ curve free from 0 to some θ is given by:

$$E = \int_0^{\theta} GZ d\theta \text{ ft. - deg.} \tag{10}$$

The integral was evaluated numerically using either Simpson's first or second rule using the GZ-θ results from Fig. 2.

Table 3, indicates the parent and the four variant hulls meet the Rahola's stability criteria. Table 4 shows an equally large margin between the U.S. criteria and calculated GZ-θ values. The IMCO criteria has five separate requirements which is the largest number to be satisfied. As Table 5 illustrates, the calculated GZ-θ curves for the parent and four variants adequately meet these IMCO criteria.

It should be noted that, even in the most severe conditions, all criteria requirements are met. However, these calculations assume a lightship condition. The operating vessel will have a considerably higher center of gravity due to the large weight of the crab pots and other deck equipment. In addition slack fuel, water, and crab tanks will result in a free surface effect which could cause a substantial loss of stability index given either by GMT or GZ. It remains as a future task to quantify these influences.

The main benefits of this study are of illustrative value. First, it has presented the comparative severity of several sets of stability criteria. Secondly, it has illustrated the effects that small design changes have on a vessel's stability. While the present study involved a number of factors, it is in no way complete. The authors recommend continued work on this topic. One possible study would be to perform similar calculations on a series of fishing vessels and develop appropriate regression equations to predict changes in a vessel's stability (i.e., area under the GZ curve) caused by dimensional changes during the preliminary vessel design. Such information could be an invaluable tool for fishing boat designers, and would contribute to preventing future loss of life at sea.

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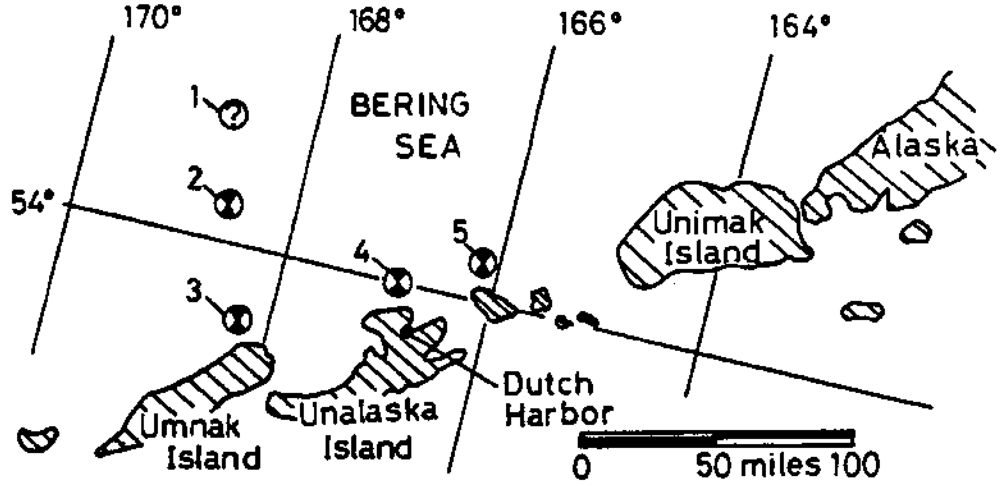
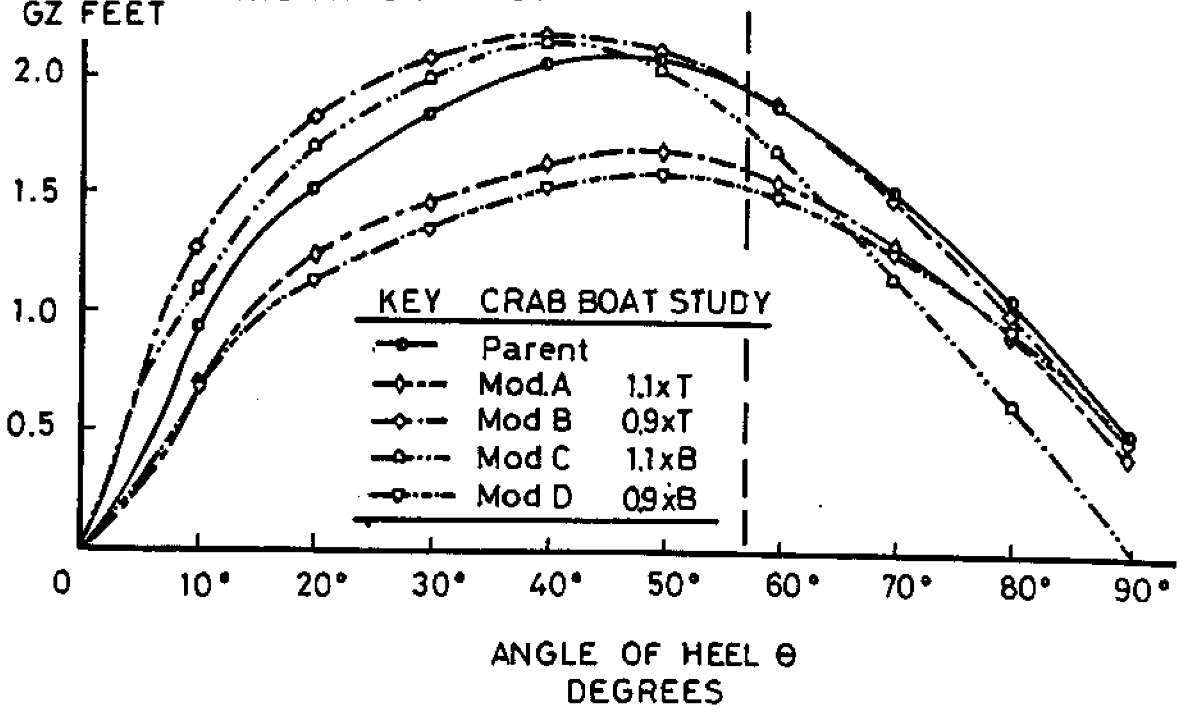
FIG.1 LOCATION OF CRAB BOAT CAPSIZINGS /12/



KEY	VESSEL	DEATHS
1	Altair	7
2	Americus	7
3	Sea Hawk	1
4	Artic Dreamer	0
5	Fly Boy	1
	1983 Feb-March	16

38 Dead : 1/80 - 3/83

FIG. 2 COMPARISON OF CALCULATED RIGHTING ARM CURVES GZ-θ



MODEL TESTS OF THE DYNAMIC RESPONSE OF A FISHING VESSEL

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Abstract

Testing of a scale model of an Alaska king-crab fishing vessel in the natural environment of a fresh water lake is described. The wave environment was determined using an array of capacitance wave staffs mounted on a mobile wave measuring platform. During the tests, the dymanic response of the model was measured for various vertical center of gravity locations and model headings relative to the predominant wave direction. The time series were analyzed and response and wave spectra computed. The model testing program has also provided an opportunity to observe the capsize phenomena at first hand and learn how an operator should respond to a potential capsizing situation.

Introduction

The fishing vessel model testing program under way at the University of Washington has many facets. It provides an opportunity to study the dynamic response of a fishing vessel model in a wind driven sea. The model response is highly complex and difficult to predict using current ship-motion theories. The model testing program provides voluminous quantitative data and a growing intuitive data base of qualitative information based on observations. In the long run this should lead to better prediction of vessel dynamics and to correlation between the data measured in the natural environment and model basin.

The phenomena associated with capsizing may be observed and recorded. By testing a series of models the applicability of the International Maritime Organization standards may be evaluated and devices such as stability alarms tested.

Test Site

Lake Washington is situated close to the University of Washington campus in Seattle. The lake is over 15 nautical miles long in the north/south direction and 1 to 2 nautical miles wide in the testing area. Figure 1 shows the portion of the lake used for testing. It is bounded on the south by the Evergreen Point Floating Bridge which limits the fetch to about 2.3 miles for the predominant southerly winds during the testing season. Appropriate wind speeds for testing are in the range of 8-17 knots. Winds above this range cause model motions which are too severe for the sensors to measure properly.

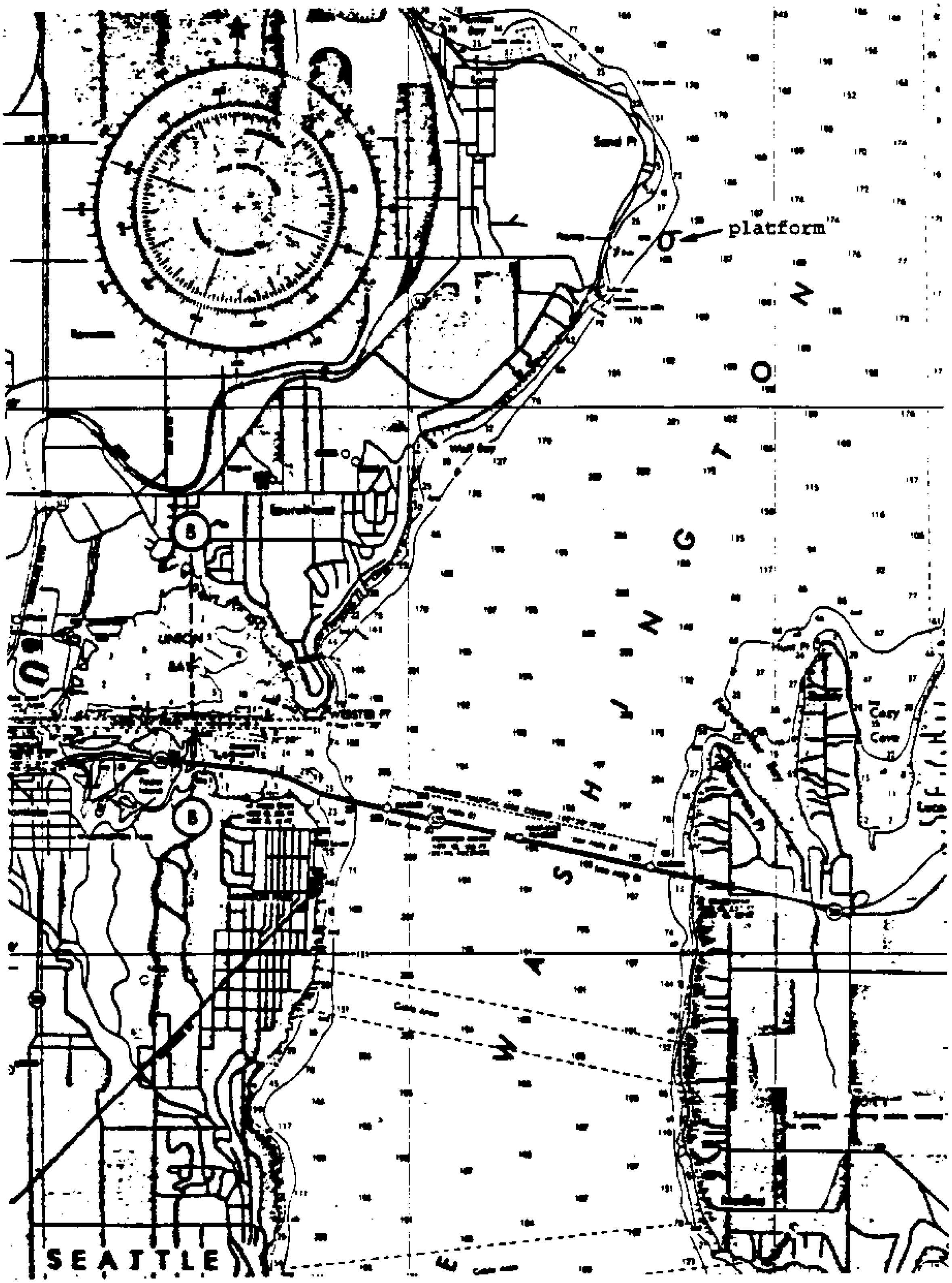


Figure 1. Model testing location.

Wave Measurement

The waves encountered in the test area are typical of fetch limited regions. The waves are measured using an array of six capacitance wave staffs mounted on the Ocean Engineering Mobile Wave Measurement Platform. The platform, shown in Figure 2, is of a semi-submersible design and is held on station with anchor lines extending from each of the four corners. For the winds and waves encountered at the testing site, the platform is extremely stable with very little motion in response to the waves.

On the platform, data acquisition is controlled by a PDP 11/2 microcomputer which samples the data from the six wave staffs as well as wind speed and direction data from an anemometer. The digital data is stored on cassette tapes at a sampling rate of 5 Hertz for later analysis. During testing, a record of 1024 points for each channel is taken at ten minute intervals.

Figure 3 shows the averaged wave spectrum measured on 23 March 1984.

The major reason model testing is performed in the open water on Lake Washington is to examine the dyamic response of the fishing vessel models to wind generated waves. In order to properly interpret the measured motion response it is necessary to compute the directional wave spectrum. For this purpose, the modified maximum likelihood method developed by Regier [1] was used. This method offers significantly improved resolution over the Fourier series method [2]. The directional wave spectrum for 23 March 1983 is plotted in Figures 4, 5 and 6.

The Model

The experiments were conducted using a self-propelled, radio-controlled model of a king-crab fishing vessel. The model was constructed of wood with plexiglass decks which are completely sealed. Particulars of the model are given in Table 1 and the lines are shown in Figure 7.

Table 1. Fishing vessel and model dimensions

PROPERTIES	FULL SCALE (FT)	MODEL (FT)
Scale Ratio	11.16	1.00
Length Overall	91.00	8.15
Length Waterline	85.00	7.62
Beam at Main Deck	25.75	2.31
Design Draft	12.00	1.08
Displacement	374.30 tons	603.0 lb
Moveable weight position 2		2.0 in above datum
Moveable weight position 3		4.0 in above datum

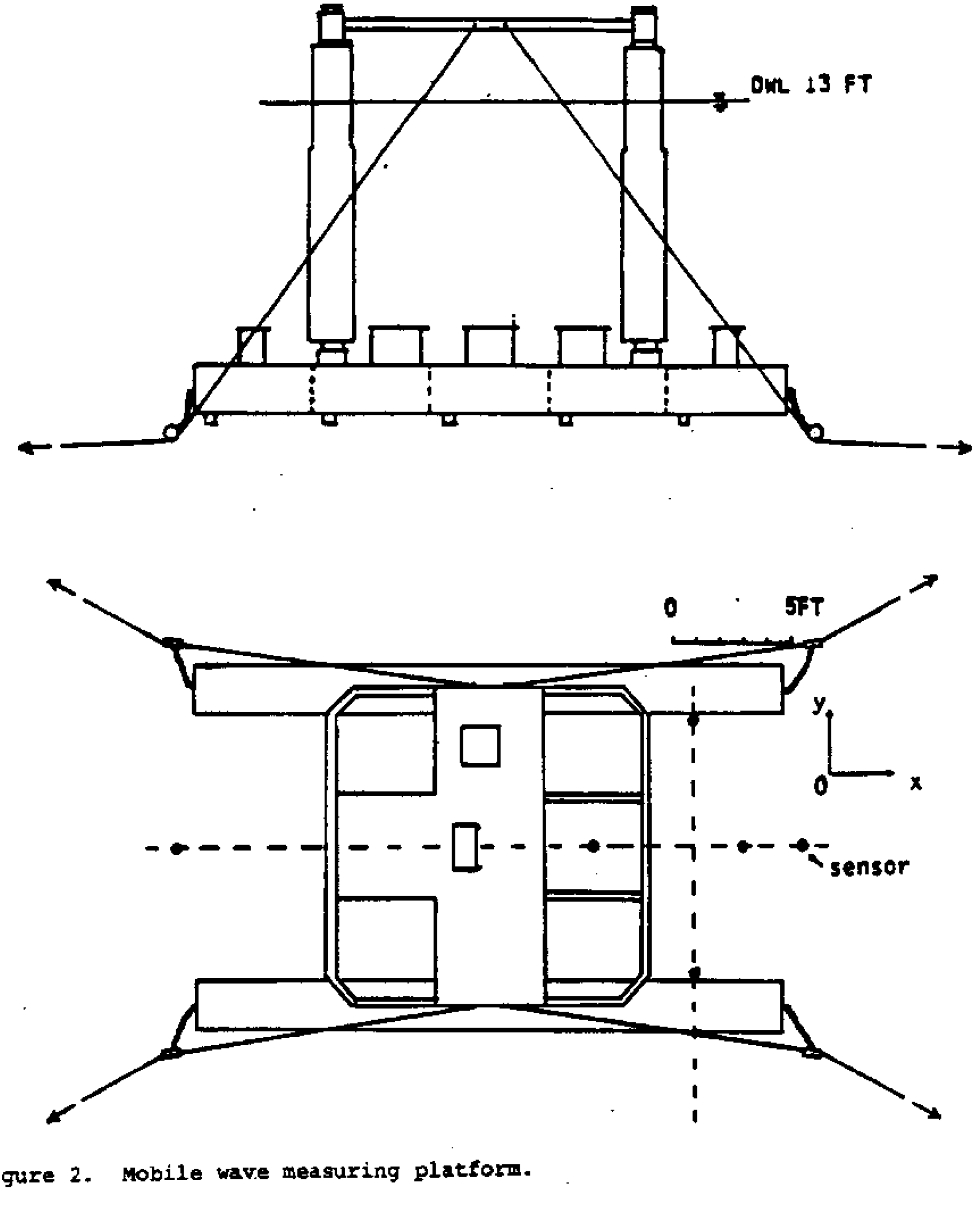


Figure 2. Mobile wave measuring platform.

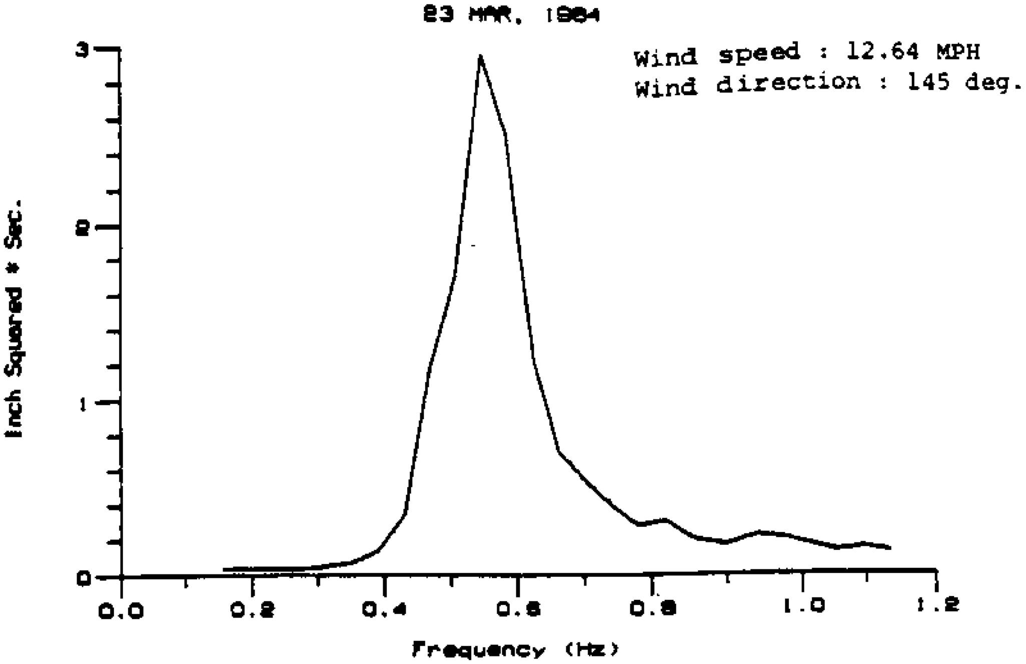


Figure 3. Averaged wave spectrum measured 23 March 1984.

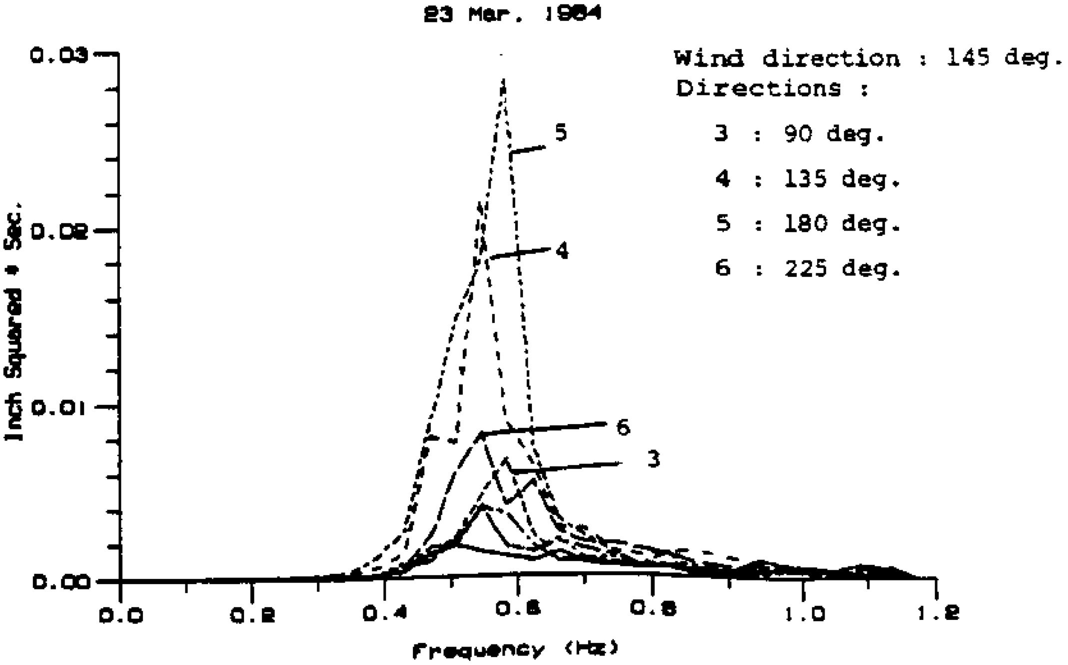


Figure 4. Directional wave spectrum measured 23 March 1984.

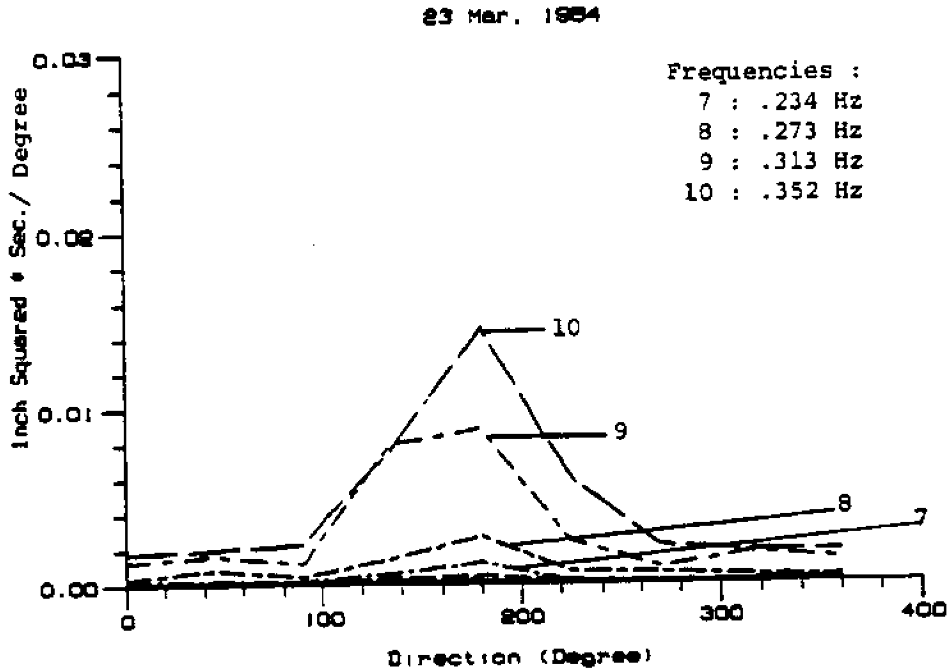


Figure 5. Directional wave spectrum measured 23 March 1984.

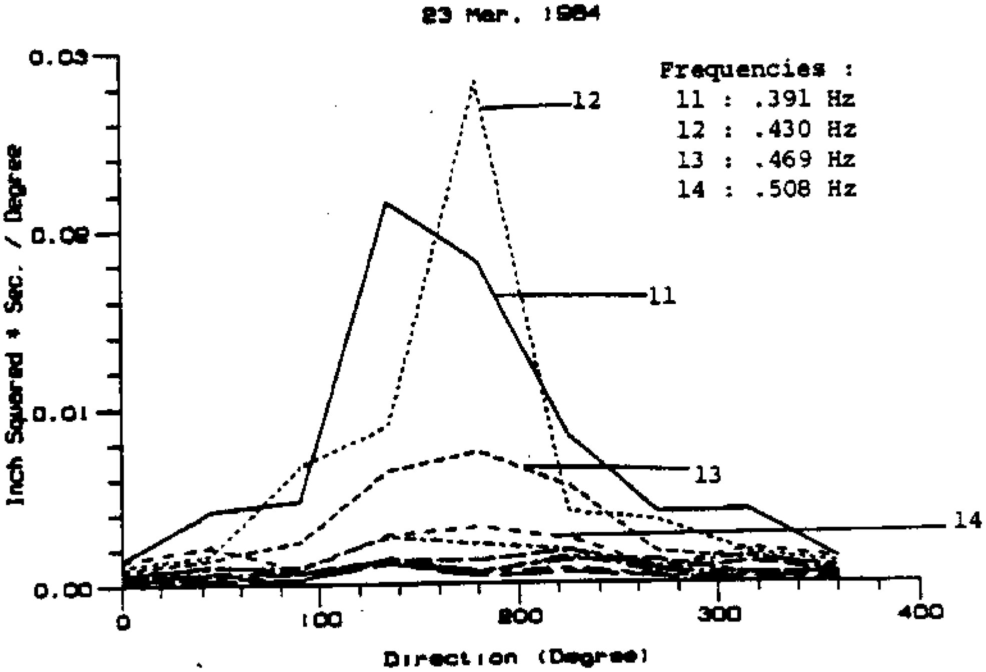


Figure 6. Directional wave spectrum measured 23 March 1984.

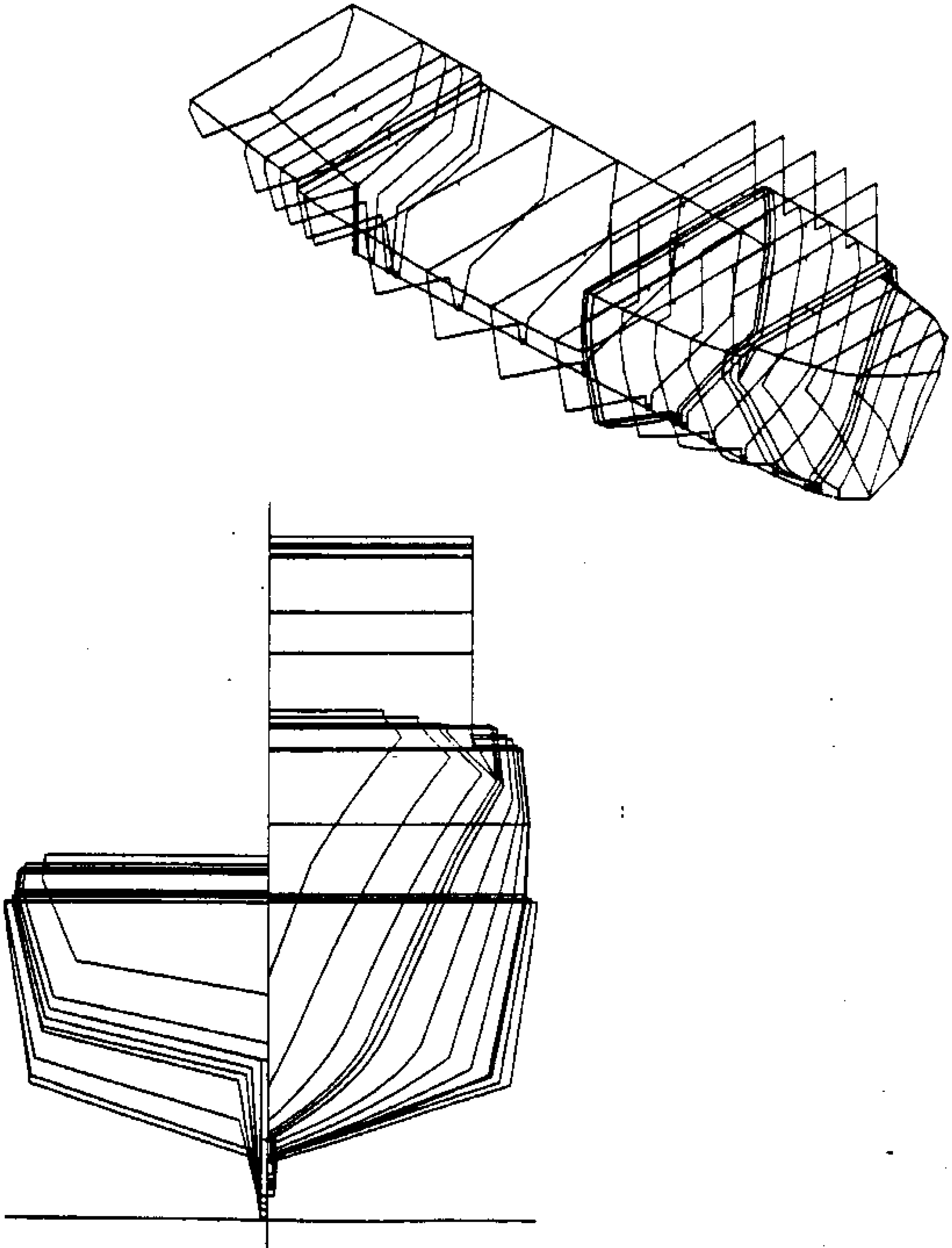


Figure 7. Body plan of fishing vessel model.

The weight and center of gravity of the model are controlled by the placement of internal equipment and lead ballast. In the deckhouse area a 78.7 pound moveable lead weight is mounted. This weight may be moved up or down using a switch mounted on the outside of the deckhouse. Once the proper fore and aft trim is established by placing the fixed ballast, the vertical center of gravity is altered by changing the position of the moveable weight.

Power for the model is provided by rechargeable gell cell batteries. Separate circuits are provided for the main engine and for the instrumentation.

Data acquisition is under the control of a PDP 11/2 microcomputer. Currently ten of the sixteen available analog to digital channels are used during the experiments. Digitized data is recorded on cassette tapes for later analysis in the laboratory. The data channels include:

1. roll angle
2. pitch angle
3. heave acceleration
4. sway acceleration
5. surge acceleration
6. heading angle (2 channels)
7. rudder angle
8. vertical position of moveabale weight
9. motor RPM.

Model Testing Procedure

On a day when testing is to be performed, the first step is to set up the instruments on the wave measuring platform and start the data acquisition system. The model is launched at a public boat launching ramp and then towed to the test site in the vicinity of the platform where the model's electronics are turned on.

For a given vertical center of gravity position, tests at five different headings relative to the predomionant waves are run. They are:

1. head seas
2. bow quartering seas
3. beam seas
4. stern quartering seas
5. following seas.

These directions are all selected by the person at the helm of the radio control. During each test 1024 data points are recorded at a sampling rate of 5 Hertz. When a set of 5 tests are completed the vertical center of gravity position is changed and another set of 5 tests are run.

If the model capsizes during a test, the vertical gyro is caged and the computer allowed to complete data acquisition for the run. The gyro is then uncaged and the remaining runs completd. If possible (sufficient battery power and space on the data tape), the test during which the model capsized is repeated.

Observations

Figures 8 and 9 show the time histories of two model tests in stern quartering seas on the same day. For the test shown in Figure 8 the moveable weight is located 2 inches above the datum position and for Figure 9 the weight was 4 inches above the datum position. Both of the capsizings were very sudden. Neither was preceeded by a slowly increasing amplitude of roll. From the rudder movement throughout the tests it is quite easy to conclude that attempting to maintain a steady heading under these conditions is very difficult. If the controls are fixed and not used, the model will quickly broach in either following or stern quartering seas.

The observed sequence of events in these capsizings appears to be:

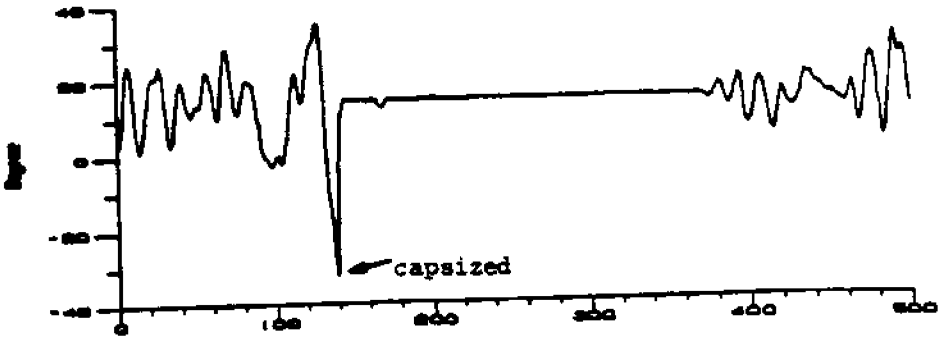
1. The wave overtakes the model from astern and comes up under the stern.
2. This lifts the stern causing trim by the bow and an increase in speed.
3. Because the wave energy is not perpendicular to the centerline of the model, the wave striking either the port or starboard side of the stern causes the model to yaw.
4. As the wave crest moves amidship the model momentarily loses stability and rolls to a large angle while continuing to yaw.
5. When the crest approaches the forward end of the model, the wave force tends to restore the model back toward its original heading. With the crest moving forward the model's stability also increases and the model then begins to roll in the other direction. By this time a rudder correction has also been made in an attempt to return to the orginal heading.
6. The next wave then hits the model, but this time the model has an initial yaw and roll velocity and it will roll to the opposite side (when compared with the initial large roll) and will capsize.

Even when operating in a low stability condition, capsizing was not extremely frequent. In most cases, the model is excited as described above by a group of steep waves, but the wave group passes before the sequence leading to capsize is completed. There are far more "near misses" than actual capsizings.

For the tests when the model was operating with low stability, capsizing also occurred during maneuvering . This was particularly true as the model completed a test in beam seas and was turning to prepare for a following seas or stern quartering seas test.

It is interesting to note that at no time during the testing has the model capsized in beam, head or bow quartering seas. This may be

(a) Roll Angle



(b) Rudder Angle

(c) Pitch Angle

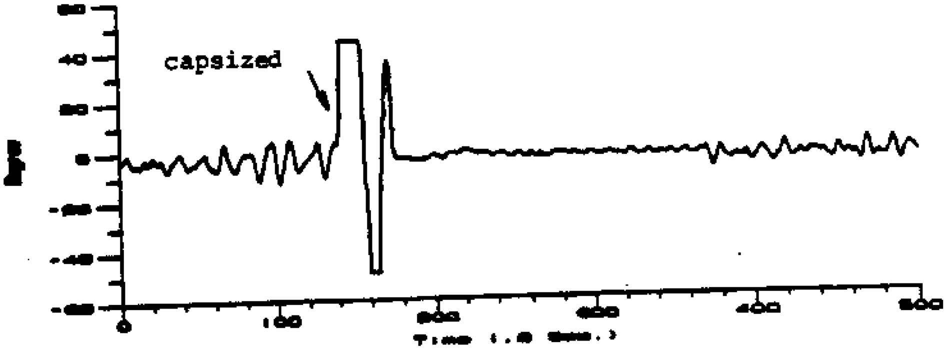
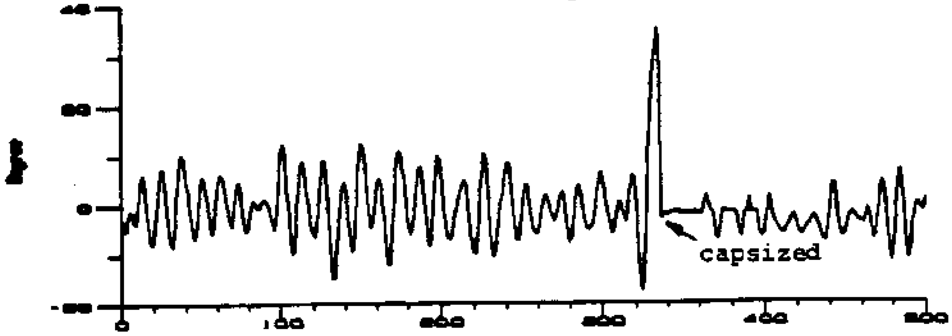
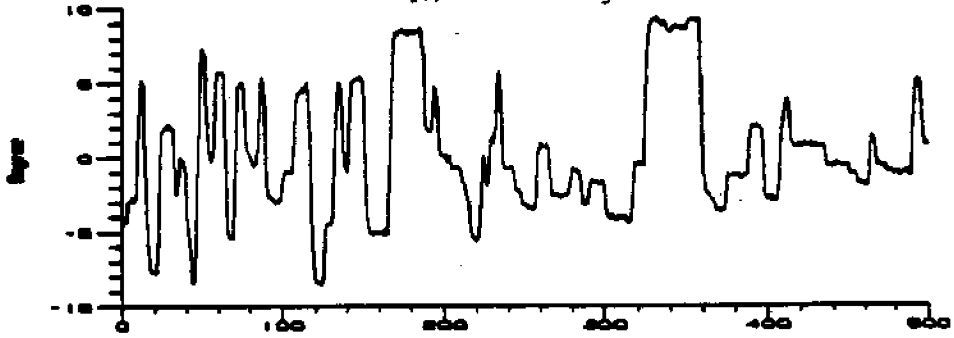


Figure 8. Roll, pitch and rudder angle time series 23 March 1984 at weight position #2, stern quartering sea.

(a) Roll Angle



(b) Rudder Angle



(c) Pitch Angle

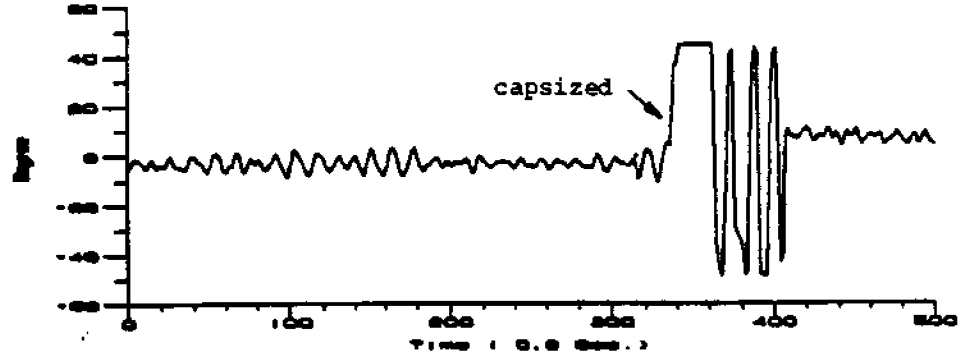


Figure 9. Roll, rudder and pitch angle time series 23 March 1984 at weight position #3, stern quartering sea.

due to the fact that no bulwarks are installed. Others have observed that capsizing in those seas is closely tied to the additional problem of operating with water on deck.

The computed spectra for roll, pitch and heave response are shown in Figure 10, 11 and 12, respectively.

It was surprising that the rolling response in following and stern quartering seas was considerably greater than in beam seas. The peak at about 0.4 Hertz is associated with the natural frequency in roll (slightly lower for the less stable weight postion 3). In Figures 10,11 and 12 the frequency scale is actually frequency of encounter which explains why this peak is at a lower frequency for following seas when compared with beam seas.

The second peak in the roll spectra between 0.1 and 0.2 Hertz was not expected. A possible explanation is that this is related to the presence of water on deck. The total load for the model in these tests was very heavy leaving a minimum freeboard. As a result, water would frequently come onto the deck. In two-dimensional model tests conducted in a wave tank it was clearly established that the effect of water on deck was much more pronounced in lower stability conditions. The difference between the the roll response spectra is also clearly evident in the time series for these tests.

The pitch and heave response spectra shown in Figures 11 and 12 are about as expected. The shift in the frequency of the maximum response is a result of plotting frequency of encounter.

Conclusion

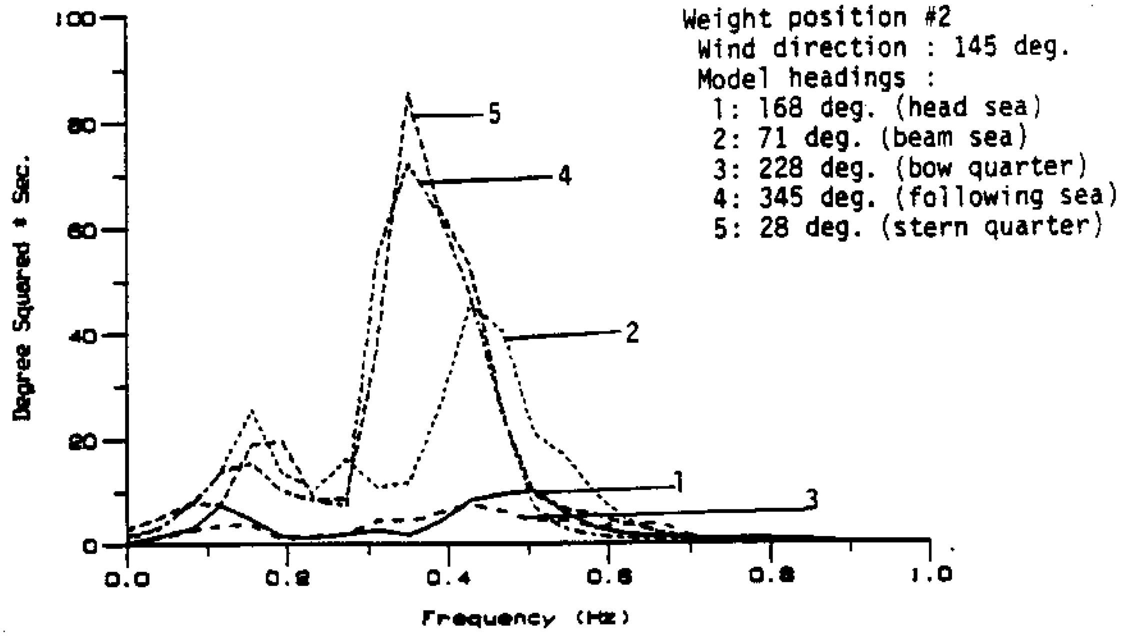
Similar type model tests performed on a model of a fast cargo liner in San Francisco Bay[3] lead to similar conclusions regarding the dangers of operating in stern quartering and following seas.

We are beginning to learn a great deal more about the dynamic response of fishing vessels from the model testing program. As additional models are constructed and tested the program should prove valuable for the data produced and the study of relatively rare phenomena which can be frequently reproduced.

Acknowledgement

The support of the Washington Sea Grant Program under a grant from the National Oceanic and Atmospheric Administration is gratefully acknowledged. Their support of the Fishing Vessel Safety Center at the University of Washington has made this paper possible. The model used in these tests is on loan from the U. S. Coast Guard. Duwamish Shipyard in Seattle has provided the use of a support vessel during the tests. The assistance of several naval architects has also been very valuable.

(a) 23 Mar. 1984



(b) 23 Mar. 1984

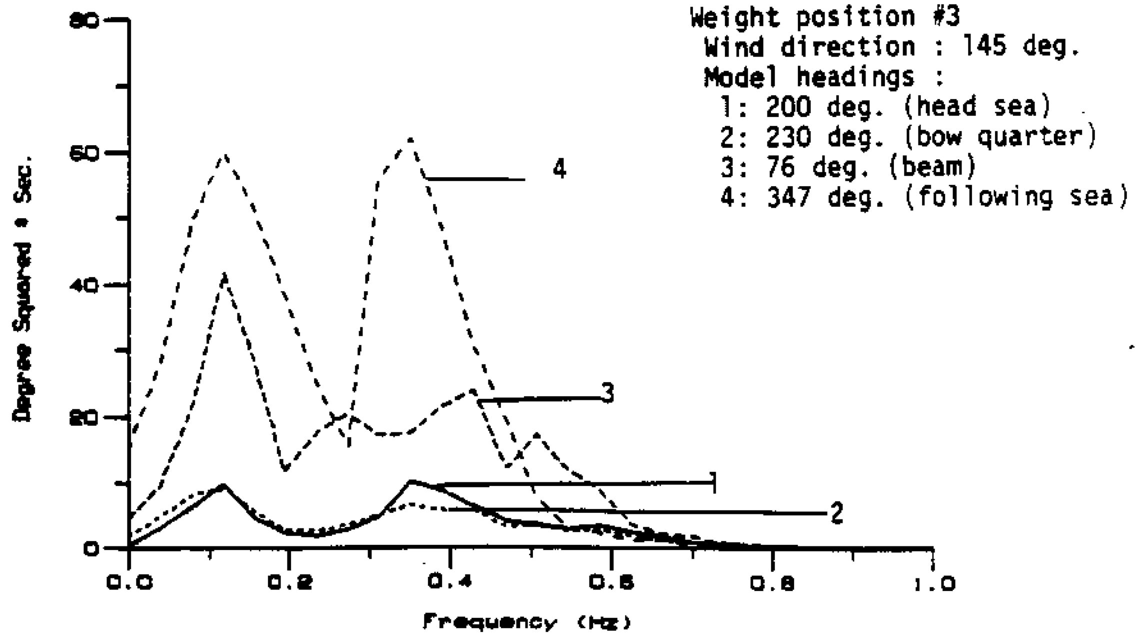
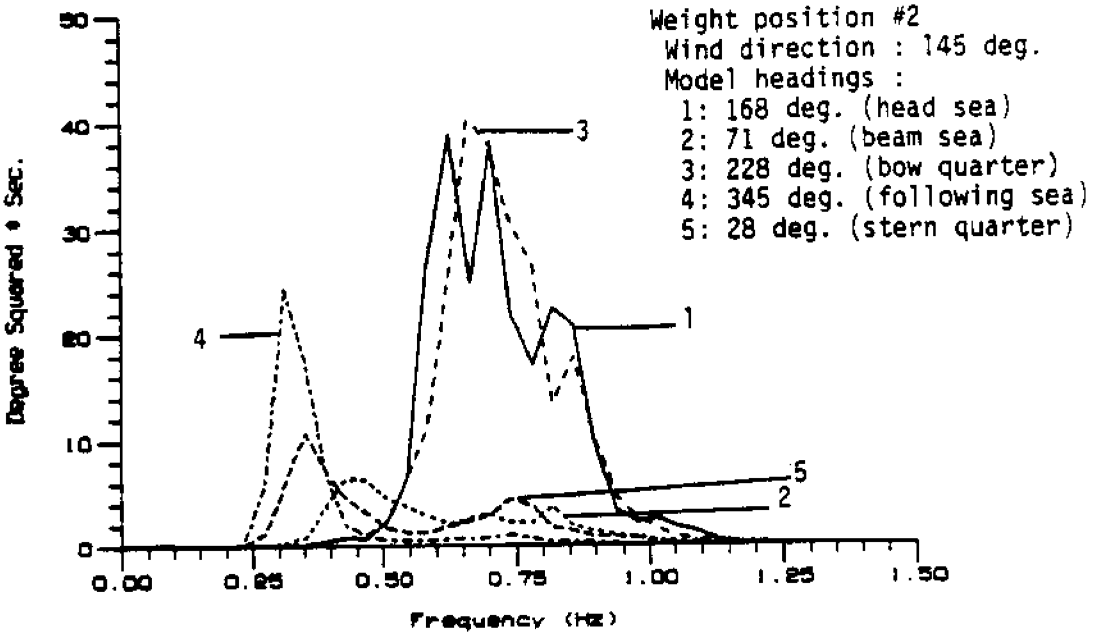


Figure 10. Roll spectra 23 March 1984.

(a) 23 Mar, 1984



(b) 23 Mar, 1984

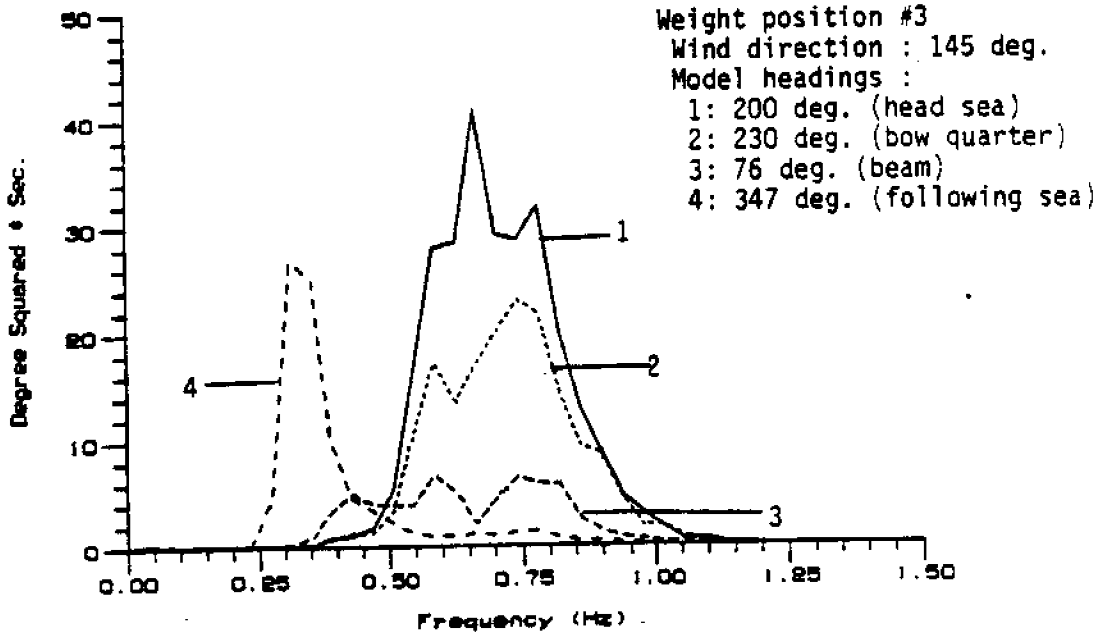
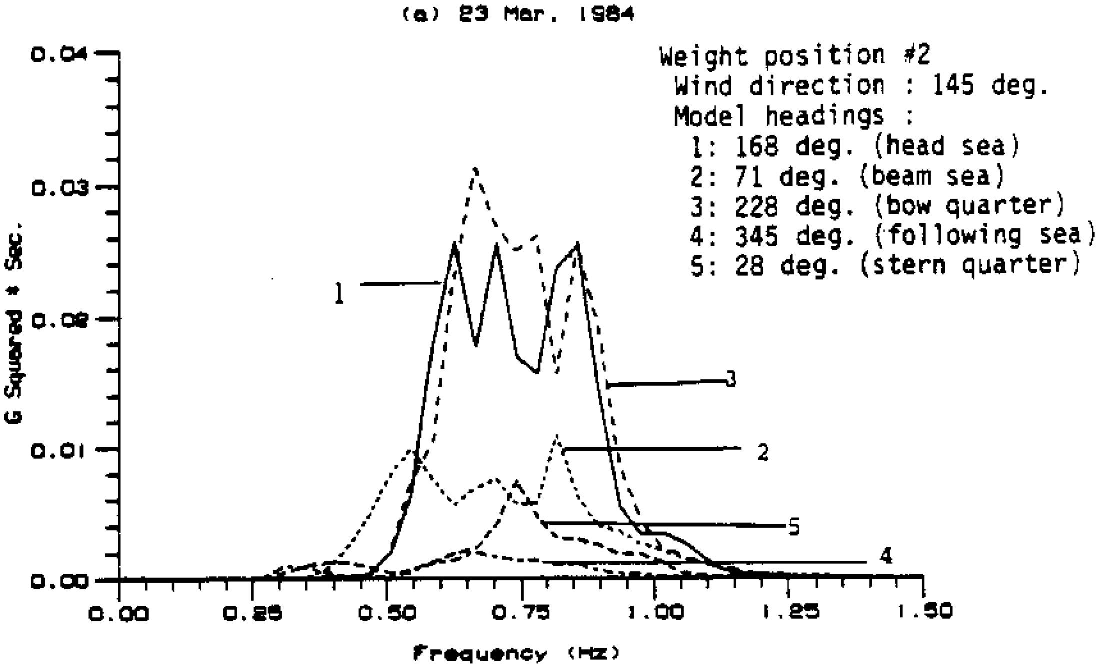


Figure 11. Pitch spectra 23 March 1984.


(a) 23 Mar. 1984

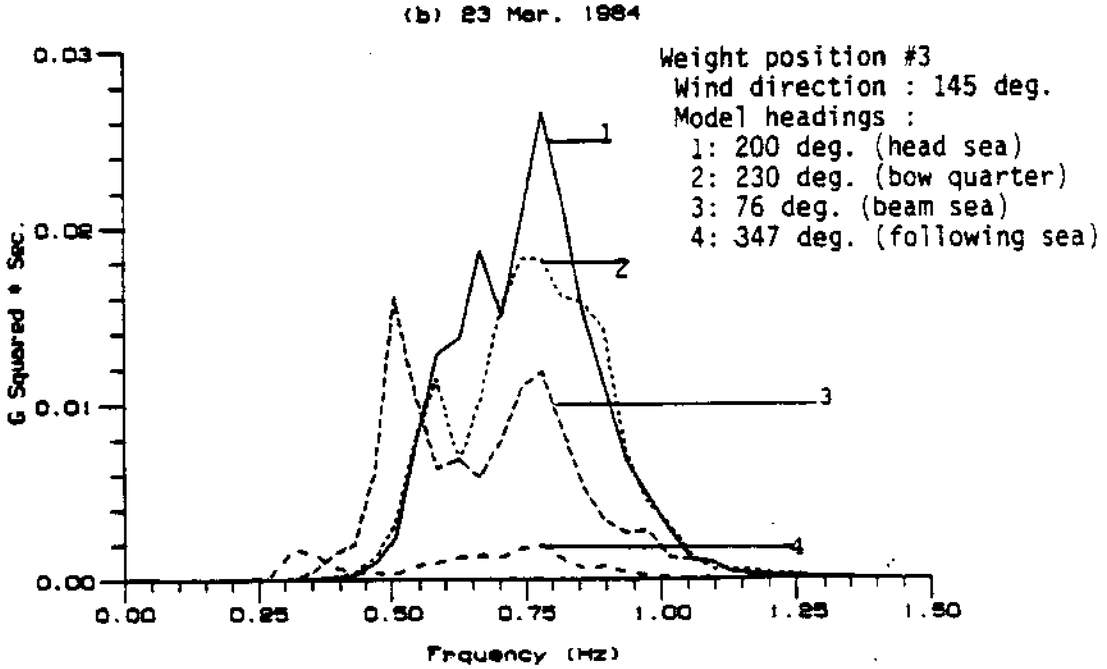

(b) 23 Mar. 1984

Figure 12. Heave spectra 23 March 1984.

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THE EFFECT OF LOADING ON FISHING VESSEL STABILITY: A CASE STUDY

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Abstract

Scallop vessels operating out of Port Canaveral, Florida are susceptible to stability problems due to the adverse loadings imposed on them. The problem is that the boats are loaded with up to 30 tons of scallops on the main deck, instead of carrying the catch in the hold. This paper will address the problem by analyzing one typical vessel throughout an entire fishing trip. Different loadings will be tested and the critical conditions will be identified. Recommendations for improving the stability to some extent will be given.

Introduction

Scallop vessels operating out of Port Canaveral, Florida are susceptible to stability problems due to the adverse loadings imposed on them. There were five capsizings in the years 1982 and 1983. Most of the casualties occurred in the port just prior to the unloading of the vessels.

The number of capsizings has prompted the insurance industry to provide only one underwriter to insure vessels for scalloping. The premium for scallop vessels is about 6 1/2 percent of hull value, as compared to 3 - 5 percent for shrimp boats. Actually the scallop fleet consists mainly of converted shrimp trawlers. The problem is that current operating procedures in the scallop fishery dictate that the catch be carried on the deck, instead of in the hold. This is in order to facilitate the method of unloading employed, which is to use a mechanical clam-shell shovel and scoop the scallops off the deck. The heavy deck load (as much as 30 tons) raises the vessel's center of gravity and imposes a large trim by the stern, resulting in decreased stability for the boat. In addition, the short duration of trips (about 24 hours) precludes the need for the operators to carry much fuel. The vessel's center of gravity could be lowered substantially with increased fuel.

The goal of this paper is to display the stability conditions of a typical boat during all stages of a trip. These conditions may then be compared, and the critical loadings when the vessel is in a dangerous state can be determined. Suggested alternative loadings would then be evaluated as possible solutions to the stability problem.

The test vessel picked for analysis is shown on Figure 1. It is a 75 ft. steel, hard-chined trawler. It is known that

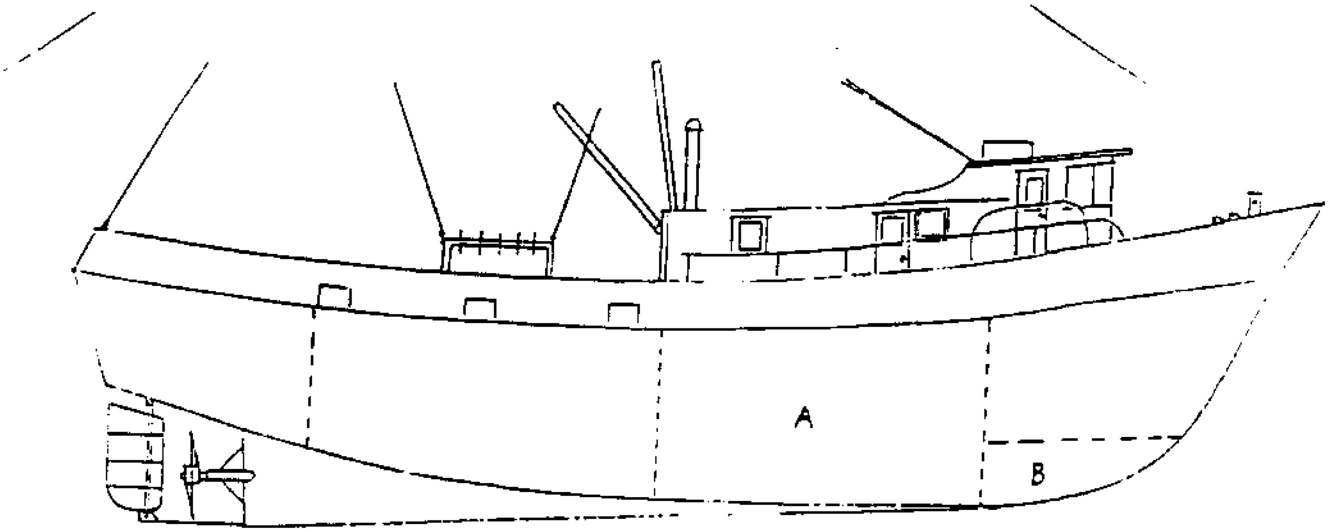


FIGURE 1

TEST VESSEL
75' steel, hard-chined shrimp trawler.
Tank A.....Fuel Wing Tanks (8200 gals.)
Tank B.....Water Tank (2000 gals.)

there are 17 sister ships and a number of similar vessels at the port, so the selection of this boat represents a large percentage of the fleet (about 30%). The following table lists some of the principle parameters of the test vessel:

Overall length	75 ft.
Length BP	66 ft.
Breadth	20 ft.
Depth	10.5 ft.
LS Displacement	108.5 tons
Fuel capacity	14,000 gals.
(8200 in wing tanks, 5800 in stern tanks)	
Potable water	2000 gals.
(2000 in forward tank)	

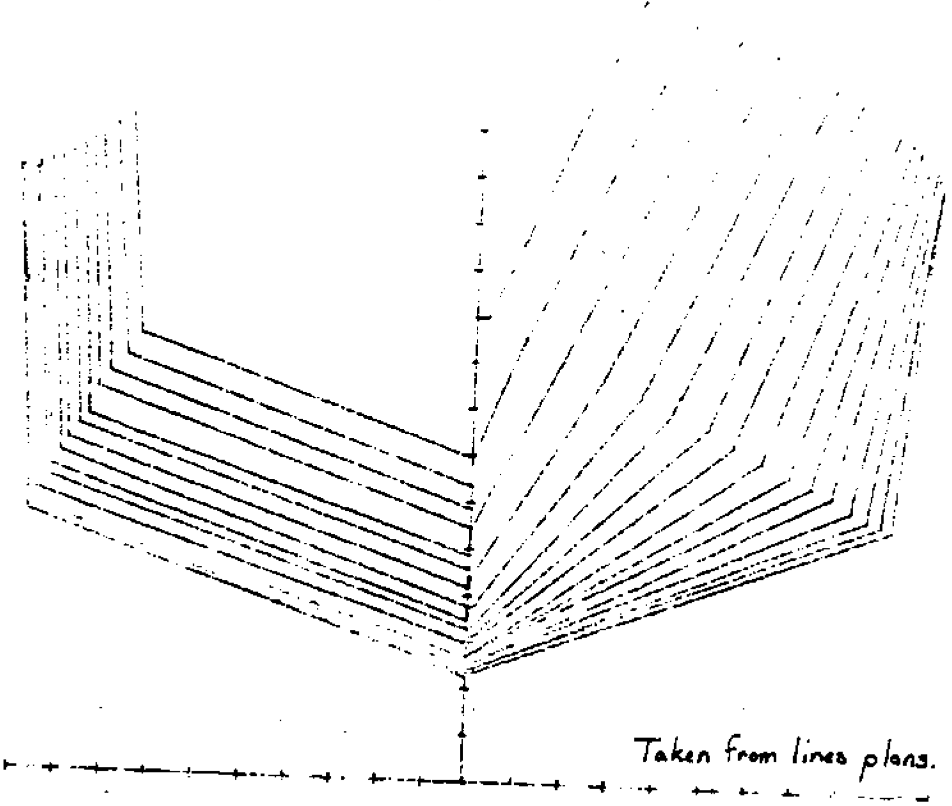
Method of Analysis

A sequence of computer programs was written and used in order to analyze numerous loadings of the test vessel and to provide the ability to analyze several different vessels. The sequence begins with a hull modeling program using B-splines to create a table of offsets to be used in the subsequent routines. If a table of offsets of sufficient accuracy is available, this routine need not be used. In the case of the test vessel, a set of lines plans was available so a table of offsets taken off the plans was used for the analysis. A comparison of a computer generated hull of the test vessel to the hull taken from the plans is provided by figure 2. The program gives very close results in general but tends to make the vessel slightly narrow at the bow. The program is especially useful in creating a hull from a minimum of points, such as when a vessel is drydocked and points are taken off the hull.

Once a table of offsets is generated, it is entered into a routine which develops cross-curves of stability at eight displacements for eight angles of heel (up to 80 degrees). This routine automatically balances and trims the boat at every heel angle. The cross-curves are calculated for a center of gravity entered by the user. A typical set of these curves is shown by Figure 3.

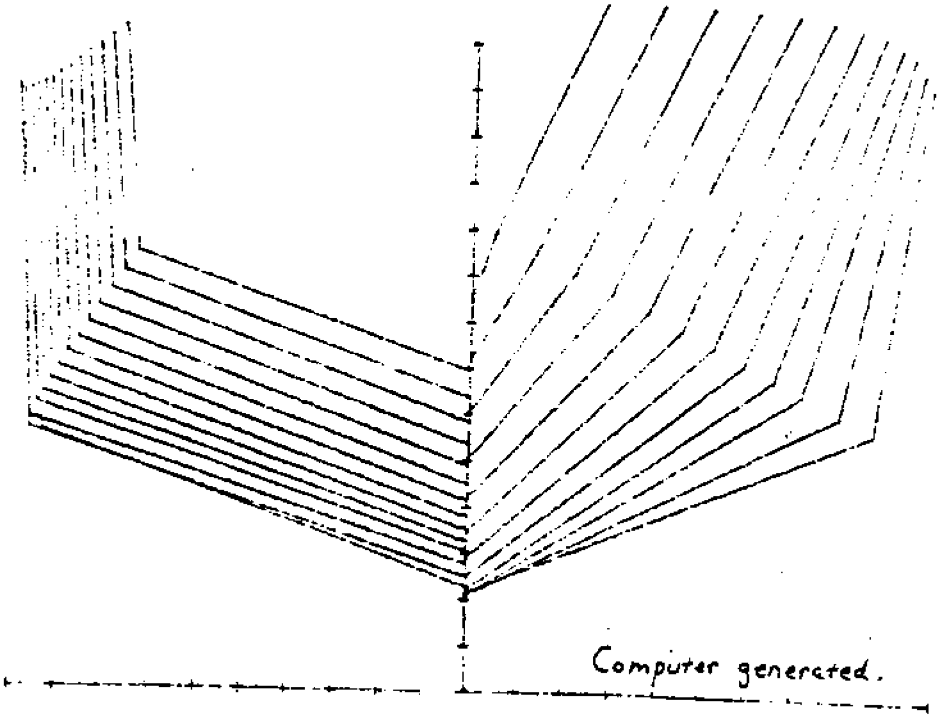
Static-stability curves are then plotted by a separate routine for any displacement which has been included in the set of cross-curves. The routine calculates the righting-arms (GZ) for the given displacement by interpolating between points on the cross-curves and plots a second-order curve between the nine (counting zero) GZ points. The areas under the curve between each angle are calculated and summed for reference to stability criteria. Next, the program superimposes either a wind- or turning-heeling arm on the static-stability curve and applies the United States Naval stability criteria as a reference. If the vessel fails the criteria a message is printed indicating the

Figure 2



Taken from lines plans.

scale: 1/4 in. = 1 ft.



Computer generated.

scale: 1/4 in. = 1 ft.

failure. The test results are evaluated using these curves and they comprise the remainder of the figures in this paper.

The initial metacenter (GM) can be determined for any trim condition for which cross-curves have been generated. Another routine determines the waterplane moment of inertia and vertical center of buoyancy and calculates the GM. In addition, a suggested minimum GM as determined by IMO standards is displayed for comparison. The GM value determined from this routine can be compared to the GM picked off the static-stability curves by extending the tangent of the curve at zero to 57.3 deg.(1 radian).

Analysis of the Test Vessel

The remainder of the paper will display the results derived from analysis of the test vessel using static-stability curves calculated for the various loadings encountered during scalloping. First, some information about the standard operating conditions and the vessel itself will be provided.

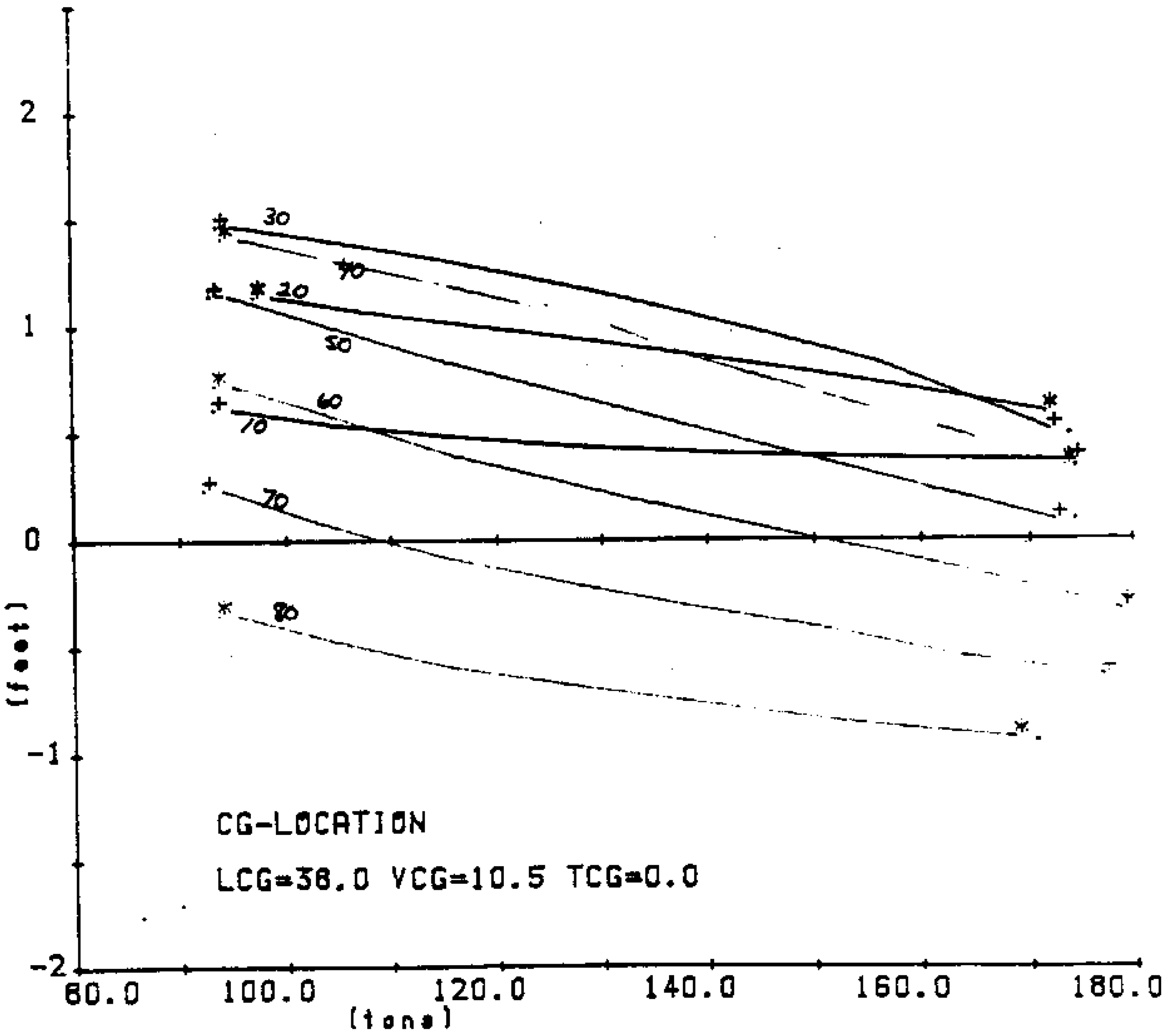
A typical trip lasts about 24 hours and includes four hours of travel time to the grounds, 16 hours of trawling and a return time of four hours. As soon as the boat clears the port, the outriggers are lowered simultaneosly. They are kept lowered until the boat returns to port. The length of a single trawl lasts anywhere from ten minutes to about 45 minutes. The doors are generally not kept on deck, but remain hung off the outriggers, along with the nets. If we assume a fuel consumption of 18 gals/hr, then the amount of fuel used during the trip is less than 450 gals. The aft fuel tanks are generally kept empty, while the forward water tank is kept full.

Now some specific characteristics of the test vessel will be given. The outriggers have been shortened from the standard length for shrimping of 40 ft., to about 20 ft. Their weight has been estimated to be 700 lbs. each. The doors are approximately 350 lbs. each, and each net about 645 lbs. When calculating the effect of the gear weight, the weight of each piece of equipment was increased to account for water absorption. The center of gravity for the lightship condition with the doors and nets on the deck was determined by performing an inclining experiment. This location was found to be 11.45 ft. from a baseline 2.16 ft. below the hull at amidships. This corresponds to about one ft. below the deck at amidships. The horizontal location was 34.3 ft. from the bow.

The following loading conditions were evaluated in order to display the change in stability of the test vessel through a typical trip. The effect of decreased fuel loading is demonstrated by comparing the results from set [1A] and set [1B] which represent fuel amounts of 2000 gals. and 8200 gals.

FIGURE 3

CROSS-CURVES OF STABILITY



respectively. The first set of loadings represent a fuel amount of 5000 gals. The effect of raising and lowering the outriggers, nets and doors can be seen by comparing lightship conditions [I], [J], and [K].

---Loading Conditions---

Condition	Fuel	Water	Catch	Stores	Nets	Outriggers	Doors
[1]	14.29	7.14	0	.90	up	down	down
[2]	13.43	7.00	25	.90	up	down	down
[3]	13.43	7.00	25	.90	up	up	up
[1A]	5.71	7.14	0	.90	up	down	down
[2A]	4.85	7.00	25	.90	up	down	down
[3A]	4.85	7.00	25	.90	up	up	up
[1B]	23.43	7.14	0	.90	up	down	down
[2B]	22.30	7.00	25	.90	up	down	down
[3B]	22.30	7.00	25	.90	up	up	up

Note: Fuel, water, catch and stores are given in tons. Loadings 1, 1A, 1B are for the vessel leaving port, 2, 2A, 2B are leaving the grounds, and 3, 3A, 3B are arrival at port with catch.

---Lightship Conditions---

[I]	[J]	[K]
Outriggers-up	Outriggers-up	Outriggers-down
Doors-------on deck	Doors-------up	Doors-------down
Nets--------on deck	Nets--------up	Nets--------up
Displ.------108.5 t	Displ.------108.5 t	Displ.------108.5 t
LCG---------34.3 ft	LCG---------34.3 ft	LCG---------34.3 ft
VCG---------11.45 ft	VCG---------11.78 ft	VCG---------11.60 ft

Note: Condition [I] is the original lightship condition during the inclining test, minus the fuel, water and stores present at the time of the test. The LCG values are the distance aft of the bow, and the VCG values are the distance up from the baseline. For reference, 11.45 ft. represents a center of gravity located about one ft. below the main deck at amidships.

The corresponding static-stability curves are shown by Figures 4 through 15. All of the curves have been corrected for free-surface effects, by considering the wing tanks and the water tank. From these curves it is can be seen that the vessel's stability is decreased greatly by the large deck load, and then decreased further by the action of raising the outriggers when entering the port.

Analysis of the Results

It has already been mentioned that many of the capsizings occurred in the port. To investigate the cause of these accidents it was necessary to model the conditions surrounding the return of a boat to port with its load. After studying the records of several of the casualties, a list of adverse forces and loadings was prepared. This list included such heeling moments as a beam wind, unsymmetric fuel loading, turning moments and running aground. The first three moments are relatively simple to apply. Figures 6, 9, and 12 show static-stability curves for the three loading conditions 3, 3A, 3B which represent fuel loadings of 5000, 2000, and 8200 gals. respectively. A wind and heeling arm curve has been superimposed on the stability curves. The heeling curve is a composite of the effects of a 30 knot beam wind and a slow turn. It is felt that the values used for the calculation of this curve were conservative. For example, the turning speed used was two knots and the turning radius was 100 ft. If a boat was instead trying to turn at five knots, the heeling curve due to turning would be six times as great as for two knots.

Another way to decrease the stability quickly is by loading the vessel unsymmetrically about the centerline. There is a documented casualty in which it was discovered that one of the reasons for the capsizing was that one tank was completely full while the other was empty. This was obviously an unusual occurrence, but we can simulate the problem by considering the following more likely scenario. If a boat drew fuel out of only one wing tank for at least one trip and then half-way through another, the total imbalance would be about 600 gals. This would cause a moment of 12.57 ft-tons creating a heeling arm of between .076 and .086 ft. depending on the displacement. Adding this upsetting force to the wind and turning arms we see that the stability of the vessel is very nearly depleted. Figures 12 through 15 show the composite wind, turning and weight shift heeling arm curve. A large weight shift also occurs if the outriggers are not raised simultaneousely, but rather one beats the other to the top. For example, consider one outrigger at 60 deg. and the other at 30 deg. The moment due to this difference is about 4.71 ft-tons. The heeling arm created is then .031 ft.

Finally we will consider the consequences of bumping the bottom while in port. According to Coast Guard reports, this was a factor in a several of the casualties. Unfortunately, calculating the exact effect that grounding would have on a vessel is not straightforward. The effect of striking the bottom, or any submerged fixed object will tend to raise the vessel's center of gravity since some of the weight is taken by the fixed object. To approximate the effect of grounding we assumed the simplified case that the boat hits directly on the keel at amidships such that no trim occurs. Now assuming that the boat rises just one inch due to grounding, the force on the bottom will be about 2.4 tons. The resulting rise in the center of gravity is then 0.155 ft., or 1.86 inches. From Figure 14 it can be seen that there is only about 0.1 ft. of maximum GZ after the vessel is subjected to the wind, turning and weight shift heeling arms. This rise could easily capsize the vessel when applied with the other upsetting forces.

Recommendations

There are procedures which if followed can reduce the chance of capsizing when the vessel is in the conditions where capsizing is most prevalent. However, these procedures are not considered adequate to provide satisfactory stability for the vessel under these loadings. The vessel will still not meet the standard IMO criteria for minimum areas under the GZ-curve. It can be seen that the vessel's stability improves with increased fuel loading. Therefore, it would benefit the operators to try to top off the fuel tanks before making another trip. Most importantly, the operators should avoid unsymmetric fuel loadings by switching tanks regularly and keeping track of the actual loadings present on the vessel at all times. Of course the policy of always keeping the forward water tank full should be continued. By keeping the doors stowed in the racks instead of raising them when arriving at port, the CG can be reduced about 1.05 inches. It is understood that this is difficult because of the deck load, so perhaps they could be dropped to the deck forward of the load (this is where they were during the inclining test). Figure 16 displays a curve for the condition of full fuel, symmetrically loaded and with the doors on deck. The situation is improved but still considered dangerous according to the IMO criteria. The outriggers should be raised only when necessary and of course great care should be taken to raise them together. They should only be handled when the vessel is not turning. Lastly, the operators should be aware of shallow areas near the docks and know the drafts of their boat at all times.

FIGURE 4

STATIC-STABILITY CURVE FOR LOADING III

<u>Loading (tons)</u>

Fuel........ 14.29
Water....... 7.14
Catch....... 0.00
Stores...... 0.90

<u>CG Effects (ft.)</u>

Free Surface....0.1921
Wind Heeling....0.0
Turning Heel....0.0
Weight Shift....0.0

Vertical Center of Gravity -- 10.45

/ GM = 2.58

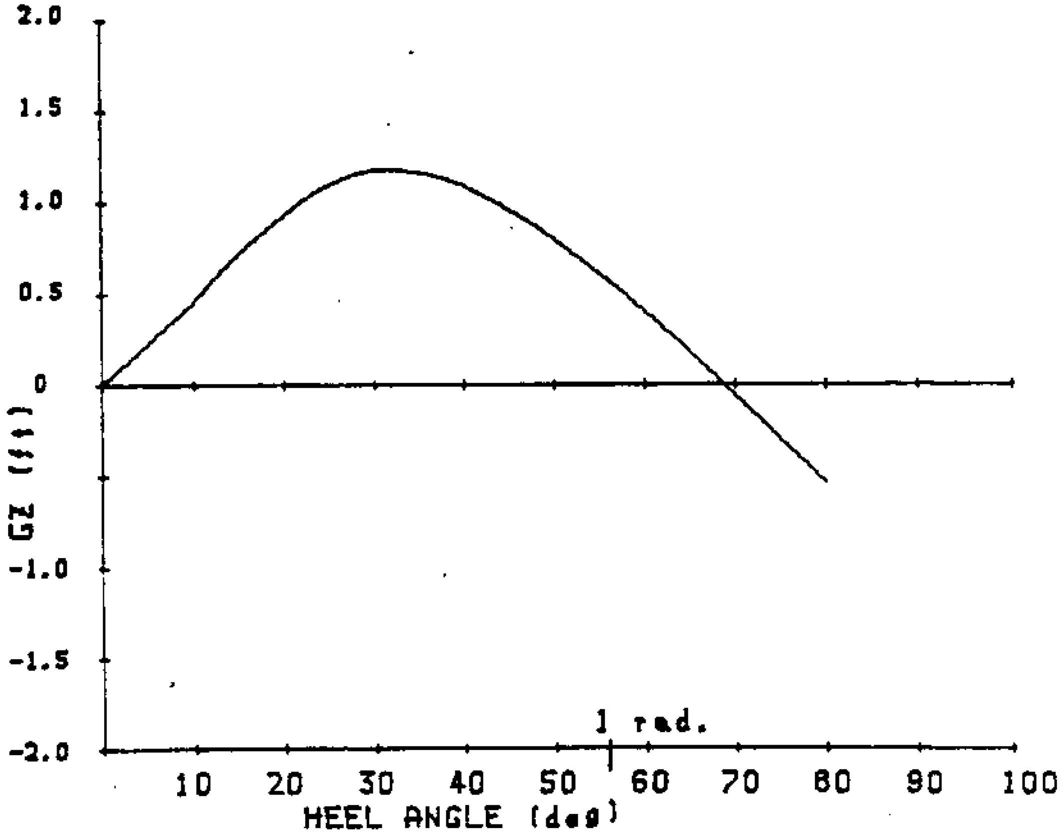


FIGURE 5

STATIC-STABILITY CURVE FOR LOADING III

Loading (tons)	
Fuel.........13.43	
Water....... 7.14	
Catch.......25.00	
Stores...... 0.90	

GZ Effects (ft.)	
Free Surface....0.0793	
Wind Heeling...0.0	
Turning Heel.....0.0	
Weight Shift....0.0	

Vertical Center of Gravity (ft) 11.52

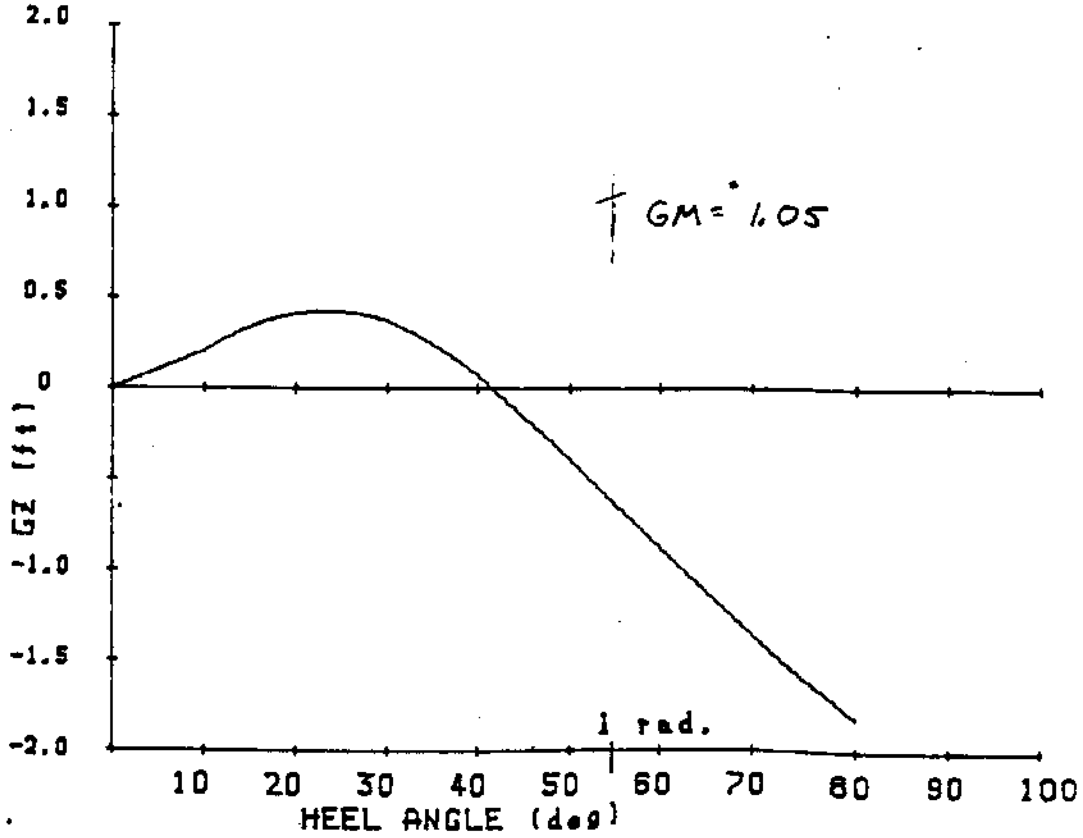


FIGURE [illegible]

STATIC-STABILITY CURVE FOR LOADING [illegible]

<u>Loading (tons)</u>

Fuel.........13.43
Water....... 7.14
Catch.......25.00
Stores...... 3.91

<u>CG Effects (ft.)</u>

Free Surface....0.0793
Wind Heeling....0.[illegible]
Turning Heel....0.0039
Weight Shift....0.0

Vertical Center of Gravity == [illegible]

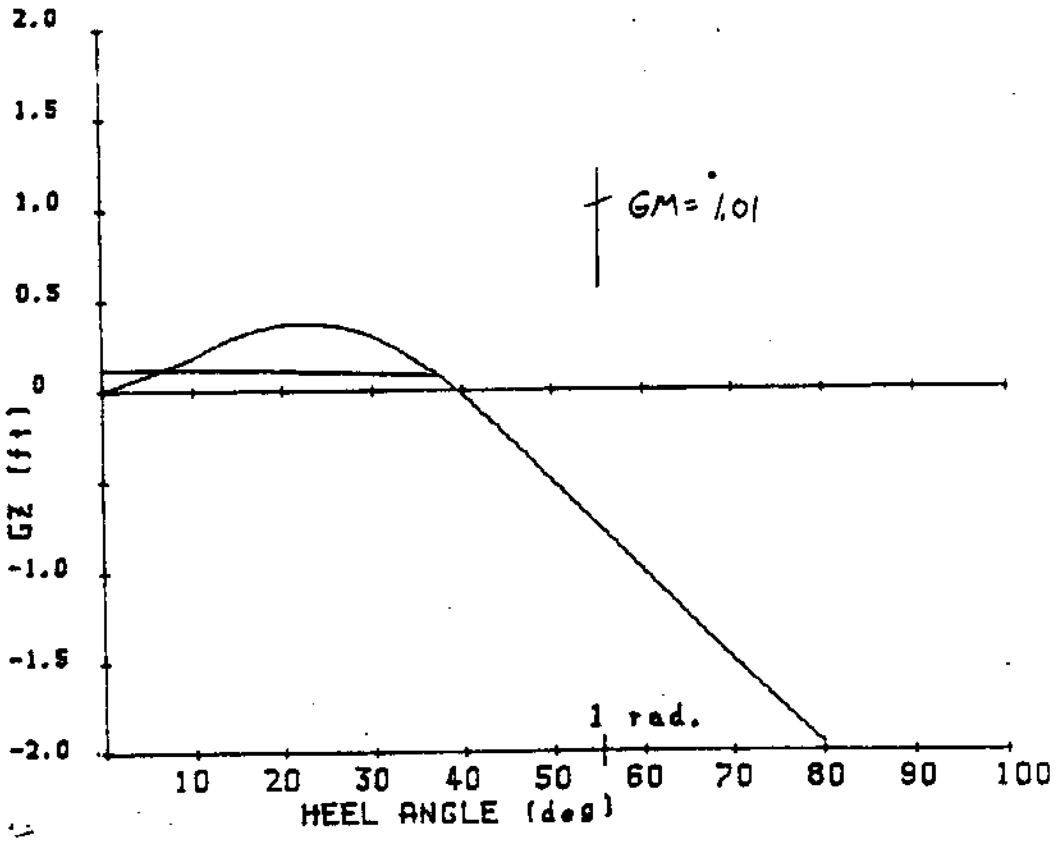


FIGURE 7

STATIC-STABILITY CURVE FOR LOADING (14)

Loading (tons)	CG Effects (ft.)
Fuel 5.71	Free Surface....0.[illegible]
Water....... . 7.14	Wind Heeling....0.0
Catch........ 0.00	Turning Heel....0.0
Stores....... [illegible]	Weight Shift....0.0

Vertical Center of Gravity (ft) ...[illegible].82

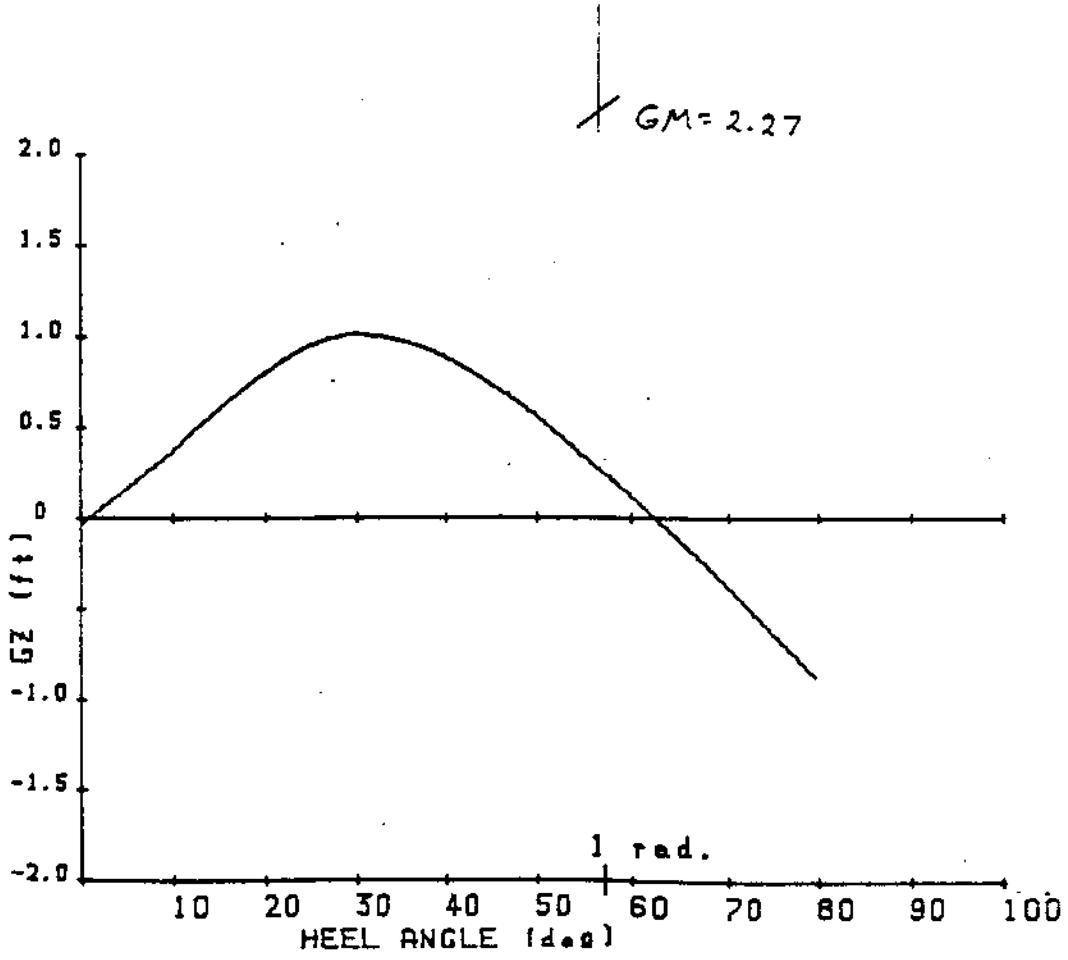


FIGURE 9

STATIC-STABILITY CURVE FOR LOADING (3A)

Loading (tons)

Fuel......... 4.85
Water........ 7.00
Catch.......25.00
Stores...... 0.90

CG Effects (ft.)

Free Surface....0.0934
Wind Heeling....0.0
Turning Heel....0.0
Weight Shift....0.0

Vertical Center of Gravity (ft) 11.84

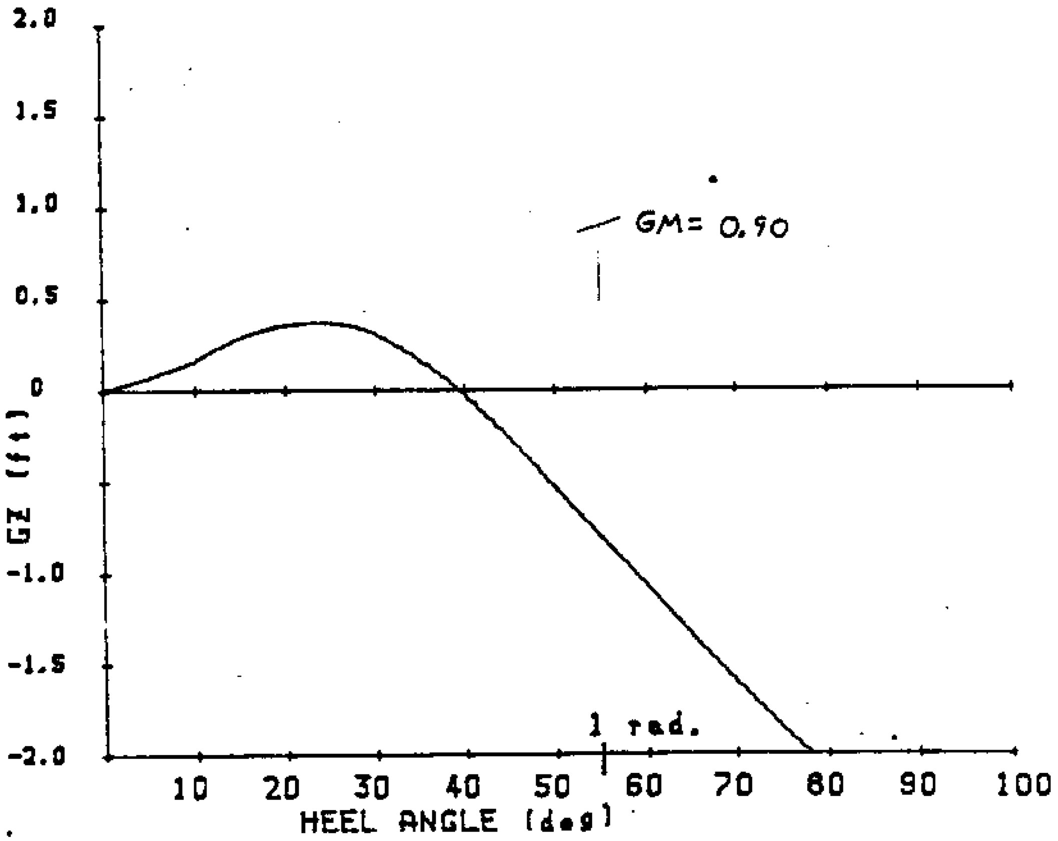


FIGURE 7

STATIC-STABILITY CURVE FOR LOADING (34)

Loading (tons)

Fuel.......... 4.95
Water........ 7.00
Catch........25.00
Stores...... 0.90

CG Effects (ft.)

Free Surface....0.0224
Wind Heeling....0.0347
Turning Heel....0.0233
Weight Shift... 0.0

Vertical Center of Gravity (ft) 11.54

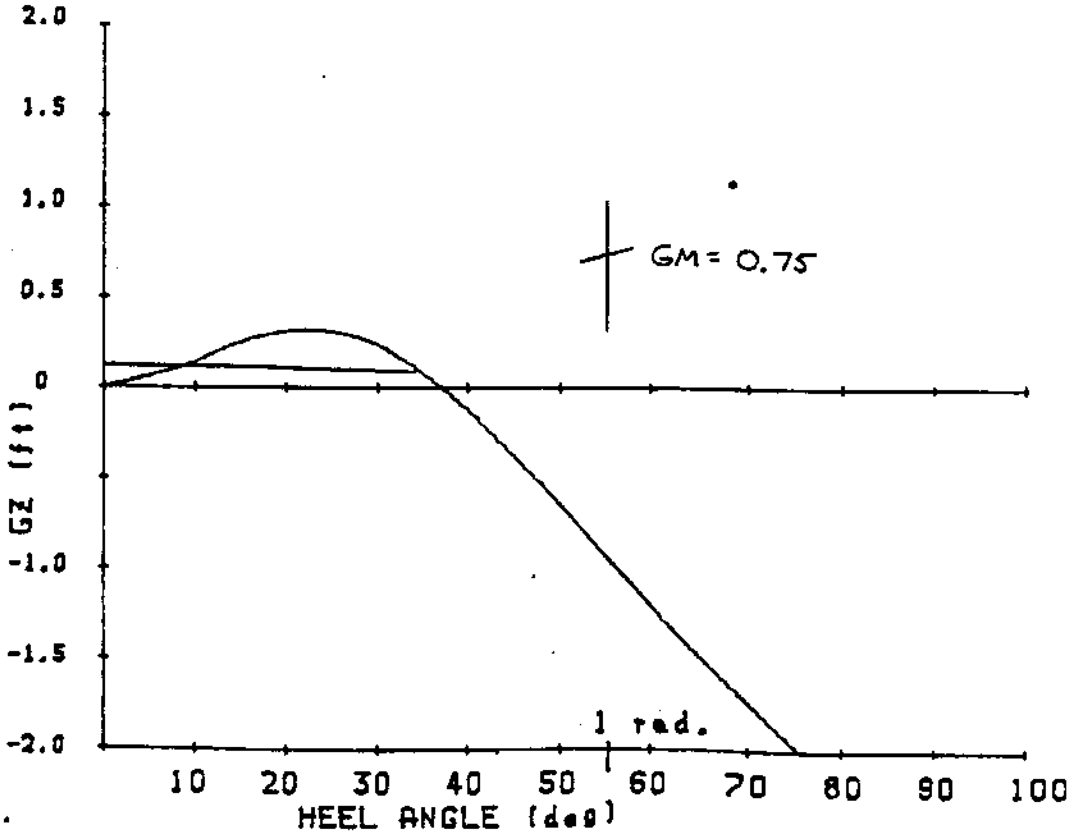


FIGURE 11

STATIC-STABILITY CURVE FOR LOADING (1E)

Loading (tons)

Fuel..........23.49
Water........ 7.14
Catch........ 0.00
Stores... .. 0.90

CG Effects (ft.)

Free Surface....0.0921
Wind Heeling....0.0
Turning Heel....0.0
Weight Shift....0.0

Vertical Center of Gravity -- 11.35

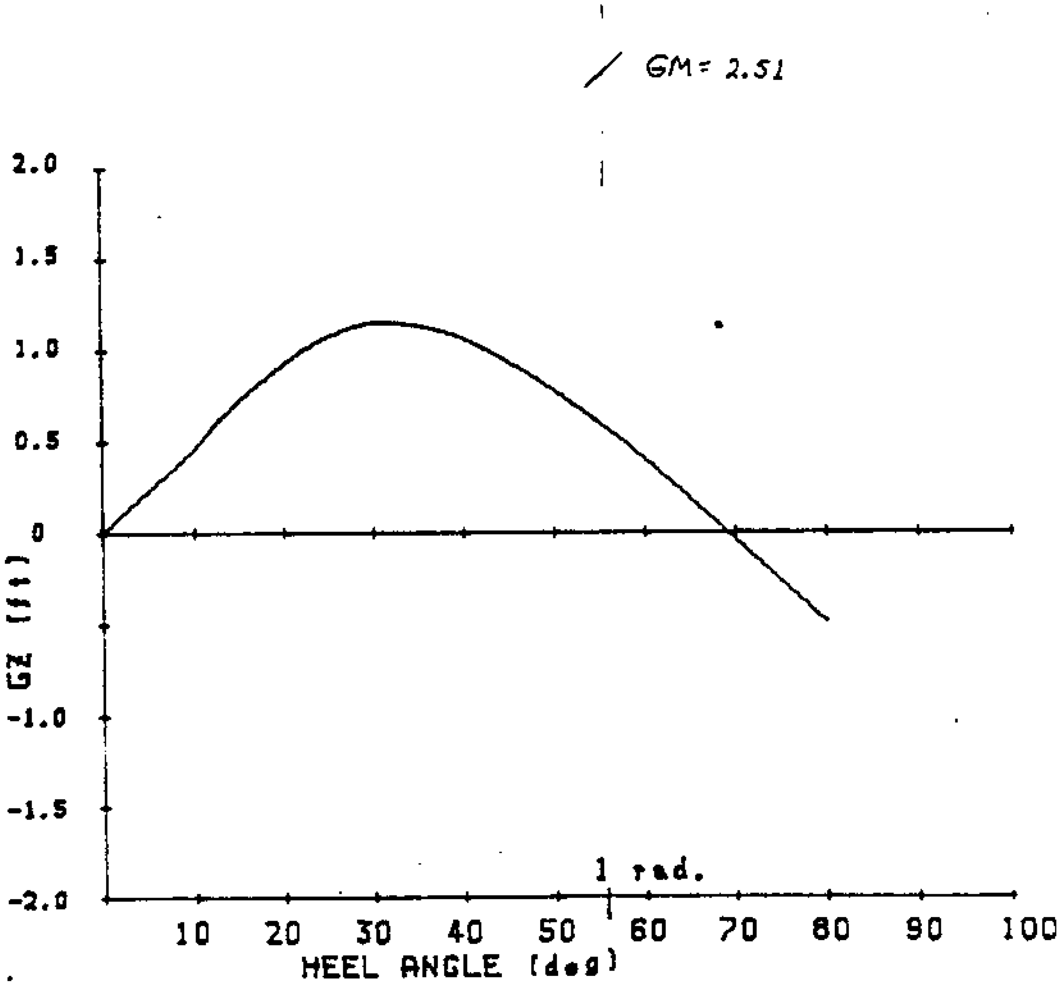


FIGURE 11

STATIC-STABILITY CURVE FOR LOADING [illegible]

Loading (tons)

Fuel.........22.45
Water........ 7.00
Catch........25.00
Stores....... 0.90

CG Effects (ft)

Free Surface.....0.081
Wind Heeling....0.0
Turning Heel.....0.0
Weight [illegible]....0.0

Vertical Center of Gravity -- 11.33

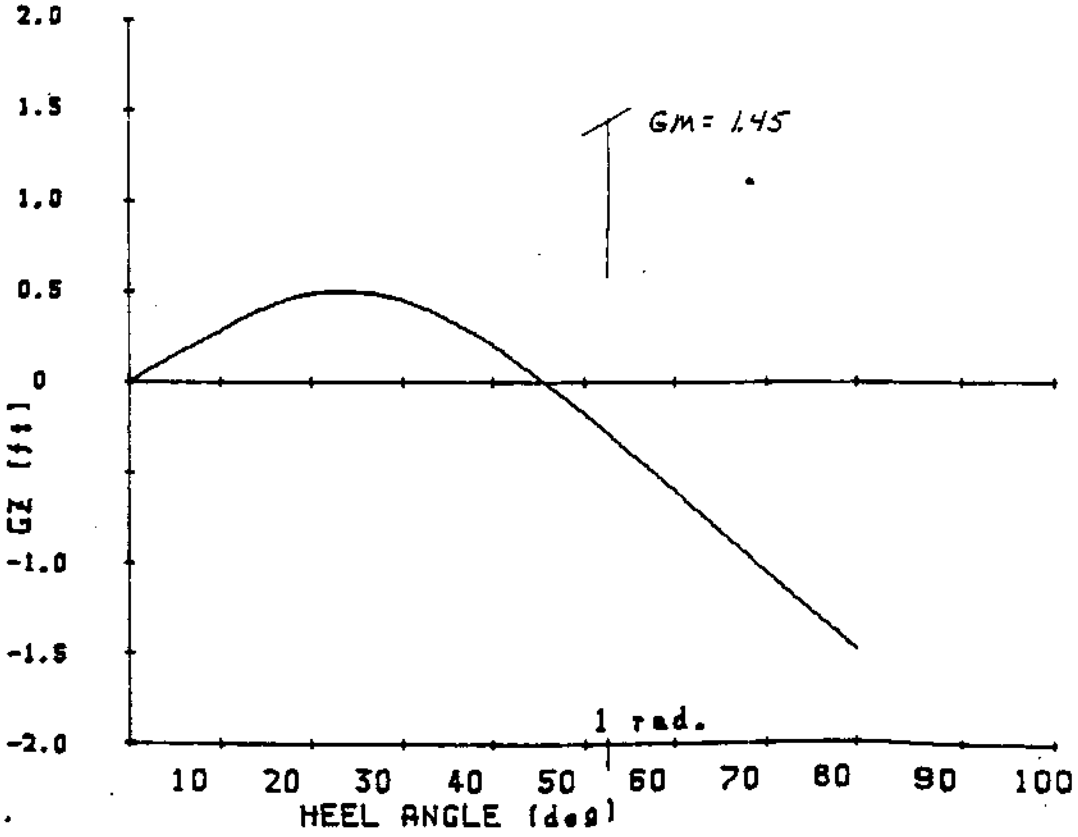


FIGURE 12

STATIC-STABILITY CURVE FOR LOADING (28)

Loading (tons)		CG Effects (ft)	
Fuel.........22.45		Free Surface....0.0757	
Water........ 7.00		Wind Heeling....0.0[illegible]	
Catch........25.00		Turning Heel....0.0[illegible]	
Stores....... 0.90		Weight Shift....0.0	

Vertical Center of Gravity (ft) 11.5[illegible]

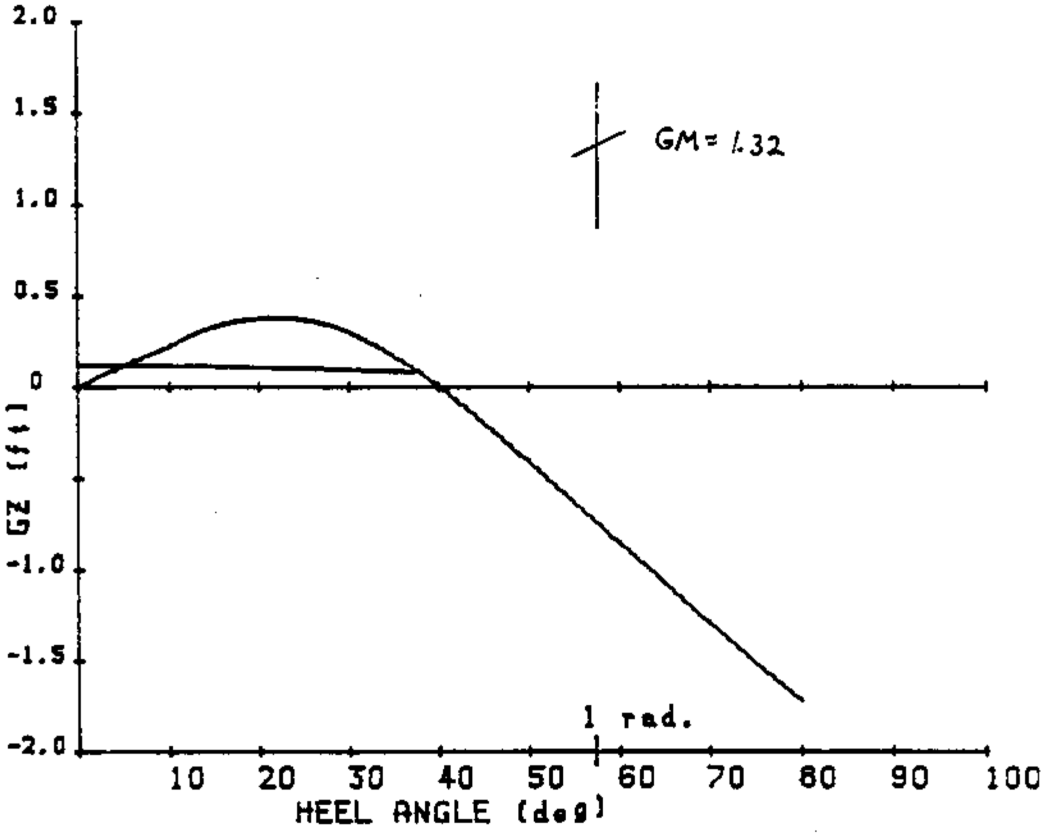


FIGURE [illegible]

STATIC-STABILITY CURVE FOR LOADING [illegible]

Loading (tons)		CG Effects (ft.)	
Fuel	13.43	Free Surface	0.0783
Water	7.00	Wind Heeling	[illegible]
Catch	25.00	Turning Heel	0.0238
Stores	0.90	Weight Sh ft	[illegible]

Vertical Center of Gravity (ft) ---> [illegible]

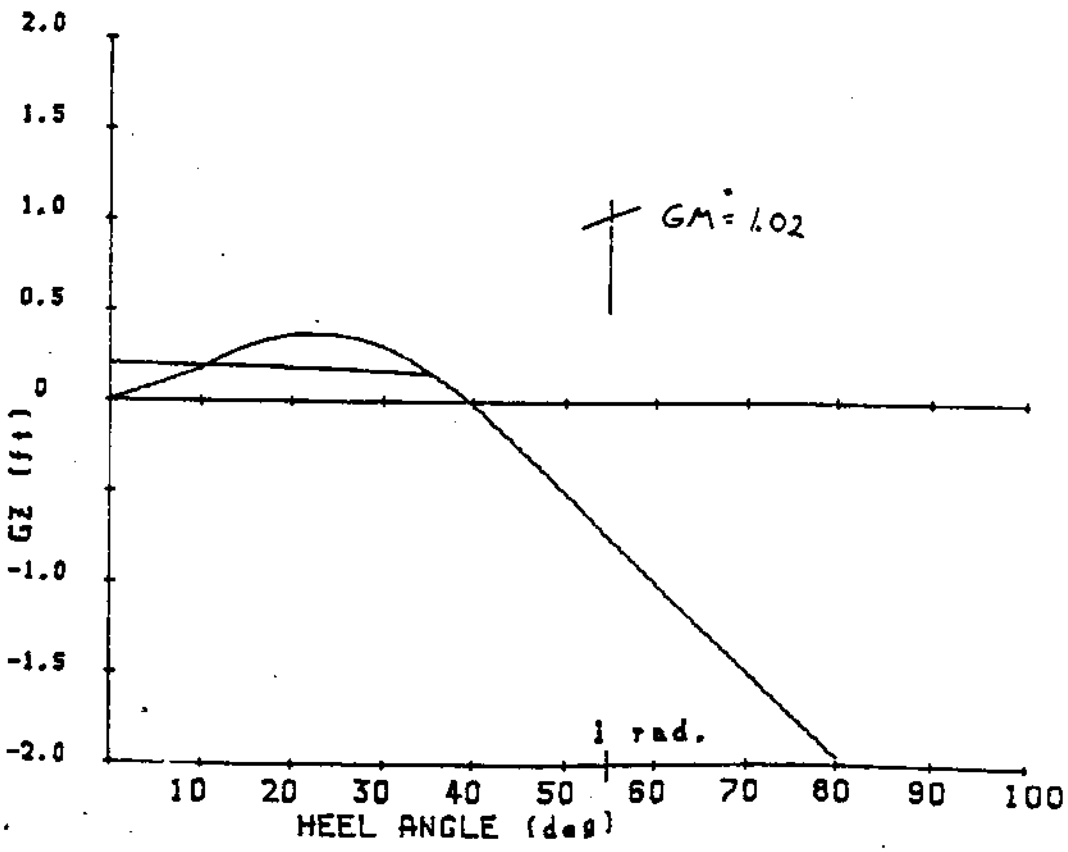


FIGURE 14

STATIC-STABILITY CURVE FOR LOADING CASE [illegible]

Weights (tons)

Fuel.......... 4.85
Water......... 7.00
Catch.........25.[illegible]
Stores........ [illegible]

GZ Effects (ft)

Free Surface....[illegible]
Wind Heel Arm...[illegible]
Turning Heel....[illegible]
Height GM ......[illegible]

Vertical Center of Gravity (ft) -- [illegible].34

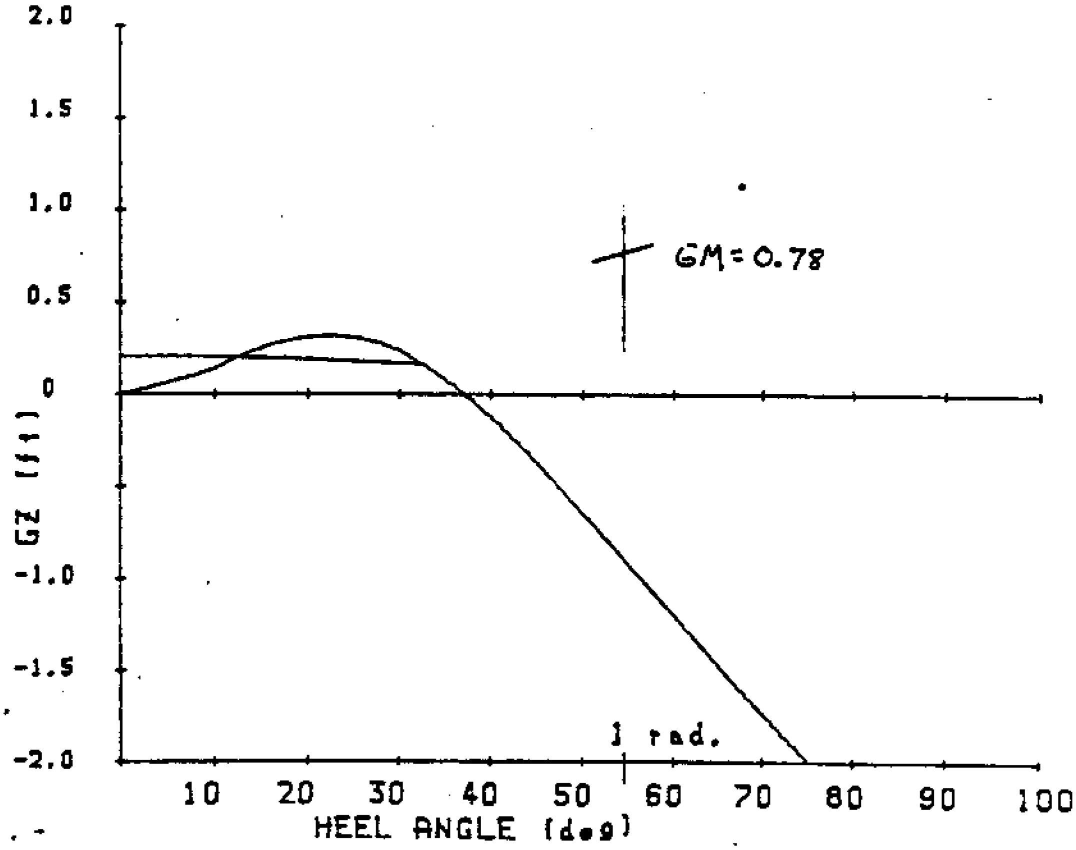


FIGURE 15

STATIC-STABILITY CURVE FOR LOADING CASE 1

Loading (tons)		CG Effects (ft.)	
Fuel..........	23.65	Free Surface....	0.0757
Water.........	7.00	Wind Heeling....	0.0965
Catch.........	25.00	Turning Heel....	0.0238
Stores........	0.90	Weight Shift....	0.0755

Vertical Center of Gravity -- 11.26

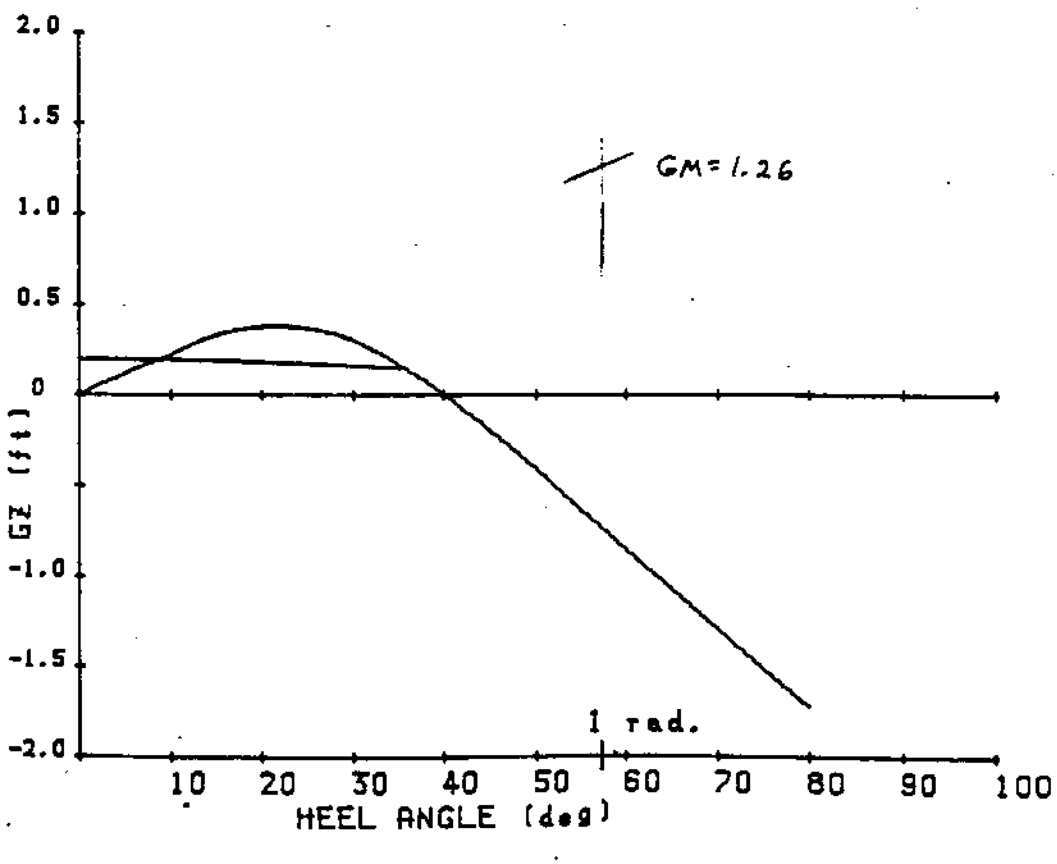


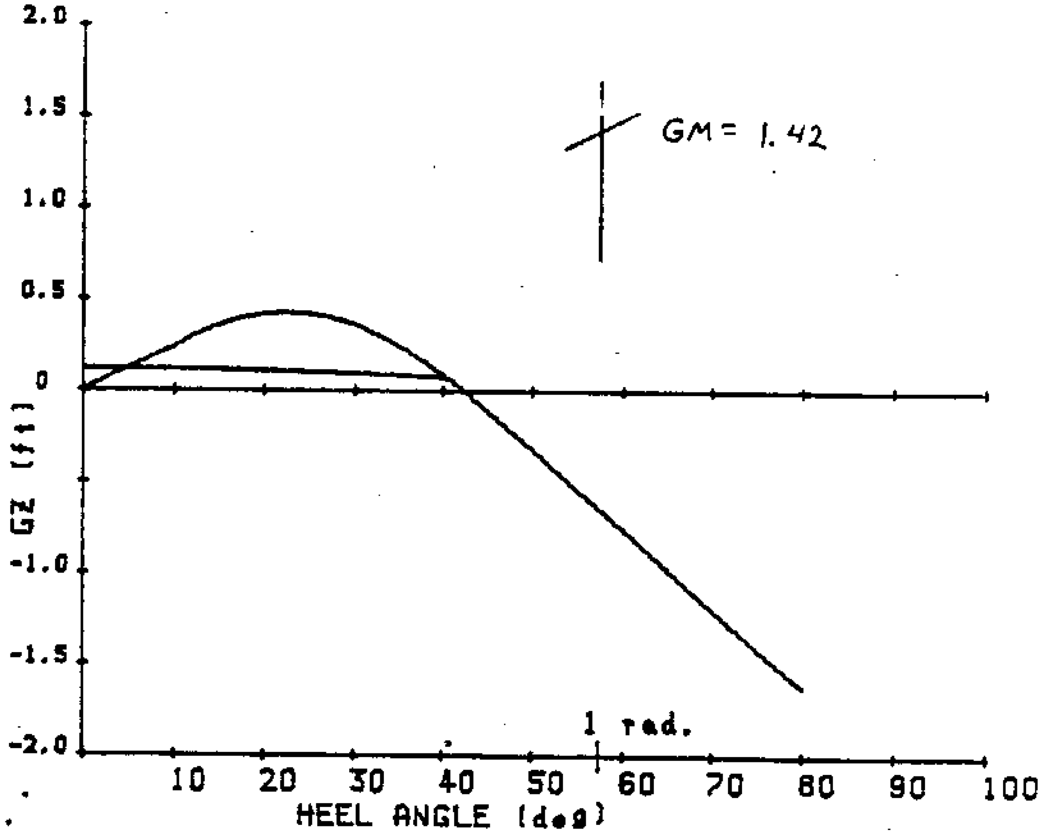
FIGURE 16

STATIC-STABILITY CURVE FOR LOADING [3B"]

Loading (tons)	
Fuel	23.65
Water	7.00
Catch	25.00
Stores	0.90

CG Effects (ft.)	
Free Surface	0.0757
Wind Heeling	0.0965
Turning Heel	0.0238
Weight Shift	0.000

Vertical Center of Gravity --> 11.26



WRITTEN CONTRIBUTION

Bruce Culver

1. Regulations similar to those used for tugs and supply boats might be applied. Basic criteria should include at least a minimum initial GM and a minimum freeboard, calculated from some simple criteria.
2. I agree that conservation regulations have an adverse effect on vessel operations, but I'm afraid very little can be done about it.
3. Boats of normal form designed to meet IMO criteria are generally safe from capsizing. The IMO criteria are too severe for boats of less than perhaps 25 meters in length.
4. No damage stability criteria are practical for boats of this size. If you poke a hole in it too big for the bilge pumps all bets are off.
5. Some form of simplified licensing should be applied to captain's only. Forget 200 gross ton limit.
6. No ABS classification should be required-not economic.
7. Legislate size and design of freeing ports.
8. Survival suits and emergency locator beacons should be mandatory.
9. Small crews and inadequate rest contribute to accidents, but rules on this would be uneconomic, hard to enforce, and strongly resisted by the industry. Laissez-faire policy is best.

WRITTEN CONTRIBUTION

Cliff Goudey

It is interesting to note, and I'm sure represents a source of frustration to Dr. Adee, that draft marks are not used on these vessels.

Reductions freeboard are of course observable but difficult to quantify without some reference points. These draft marks would seem to be of day to day value to an operator in maintaining efficient trim.

I wonder if West Coast naval architects include draft marks on their designs or are they simply painted-out during their first dry docking?

WRITTEN CONTRIBUTION

Cliff Goudey

As a representative of the MIT Sea Grant Program, I also participated in the Fishing Vessel Safety Conference described by Mr. Shephard. At that time MIT's role in fishing vessel safety, both past and present, was outlined.

We have conducted a variety of research and advisory projects addressing saftey issues brought to our attention by our local fishing industry. Innovations in deck hardware have resulted from two of these projects allowing reduced risk of injury during fishing activities. A more recent project has developed a thermographic technique for the detection of flaws in GRP hulls. Losses from failure due to imcomplete resin saturation or missing laminates could be reduced by this technique.

Other projects have had an indirect relation to safety. The implications on safety and vessel survivability was an important phase of our sail-assist project reported at last year's conference in Tarpon Springs.

A considerable amount of effort went into the safety aspects of the shrimp trawl development which I reported on yesterday morning (paper 2.4). MIT's research on the added resistance of fishing vessels in head seas, reported yesterday afternoon, was in response to the Coast Guard's need to respond effectively to calls-for-help from distressed fishing vessels.

I believe it is important for all of us to be aware of the safety implications of our efforts, particularly if they relate to the modification of vessels, the introduction of new gear, or the development of an alternative fishery. As Mr. Adee so clearly demonstrated, the operator cannot be relied upon to be a proper judge of vessel safety. Driven by the pressures of economic survivability, situations can evolve with disasterous results. Progress towards efficiency and fishing effectiveness should not be at the expense of fishermen's lives.

The National Sea Grant College Program can help by insuring that in the projects they support that relate to fishing, the implications of safety are properly considered. In addition, the effective linking of Sea Grant advisory activities and research projects should be stressed. In this way, safety developments from other marine fields can be identified and adapted if appropriate.

The MIT Sea Grant Program, for its part will continue to respond to the needs of our local fishing industry. Participation on the safety committee will also allow our response to national problems if appropriate.

MIT's capabilities in naval architecture, dynamic analysis, mechanical design, and structures represents a formidable resource to confront the engineering aspects of this problem. A national commitment to reduce fishing vessel accidents must be accompanied by a commitment to support the cost of the needed research. Otherwise we are destined to simply continue the head count and go on wishing something could be done.

WRITTEN CONTRIBUTION

N. T. Riley

In Australia tuna vessels must have stability data. Most states require stability data to be provided for other fishing vessels also.

The stability criteria used is that promulgated by IMO but we must use free trimming cross curves.

The structure, machinery, outfit, life saving appliances as well as the stability requirements are regulated by the Uniform Shipping Laws Code. We do not see these regulations as being onerous but as a guide to reasonable design and also the one set of rules, with minor variations, covers the whole of Australia.

The rules provide for one set of stability data covering sister ships provided the light ship weights agree.

My one preference is to incline sister ships, then there is no question of errors creeping into the stability data which is used for these "similar" ships.

It is a requirement in New South Wales it is mandatory that for wooden vessels to have the lines lifted of the vessel when preparing the stability data. The reason for this requirement is that frequently the variations between the lines as designed show marked variations with the vessels as built.

I would like to make a few remarks on Professor Latorre's paper.

In a paper (1) we delivered in 1979 we looked at the effects on stability of an 80,000 ton D.W. tanker, a 3,000 ton D.W. cargo vessel and a number of fishing boats when free trimming and balanced on a standard wave.

From the results obtained we concluded the current methods adopted for assessing the stability of vessels have little resemblance to what really existed when free trimming and wave effects were considered and as such seriously overestimated the statical stability and dynamical stability available.

We further concluded that increasing beam was the most effective way of obtaining increased stability, at large angles of heel.

Ref. (1) "Some Aspects of Fishing Boat Stability", Riley and Helmore Symposium on Ship Technology, University of N.S.W. 1979.

WRITTEN CONTRIBUTION

Robert Hyde

Ultimately, the uniform recognition of an intact stability criteria (as opposed to requiring only a certain positive initial metacentric height) which addresses the complex nature of vessel stability is more important than determining which of the available standards is more suitable. Based on the available research, the IMO stability criteria (Resolution A.168) appears to be satisfactory. My concern, however, is that most of the analyses applying or evaluating that criteria fail to reflect proper consideration of downflooding as incorporated into that criteria. Downflooding on a fishing vessel can occur at extremely low angles of heel. The typical result is that the 0.03 meter-radians area requirement between 30 and 40 degrees of heel would not be met. The failure to take downflooding points into account would cast some doubt on the conclusion that the IMO stability criteria is the appropriate standard for assessing the intact stability of a fishing vessel. I suggest that researchers approach this problem more carefully.

PART V

SAIL-ASSISTED FISHING VESSELS

DESIGNS FOR WIND-ASSISTED COMMERCIAL FISHING VESSELS

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(Editors' Note: We regret that due to the length of this paper it is possible to include only the narrative portion. Forty-six pages showing hull designs and other data are omitted. Anyone wishing to review these pages may contact Dr. John Sainsbury, Department of Oceanography and Ocean Engineering, Florida Institute of Technology, Melbourne, FL 32901.)

ABSTRACT

Hull lines drawings, offsets and particulars are given for two catamaran and five single hull configurations suitable for use as commercial fishing vessels. Recommended sail areas are given using the criterion of ten degrees of heel in 20 knots of apparent wind velocity. Comments are included on the operational decisions necessary to choose between twin and single hull vessels as commercial fish boats. Stability data on the single hull craft are shown with righting arm curves from zero to 180 degrees. One hull is a classic 45 foot workboat from a book of Chapelle's. Two 32 foot and one 38 foot versions have been derived. While most suited for implementation by other naval architects, there is sufficient information for experienced boatbuilders to construct these hulls. All are of the hard chine type for ease and economy of building.

This article was developed under the auspices of the Florida Sea Grant College Program with support from the National Oceanic and Atmospheric Administration, Office of the Sea Grant, U.S. Department of Commerce, Grant No. NA80AA-D-00038.

INTRODUCTION

The investigation by the author of the operational and engineering practicality and the economics of wind-assisted commercial fishing vessels comes to a close with this report, the 27th during the past three and one-quarter years. A few of the more pertinent ones are listed under references. Up to now, the research and design has been concerned largely with the retrofit case and to answer the question: is it economically and operationally sound to equip existing fishing boats with sail rigs? The answer was found to be usually yes but is highly dependent on such considerations as the proximity of low bridges, range to the fishing grounds and clean superstructures among others.(1)(2)(4). Originally, it was planned to retrofit a snapper-grouper boat or a shrimp trawler and take actual measurements of fuel useage and performance at sea in a full-scale experiment. A computer-based instrumentation package was developed for this purpose and is reported on separately.(7) This was found to be unneeded with the successful full-scale demonstrations in: Federal Republic of Germany, France, England, Norway, Sweden and the U.S.A. and the general agreement of those results with the computer models developed here.(1)(5)(6) In summary, depending on the fishery, the use of soft sails on commercial fishing vessels can be expected to show savings of between 15 and 40% in fuel useage, and sail rig pay back time ranges from one to two and one-half years for simple, inexpensive arrangements.

This paper presents lines drawings for seven possible hulls which could be used for commercial wind-assisted fishboats. (Not included. See Editors' Note.)

HULL DESIGNS - GENERAL CONSIDERATIONS

These hulls were designed on an Apple IIe microcomputer equipped with the Letcher Offshore Design software for hull design, hydrostatics and stability computations.(8) All hulls are of the hard chine variety with either one or two chines each side of the centerline. Two builders in Florida were interested in constructing wind-assisted fishboats. Both wanted hard chine hulls. One is to build a single hull craft and the other a catamaran. These builders: Howard Branch and Garland Webster, gave the impetus for these designs. Hard chine hulls have several advantages over round-bilged. They are inexpensive and relatively easy for one man to build. Materials can be either marine plywood or sheets of fiberglass cast on levelled formica tables and bent around frames or moulds. The chines will give some resistance to leeway though certainly not to the extent of keels or boards. Their major disadvantage is perhaps a ten percent reduction in hull volume for a single chine versus a round hull.

BUILDING CONSIDERATIONS

Sheet materials were mentioned above. Aluminum could be added to this list as could cored fiberglass using such materials as: balsa, foam, honeycomb and the like. The disadvantage of these two materials is cost and added skills needed. These vessels are too small to consider steel construction, except perhaps in the case of the 45 footer. Marine plywood covered with fiberglass and all edges sealed with epoxy can produce a very robust structure and is well-suited for workboats. A simplified method of building the chine log has been described by Jim Brown.(9) The builder simply butts the plywood sheets at the chines. On the inside, he builds shallow wooden troughs about three or four inches wide along each chine. Into these, wet fiberglass matt is laid in several layers. The chines are then sanded off on the outside. This is much simpler and less time-consuming than spiling and bevelling solid wooden chine logs.

Scantlings are not given but may easily be developed by any of a number of standard techniques. Note that for each craft, areas at each station are given from which materials estimates of bulkheads can easily be made. Also given are total hull areas and surface areas under the waterline only which will assist making materials estimates, developing cost figures and determining final center of gravity. The builder should remember that weights must be grouped evenly fore and aft about the longitudinal center of buoyancy if the hull is to float "on her lines."

SAIL AREA

In each case, a sail area is recommended suitable for use in subtropical waters such as exist near the southeastern United States. The amount of sail area is based on a maximum of ten degrees heel in 20 knots of apparent wind. Rigs may be of the simple, freestanding mast type with one main sail or be one-masted sloop, cutter or cat-rigged. A convenient arrangement often is to step the mast on a thwartships bulkhead just aft or forward of the fish hold.

LATERAL PLANE AND RUDDERS

No lateral plane devices or areas are suggested here. Due to extremely shoal waters in this region, the only practical keel will be a long, low aspect ratio skeg which also will protect the hull and propeller when grounding. Some may wish to use leeboards to help prevent drift to leeward. However, since these are motor sailers, use of the engine tends to limit leeway angle to a few degrees. Rudders can be as simple as outboard, barn-door types to retain shoal draft or of the posted types through the hull near the stern. The builder should remember that at low speeds, the rudder area required is considerably more than at high speed and take into account his planned service speed.

ENGINES AND ENGINE USE STRATEGIES

Wind assist means just that - when of a favorable strength and direction the wind is used to assist the engine to propel the boat. Except in the case of engine failure, it is never assumed that the fisherman will use sails alone, although an enterprising fisherman concerned with maximum fuel savings could at times do that. Each operator will have his own engine use strategy, and this depends also on the average strength and direction of the winds in the particular area sailed - hence no horsepower estimates have been given. The fish boat owner must decide on a long term average what percentage of power he wishes to use from the wind and how much from the engine. In any case, maximum rated engine horsepower can be reduced to approximately half that required for pure power boats. A major factor in the decision on rated horsepower is for the owner to set a minimum service speed under engine alone and a maximum wind velocity against which the engine must be able to propel the boat, even at low speed. In any case, speeds on the order of the square root of the waterline length will probably be near maximum.

In Table 1 are summarized a number of parameters of five single hull craft designed for wind assist. The Multichine 41 appears bargelike at the aft endings due to the client specification that all major weights be aft of midhships. The longitudinal center of buoyancy is a rather extreme 58% aft! The Chapelle 45 is taken from Reference (10). Each of the following three is a derivative of the 45 with constant block and prismatic coefficients. In each case the righting arm: GZ in feet is a measure of stability at ten degrees heel and thus is a measure of sail carrying power. Drawings of each of these hulls are shown in the annexes. (Not included. See Editors' Note.)

TABLE 1

PARTICULARS - SINGLE HULL VESSELS

ITEM	MULTICHINE 41	CHAPELLE 45	CHAPELLE 32A	CHAPELLE32B	CHAPELLE38
LOA:	41	45	31.7	31.7	37.8
LWL:	36	44	31	31	37
BOA:	11.34	13.7	9.28	13.7	11.66
BWL:	10.42	12.7	9.02	12.7	11.40
DRAFT:	2.64	2.33	1.66	2.33	2.33
DISPL.:	32993	29814	10705	21172	22600
L/B:	3.45	3.5	3.4	2.4	3.2
B/T:	3.95	5.45	5.43	5.45	4.89
AREA WS:	366	441	222	314	341
AREA HULL:	685	792	392	557	623
RIGHT ARM:	0.6	1.12	0.74	1.12	0.87
PRIS.COEF:	0.63	0.52	0.53	0.53	0.53
BLK.COEF:	0.52	0.36	0.36	0.36	0.36
DLR:	316	156	160	317	199
LCF:	25.55	24.50	17.38	17.38	20.65
LCB:	25.97	23.73	16.85	16.85	20.01
GM(T):	2.05	7.74	4.50	6.73	5.12
GM(L):	44.4	82.11	57.13	40.0	57.13
VCB:	-0.99	-0.73	-0.52	-0.73	-0.73
AREA WP:	294.4	393	198	279	298
PPI:	1570	2096	1056	1488	1589
MT-1:	3391	4637	1644	2277	2908
HULL CG X:	23.23	23.23	16.17	IP6.12	19.15
HULL CG Z:	+.09	+.11	+.06	+.11	+.15
SAIL AREA:	638	1033	314	940	760

NOTES

1. Lengths, beams, drafts, etc. are in feet. Areas are in square feet. Displacement is in pounds.
2. LOA - length overall from bow to stern.
3. LWL - length on waterline of waterplane along centerline.
4. BOA - beam overall. (maximum width of top of hull.)
5. BWL - beam waterline. (maximum width on the waterline of the waterplane.)
6. Draft - distance from waterline to bottom of fairbody not including keel.

7. Displacement is the weight of the entire vessel and its contents.
8. L/B - length to beam ratio on waterline.
9. B/T - beam to draft ratio of fairbody (not including keel) on waterline.
10. Area WS is area of hull wetted surface (below waterline).
11. Area Hull is entire hull surface area.
12. Righting arm is the GZ at ten degrees heel with trim change.
13. DLR - displacement length ratio in tons per cubic foot.
14. LCF - longitudinal center of flotation measured from bow aft - not forward waterline ending.
15. LCB - longitudinal center of buoyancy - same. All weights must be grouped equally fore and aft about this point for vessel to float evenly on design waterline.
16. GM(T) - transverse metacentric height.
17. GM(L) - longitudinal metacentric height.
18. VCB - vertical center of buoyancy measured from waterline.
19. Area WP - waterplane area.
20. PPI - pounds per inch immersion.
21. MT-1 - monent to trim one inch in foot-pounds per inch.
22. Hull CG X is distance aft from bow ending of hull surface center of gravity. Z value is height from waterline.
23. Sail Area - allowable square footage for ten degrees heel in 20 knots of apparent wind in subtropical climes except for Chapelle 32S.

CATAMARANS AS COMMERCIAL FISH BOATS

Twin-hulled vessels have been used in commercial service for thousands of years. When the hulls are of equal length and shape and are aligned, the vessel is called a catamaran. The name apparently derives from the Indian word: Kattumaram. At this time, there are approximately 50,000 commercial fishing kattumarams on the east coast of India, most of which use sails as the chief method of propulsion.

The concept of shoal draft, economical, beachable load-carrying catamarans as fishing vessels appears eminently suited to many areas of the world whether propelled by engines or sails or both. Long narrow hulls require much less horsepower to attain a given speed than short fat ones. Effective horsepower is proportional to speed times resistance. It has long been known that resistance decreases with displacement-length ratio: DLR = /(.01L). Simply put, the major cause of drag for long narrow hulls is frictional resistance, while that for short fat ones is wavemaking resistance. Thus, catamarans can either travel faster for the same horsepower/sail area or can have reduced horsepower/sail area for the same speeds.

Others have written that 50% of the cargo in catamarans can be considered as acting as ballast. When beached, the catamaran gains stability with both bows grounded. The catamaran is safer and more practical in larger sizes designed to carry heavy cargo with perhaps water ballast. They also provide a stable non-heeling platform and large work space. Their shallow draft allows access to numerous bay and close-in fisheries. Their ease of beaching makes for more frequent and economical bottom cleaning and painting.

The most usual reason for not considering catamarans is the fear of capsize. While it is true that most catamarans are more stable upside down than rightside up, so are most motorized fishing boats which have a much greater tendency to capsize than do catamarans. The other reason is difficulty and expense in finding dock space due to the large beam.(3) Table 2 illustrates the particulars of two catamarans designed as work boats. The smaller 27.5 footer is a derivative of the first except that its hulls are more widely flared.

TABLE 2

PARTICULARS - CATAMARAN FISH BOATS

ITEM	CAT 41	CAT 27.5
LOA:	41	27.5
LWL:	36	24
BOA:	5.54	4.06
BWL:	5.05	3.45
DRAFT:	1.65	1.10
L/B:	7.1	7.0
B/T:	3.06	3.1
AREA WS:	192	86.7
AREA HULL:	462	215
PRIS.COEF:	0.65	0.65
BLK.COEF:	0.48	0.48
DLR:	87	89
LCF:	24.60	16.44
LCB:	23.77	15.93
GM(L):	90.1	61.4
VCB:	-0.54	-0.36
AREA WP:	153	69.3
PPI:	816	370
MT-1:	1905	587
HULL CG X:	22.26	14.97
HULL CG Z:	+0.81	+0.50
REC.SAIL AREA:	500	400
OVERALL BEAM:	22	15

NOTES

1. All of above figures are for one of the two hulls used in catamarans. Thus total displacement of the 27.5 is 5508 pounds. Total PPI = 740 lbs. per inch for the 27.5, etc.

2. Overall beam of two hulls connected by beams and a bridge deck is usually taken at about one-half the LOA - perhaps a bit more for work boats.

3. Recommended sail area is for light air conditions such as usually obtain in Gulf of Mexico and near-Atlantic and near-Caribbean waters.

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DESIGN OVERVIEW AND ECONOMIC ANALYSIS OF THE AQUARIA 38 SAIL ASSISTED FISHING VESSEL

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The modern era of fisheries development has been marked by an increasing awareness of energy efficiency. As fuel prices have continued to rise, energy efficiency has become nearly a developmental equivalent for economic efficiency. At the same time, world population pressures have increased the demand for marine protein so that harvest of under utilized resources has become an increasing priority. In developing nations, the quandry between need for protein and deficit balance of payments created by petrochemical imports has plagued all concerned.

One solution to these problems involves the appropriate reintroduction of sail as an auxiliary power source. We say auxiliary here because sail alone tends to limit access to the resource, both temporally and geographically. As an auxiliary, however, sail simply enhances profitability of existing fishing operations. Although there has been much discussion of the sail assist concept, it has centered on the reuse of historical hull forms and sail plan. Other applications of sail assist have concentrated on larger vessel sizes or specialized fisheries. The results of these efforts are not generally available to the fishing industry as a whole.

The present discussion centers on one sail-assist project applied within a working fleet in Florida. The vessel was designed with a Caribbean application in mind. To date four of these vessels have been built and fished in the southeastern United States as part of the Thompson Management longline fishing fleet.

The development of the Aquaria concept was an attempt to combine proven sail and fishing technologies in a form that would address our own internal needs for greater control of fish production costs and yet produce a vessel which would find a market both domestically and in developing countries. To this end, proven simplified sail technology was employed in conjunction with a modern hull form to provide an excellent fishing platform.

As mentioned previously, the real impetus for this project was the experience gained in the Thompson Management/Beeline Seafoods fleet of Thompson 60 and 90 footers. A drastic rise in production costs in the longline fishery between 1982 and 1983, coupled with a weakening resource, was forcing us to relocate our longliners to other fisheries. The development of the Aquarias was intended to provide a vessel which could continue to exploit longline resources cost-effectively. Subsequently, since the operations of these boats occurred within the same diverse, profit oriented fleet, we were also able to evaluate the relative merits of the fisheries involved as well as the vessel types.

Some background on the history of the project may be of interest in order to bring out the degree of collaboration between state of the art sail and fishing technologies that has been employed. The genesis of the project came from the U.S. Virgin Islands Saltonstall-Kennedy funded fishery project. Information and expermences gained there and throughout the Caribbean indicated that, although many of the more developed areas were converting from sail and sail assist to pure engine powered fishing boats, this conversion was not being successful. The reasons for this lack of success were various and included:

1. Small outboard powered boats, which extend the range of fishing activities have proven unreliable and expensive to operate;
2. Larger inboard powered vessels were expensive to purchase and operate and maintenance was generally unavailable to keep them running.
3. Expansion of fishing activities resulted in resource overexploitation which made the vessels become uneconomical.
4. Increased fuel utilization and import of boats created economic problems within the country in terms of hard currency necessary for import of fuel and boats.

The Aquaria project attempted to address some of these problems. The boat itself was largely the result of the stimulus and initiative of Tom Worrell, who assembled a design/application team which consisted of the authors and the respective staffs of Thompson Trawlers, Aquarian Research, Thompson Management, and Beeline Seafoods. The boat developed by this team, then was the product of a naval architect with a history of successful yacht designs, a builder of commercial fishing boats, a fleet operator/seafood company and a fishery biologist with experience in developmental requirements of small scale and underdeveloped fisheries.

The design team considered that increased efficiency could be obtained in a variety of areas, and the end product should be a vessel that could be easily operated in lesser developed areas but would still have sufficient catching power and trip capacity to be profitably operated within our own fleet operations. The areas that we concenntrated on for improvement were:

1. Speed and Power Reduction. Our other fleet operations indicated that we needed to have a 200 mile radius of operations in order to consistently find fish. An 8 knot hull speed allowed us to have this range. Analysis of resistance curves (figure 1) indicated that greater speeds would increase power requirements significantly. In order to provide the sail assist with enough carrying capacity over the range, a 6500 lb. (fish weight) fish hold was included in the design requirement.
2. Improved Propulsion and Auxiliary Power. The boat was designed to minimize power requirements through hull form and sail assist. Although the vessle is designed for motor sailing, it is fully capabile of reaching hull speed under either power source.
3. Improved Fish Targeting. We wanted the Aquaria boats to be competitive in catching power with our larger boats. Consequently, we incorporated many of the

fish finding options that had proven valuable on our 60 and 90 ft. boats wherever possible. These included: radar,colorscopes,loran,plotters,temperature sensors, single side band and VHF radios.

The design parameters selected to address these problems were as follow:

1. An overall length between 34 and 38 ft.
2. A desired speed of 7.5 to 10 knots under power.
3. Similiar performance to (2) under sail alone.
4. Fuel consumption in the 1 gph range.
5. Fuel capacity to be 200 gallons.
6. Fresh water capacity to be 100 gallons.
7. Accomodations for 3 to 4 men for 5 days.
8. Crew requirements of 2 or less.
9. Fish hold of 5000 lbs. or better.
10. Half load draft not to exceed 5 ft.
11. True multipurpose fishing capability.
12. Bollard pull of 3500 lbs.

In response, preliminary drawings and specifications were produced for a vessel close to the upper end of the specified variables in terms of overall length. This was deemed necessary to meet the demand for speed and hold capacity. We also felt that a hull form utilizing a modern airfoil section keel and skeg hung rudder would best satisfy the requirements of speed under sail and windward performance.

In addition, the hull form would have to be rather round in section to keep wetted surface to a minimum. This approach would allow the vessel to perform well in light wind conditions with no sacrifice to performance in stronger winds.

Stability was recognized as a key dimension of this design. Regard for the commercial fisherman, who may be unfamiliar with the subtleties of a sailing vessel, combined with concern over the possibility of shifting cargo, urged us to create a most stable platform (figure 2).

The final design (figure 3) incorporates a relatively shallow canoe body with a moderately high center of buoyancy. This, combined with a low center of gravity and greater than average beam insures good stability. Sections are typically round throughout while waterlines are fine forward and full aft. These design elements are intended to allow for good performance while maintaining the volume necessary for the fish hold.

The principal dimensions of the final design are as follow:

L.O.A.	37' 6"
L.W.L.	33' 4"
Beam	13' 0"
Draft	5' 0"
Displacement	25000 lbs.

Ballast	7000 lbs.
Hold Capacity	6500 lbs.
Sail Area	734 sq. ft.

Fuel capacity worked out to be 240 gal., while fresh water capacity is 100 gal. The accomodation plan (figure 4) allows for four births, an enclosed head and a galley forward. A small dinette and steering station are located in the pilot house.

In an effort to provide a substantial amount of sail area in terms of economy and ease of handling, a self furling sloop rig was chosen (figure 5). Other rig types such as free standing cat rigs were examined and eventually disregarded as less favorable. A gallows frame aft supports the backstay and provides a base for the main sheet. The standing rigging consists of upper shrouds, single lower shrouds, headstay and backstay. The jib is roller furling and the mainsail furls into the mast. The 734 sq. ft. sail area provides adequate performance for light air conditions. Configured as relatively short, yet broad on the base, the saail plan combines power and stability.

Computer predictions for sail performance (figure 6) indicate an expected speed at 7.25 knots to windward at 29 degrees apparent wind angle in a wind strength of 20 knots true. Off wind performance for the same wind strength is 8.80 knots at 96 degrees apparent wind direction.

The construction of the Aquaria 38 sail assist is of hand layup F.R.P. A split mold is used for the hull and seperate molds are used for the main deck and the pilot house. Hull reinforcement is gained through the use of strategically placed F.R.P. hat section stringers and floors. Construction scantlings and materials are to American Bureau of Shipping rules for building and classing reinforced plastic vessels. The engine beds are canted 1.5 degrees off center to starboard to provide for removal of the propeller shaft past the rudder skeg. The off center angled shaft counteracts, to a great extent, any "torque steering" tendencies due to propeller rotation. The propeller is exposed between the keel and the rudder skeg and is support by a "V" shaped strut near its hub.

Ballast is lead and V.C.G. is 4.5 in. below L.W.L. in a half load state. When fully loaded, the cargo's C.G. lies over the designed C.G. Trim is maintained throughout the vessel's load range.

The main power plant may vary in size from approximately 40 h.p. to 175 h.p. The larger engine delivers the expected bollard pull values while a 60 h.p. engine will power the boat at 8.25 knts. while burning less than 7/8 gallons per hour.

Observed speed under sail has been in accordance with our computer predictions. Her performance rivals sailing yachts of similar length and displacement on all points of sail. Performance under power yields a top speed in excess of 9 knots with a Perkins 4-154, 60 h.p. main engine. Her motion is quite normal and steering is excellent with her large rudder and balanced sail plan. Stability is what one would expect: the Dellenbaugh angle is 9.07 degrees.

With no cargo on board, the very buoyant hull form gives a slightly corky ride, which is to be expected, but as load increases, stability, which is already very good, improves and the vessel becomes more seakindly. The hull stiffness also provides a very stable fishing platform.

The captains of the four Aquarias all lacked sailing experience prior to taking over the boats. They have all learned quickly and can single hand the boat in all wind and sea conditions so that steering is easy on all points of sail. Light weather performance has been most encouraging since a large portion of the worlds underdeveloped fisheries occur in areas of light to moderate winds.

The Thompson Management fleet consisted of 17 vessels at its peak, inluding four 38 ft. sail assists, eleven 60 ft. Thompson Trawlers, and two 90 ft. Thompson Trawlers which operated as a scallop catcher/factory and a squid catcher/processor. We built all of the boats in Titusville, Florida. The fleet was involved in longlining for surface and bottom fish, groundfish trawling, squid trawling, shrimp trawling, and scallop trawling. The fleet management facility was operated in Port Canaveral, Florida. Fleet operations occur primarily in the Gulf of Mexico and Mid Atlantic with the exception of the shrimping which is in Guyana. Two of the 60 ft. longliners made one exploritoryy trip to Jamaica in conjunction with Antillean Pride, a Jamaican canning company.

Fishing performance of the Aquarias has been the most encouraging aspect of the project. Initial trials with Aquaria I included:

1. Bottom longlining with 6 mile sets of up to 3500 hooks per day.
2. Trolling under sail and motor sailing with 6 lines. We have begun to explore using Norh Pacific trolling methods with up to 15 or more lines.
3. Shrimp trawling. We have successfully pulled a 56 ft. net and doors. At 20 plus knots of wind, the can be pulled at 2.5 knots under sail alone.
4. Surface longline, trap hauling, vertical set lines, and gill netting have also been tried successfully.

The most convincing result comes from a comparison of the sail assists to our conventional 60 foot trawler in the bottom long line fishery (Table I). In terms of daily revenue, the sail assist produces nearly 70% as much as the larger boat despite the fact that the 60 footers set twice as many hooks. The reason for this is the fact that the Aquarias can set lighter gear which is more effective. This also shows in in the bait expenses, as the sail assist bait costs only amount to 8.2% of the revenue, compared to 14.6% of the sixties'.

Profitability is the real measure of commercial fishing success. The sail assists produced 75% operating profit while the 60s only produced 52% despite the fact that their capital costs were nearly 5 times greater. In other words, we have found the Aquarias to be nearly as powerful at fishing as the much more expensive larger boat. Additionally, because it requires fewer crew and has lower expenses, a greater operating profit is generated.

As mentioned earlier, fuel efficiency was an aspect emphasized in the design project. The Aquarias' fuel costs were only 5% of total revenue compared with 12% of the sixtys'. When conversion of fuel to fish is considered (Table II), the sail assists were over 2.5 times as efficient as our fleet of sixtys. When the Aquarias were compared to other fisheries, both with our own boats and other figures from the literature, the Aquarias provided one of the most fuel efficient options available.

This table also shows what every fisherman knows and that is that energy efficiency does not necessarily equal economic efficiency at every turn. Otherwise there would be little production of high priced,energy intensive, products like shrimp, scallops and sword fish which can provide revenue greatly in excess of the increased fuel involved in their production. Despite this fact, our experience in the application of sail assist from the drawing board to the very real world of operation of commercial fishing boats has been extremely encouraging. Savings from energy efficiency have been obtained, but the boats have proven to be flexible, powerful fishing machines that can hold their own when compared with larger more expensive vessels involved in the same fisheries. It is our opinion that considerable promise and profit are available through the appropriate use of sail assist.

Figure 1. Hull resistance, speed and power requirements for the Aquaria sail assist.

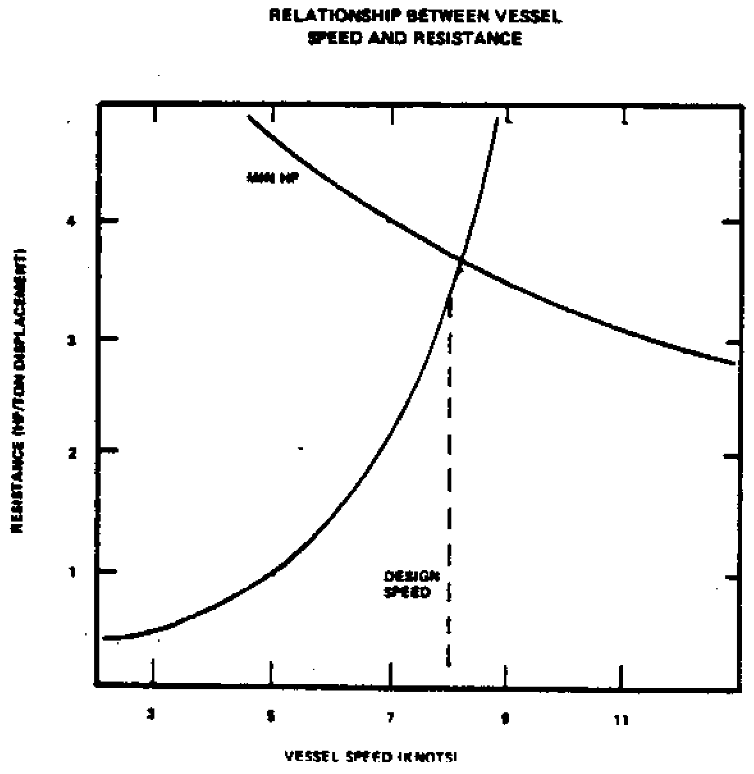


Figure 2. Righting arm vs. heel angle relationship for for the Aquaria sail assist.

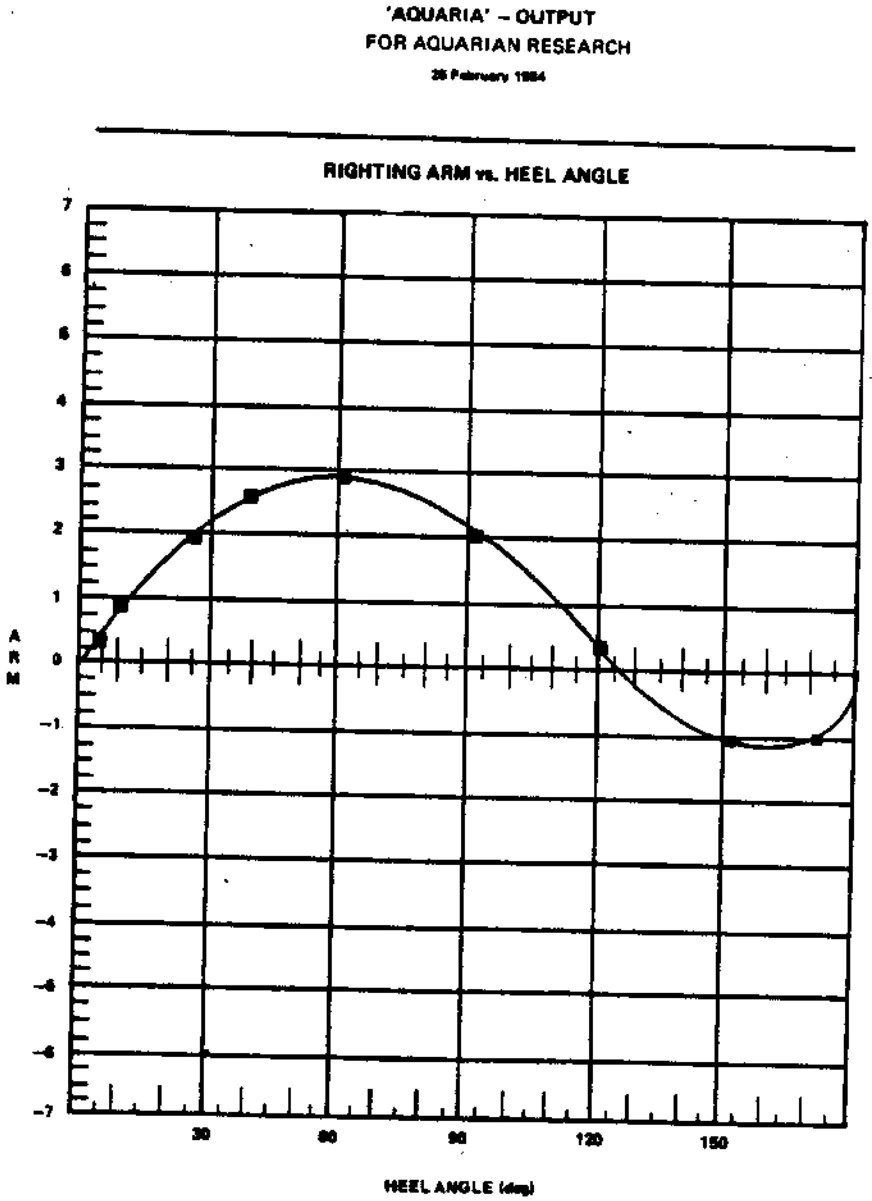


Figure 3. Final design and layout of the Aquaria sail assist showing the shallow canoe body, skeg hung rudder and ample after deck.

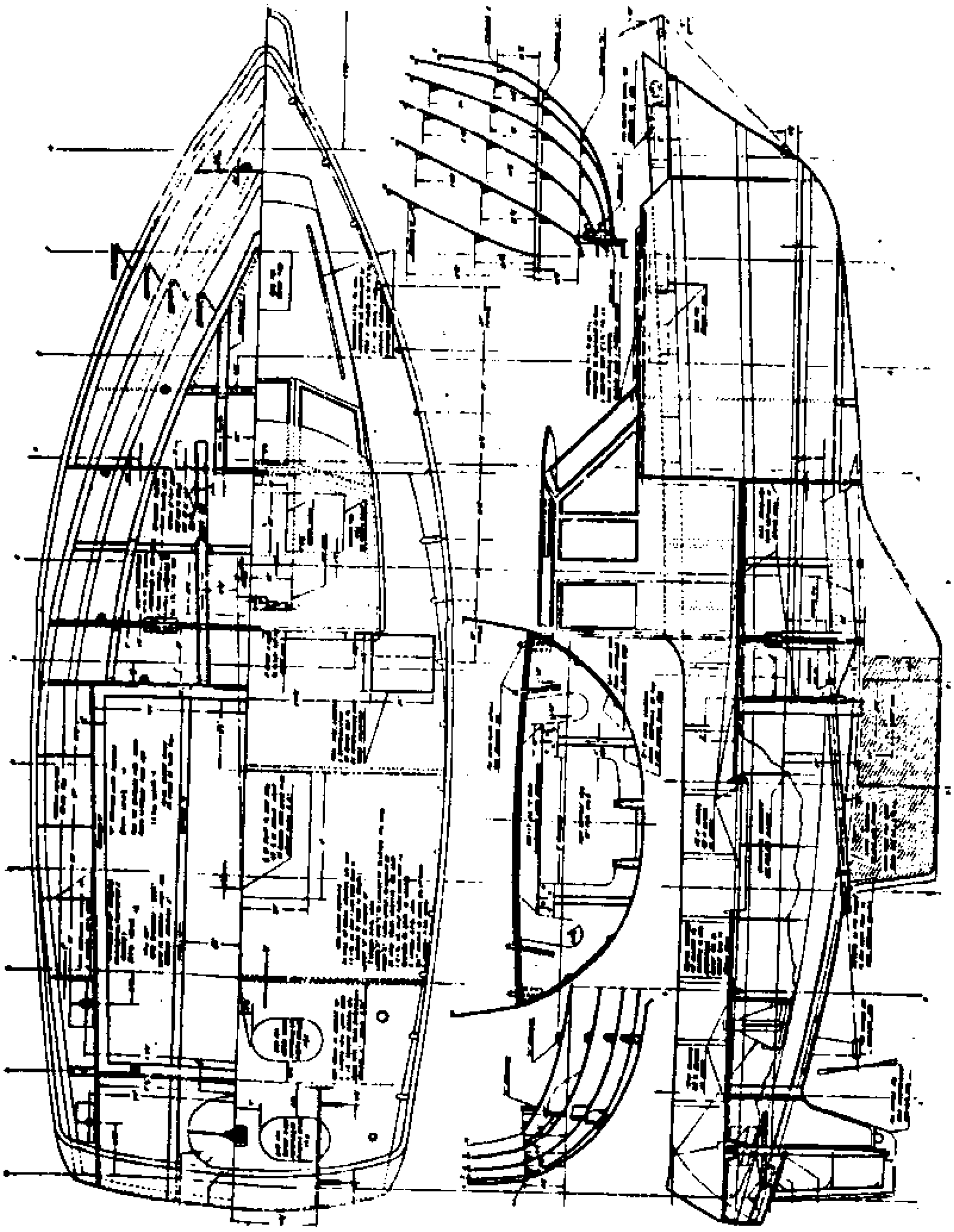
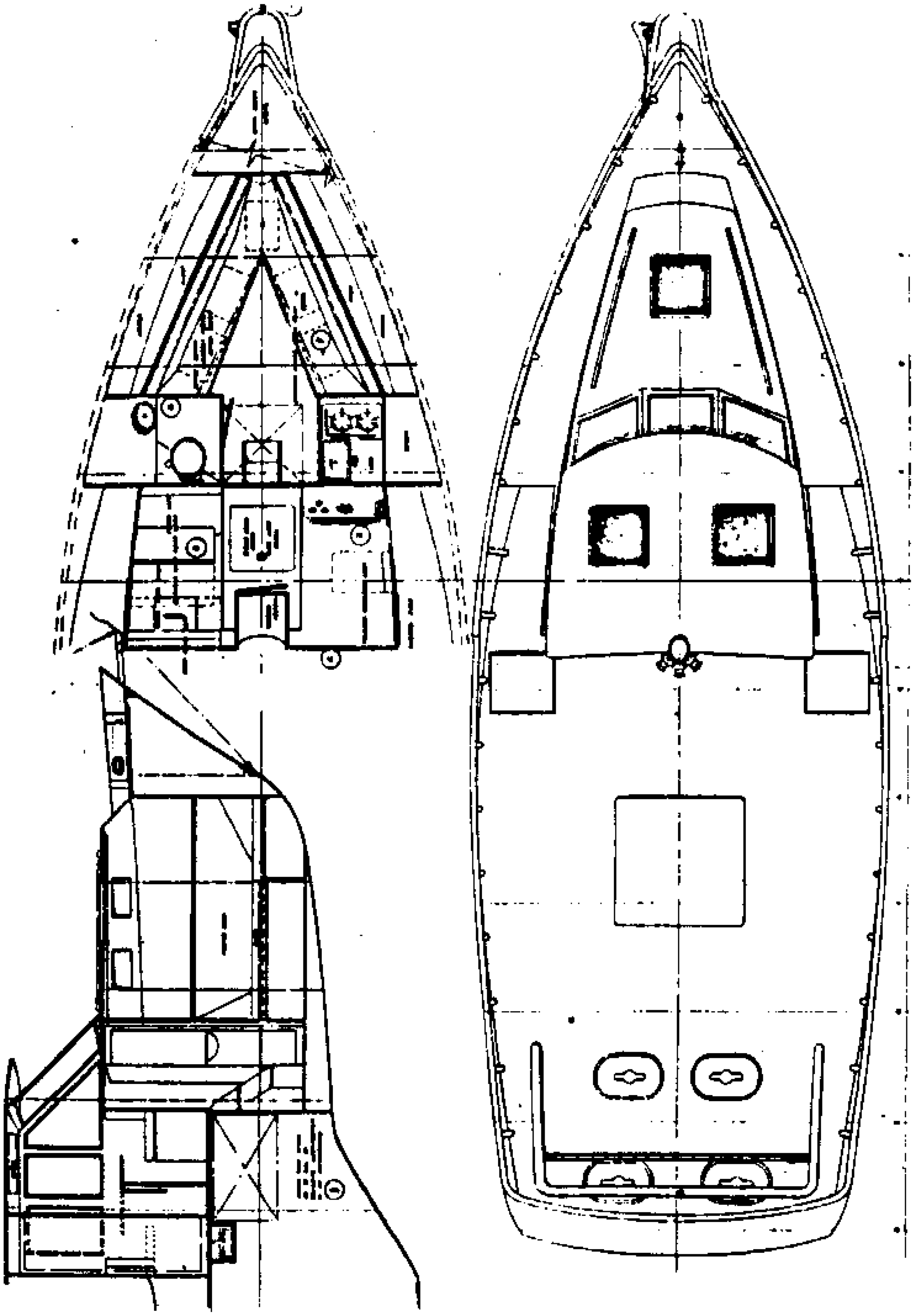


Figure 4. Deck layout and accomodations plan for the Aquaria sail assist.



Figrue 5, Sail plan for the Aquaria carries 734 sq, ft, of sail on a sloop rig with roller furling jib and main sails for ease of handling.

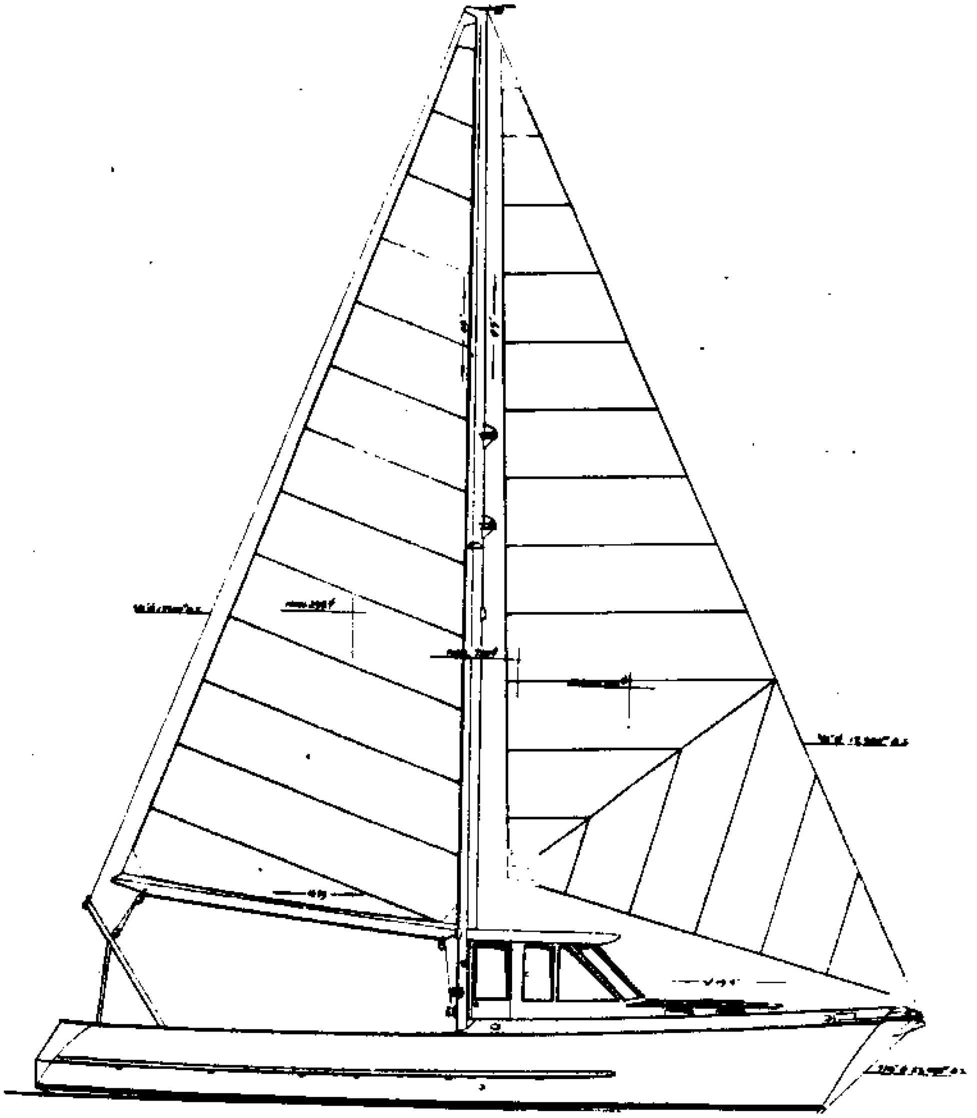


Figure 6. Polar plot of predicted sailing performance of the Aquaria suggested an easily driven, relatively close-winded performance.

'AQUARIA' – OUTPUT
FOR AQUARIAN RESEARCH
26 February 1984

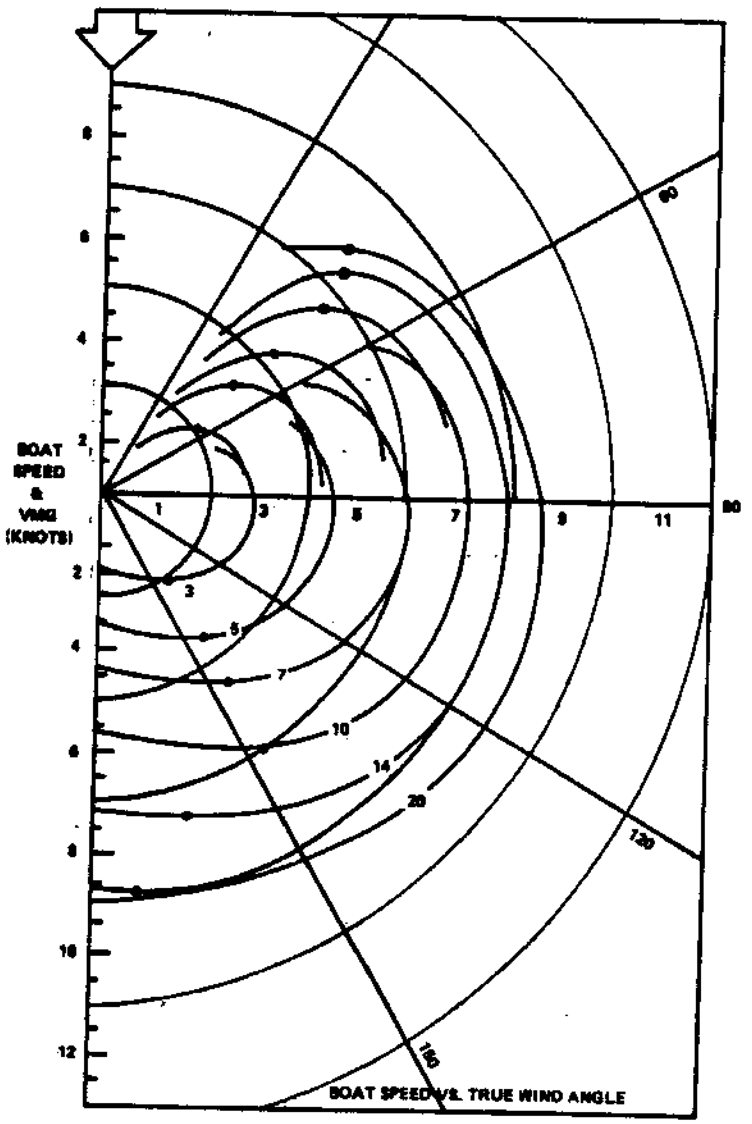


Table I. Comparison of 38 ft. sail assist to 60 ft. trawler/longliner.. Results are shown as percent of total revenue consumed by various expense categories. Total daily revenue is shown in the right hand column.

COMPARISON OF EXPENSES (GIVEN AS % OF TOTAL REVENUE) BETWEEN 60 FT. T BOTTOM LONG LINER AND 38 FT. SAIL ASSISTED LONG LINER

	TOTAL EXPENSES	BAIT	FUEL	DAILY REVENUE
60 Ft. Long Liner	47.8	14.6	12.1	$1348
38 Ft. SA Long Liner	23.6	8.2	5.1	$ 942
Ratio – 60 ft./38 ft.	2.03	1.78	2.37	1.43

Table II. Comparison of energy conversion (Kcal of fuel/ Kcal of product) for various fisheries that the Thompson Management fleet was involved in.

COMPARISON OF ENERGY CONSUMPTION (KCAL OF FUEL/KCAL OF PRODUCT)[1] FOR SELECTED FISHERIES

FISHERY	Kcal Input/Kcal Output
Northeast Ground Fish Trawler	4.1
Florida Spanish Mackeral Gillnet	7.8
TM Sail Assisted Bottom Longline (38 ft.)	7.9
TM Squid Trawler/Processor (90 ft.)	15.3
Virgin Islands Trap Fishery (22 ft.)	18.5
TM Bottom Longliner (60 ft.)	20.3
TM Scallop Dragger/Processor (90 ft.)	39.5
TM Swordfish Longliner (60 ft.)	49.8
TM Shrimp Trawler Guyana (60 ft.)	191.7
Fla./Gulf Shrimp Trawler	198.0

[1] Diesel fuel = 34,650 kcal/gal.
Fish = 0.428 kcal/gm.
Shrimp = 0.604 kcal/gm.
Scallop = 0.672 kcal/gm.

PERFORMANCE MEASUREMENTS
OF A 25 FOOT SAIL-ASSISTED FISHING VESSEL

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ABSTRACT

A condensed version of the original assessment of the performance and fuel savings potential while on passage for a typical small European sail-assisted fishing vessel is presented. Various tacks and engine output levels were compared on the basis of reduction available in the most efficient propulsive mode. The test vessel shen rigged in the sail-assist arrangement consistently proved more fuel efficient then when power alone was employed, as long as a favorable tack was maintained. The evaluation of the sea trials, while admittably conducted in ideal conditions, showed that fuel savings of more than 90 percent while on passage could be realized. The extent of the benefits was dependent on many variables, including wind conditions, the vessel course, proper sail trim, and required engine output to maintiam a particular ship speed. Based on the findings of this study and the obvious need of commercial fishermen to cut operating costs, the use of sail-assisted fishing vessels seems warranted and demonstrates at least one technique available for reducing the major expense, fuel cost.

INTRODUCTION

Sharp increases in oil prices, particularly in the early seventies prompted the consideration and adoption of energy conservation measures by the United States fishing industry. These included more efficient gear and less energy consuming fishing methods. One important development was the use of sail-assistance to provide additional power(1).

It has been shown that the use of sail-assistance in conjunction with fossil fuel power can be an economically favorable method of vessel propulsion. Early trials in the U.S. on various vessel types have shown the possibility of reducing power output by thirty percent or more, fuel use by over twenty percent, and transit time by more than ten percent through sail-assistance(2). Simply stated, the sail-assist vessels are showing a better return on capital than the mechanized vessels, a development from which the fishing industry could benefit immediately. The price of fuel oil has become the largest single expense of shipowners. While presently there is a slight decrease in the price of crude oil, it can be shown that fueloil during the last ten years has continuously risen in price overall and is estimated to cost as much as two hundred dollars per barrel in the not distant future(1). The rise in fuel prices has caused the economics the fishing operation to become out of line with the economics of the market place. This has resulted in the need to reduce fuel consumption and hence costs(3). The use of sails aboard fishing vessels is advocated to increase the financial viability of these vessels(4).

Sail power not only offers a means of reducing the power needed, and hence fuel consumption on passage, or the time required for passage making, but also provides an emergency means of propulsion should the principal power system fail(3).

TEST VESSEL

The vessel selected for the project, to act as a representative commercial fishing vessel, was a Fisher design Fairways Potter 25. This type of vessel is commonly used in crab and lobster pot fishing within Europe. The hull design is of North Sea Trawler type, with distinct f'o'csle and a wheelhouse abaft amidships. The dimensions are as follows:

Length overall	25feet 3inch
Beam	9feet 4inch
Length, waterline	21feet
Draft	3feet 9inch
Displacement	4.5tons

The existing power source is a 36 horsepower, maximum continuous bhp, Volvo-Penta model MD3B, three cylinder, four stroke diesel. The reduction gear ratio is of MS type with a ratio of 1.91:1.

The standard package, shown in figure 1, is strictly a power vessel equipped with a 41 square foot steading sail for damping rolling motion and station keeping while working gear.

The sail-assist arrangement, shown in figure 2, is a ketch rig and comprises 245 square feet of sail.

TEST PROCEDURE

The actual test methodology was developed through preliminary trials and the most satisfactory method to obtain the needed data adopted. The test procedure involved steaming between known points at various engine revolutions with engine power alone, measuring ship velocity. The runs were then repeated with engine power plus sail-assistance at engine revolutions corresponding to those in the power alone runs in a variety of wind conditions.

Engine revolutions ranged from eight-hundred, just above idle speed, to nineteen-hundred and ten rpm's but because of previous research showing a decreasing advantage to sail-assist with high engine power, emphasis was placed on readings of eight-hundred to thirteen-hundred(5).

Only a few sail-only trials were conducted as it is unlikely that "pure" sailing ships without any installed power will prove advantageous in the near future(6). It dows however provide an indication of power available to the sails.

PRESENTATION AND ANALYSIS OF RESULTS

Method of analysis-

The data collected during sea trails was analyzed in a series of steps leading to the final comparison of engine power alone, and sail-assisted, propulsion in terms of fuel consumption. All data was grouped according to wind speed, engine speed, and the tack of the vessel.

The sailed courses from which data was recorded represented four tacks (true wind direction to the vessels course) that could be used to generalize vessel performance on almost all tacks. The tacks tested included the vessel traveling on a broad reach, a reach, close hauled, and running with the wind. Wind conditions varied from about five to twelve mph, closely representing the minimum speed of the wind required for effective sail usage and the maximum wind speed for setting all sails (for certain tacks) without reefing.

Engine power-alone data was obtained by running the trials usually early in the morning while waiting for the wind to increase for the sail-assisted runs and represents generally a no-wind condition.

Results-

The results obtained from data analysis involved plotting and comparing vessel performance for power-alone and sail-assisted propulsion. While performing the sea trials the advantage of using sail-assistance was first realized as a reduction in engine revolutions while maintaining a previously attained power-alone vessel speed.

Vessel Speed and Associated Engine Revolutions- (figure 3)

The results demonstrate the ability to maintain a predetermined desired vessel speed at much reduced engine revolution levels with sail-assistance than could be obtained in the engine power-alone mode. This reduction of revolutions was shown to vary from about eight to forty-three percent. The extent of these reductions is dependent on the course sailed, the wind speed and direction to the course, and also the experience of the crew.

The largest of engine revolution reductions occurred at the lower values of engine revolutions and vessel speed. This is to be expected since the vessel is capable of making two to three knots while under sail-power alone in moderate winds. At the lower engine revolutions, which produce only slightly greater vessel speeds, the sail-assistance represents a greater contribution to the powering than at the higher revolutions where the engine has a much greater effect.

Vessel Speed and Associated Fuel Consumption- (figure 4)

The final evaluation of the test vessels performance deals with the amount of fuel required in the engine power-alone mode and reduction in the amount of fuel consumed when using sail-assistance.

To accomplish the comparison of propulsion modes the brake horsepower was calculated from the propeller dimensions and a standard Bp-δ chart. The actual fuel consumption was calculated using an engine diagram provided by the engine manufacturer.

The results demonstrate considerable reduction of fuel consumption can be accomplished by utilizing sail-assistance in a motor-sailing propulsion mode. The actual extent of fuel savings is also dependent on many variables including vessel geometry, course sailed, crew experience (efficient sail trim), desired vessel speed, sea state, and most importantly, wind conditions. The test vessel showed the greatest fuel saving potential when in the sail-assist mode at a low engine speed in winds of high true velocity. Fuel savings varied from about thirty-seven to greater than ninety-two percent depending on the values of the proviously described parameters.

CONCLUSION

The data collected and anlayzed in this study produced results that show with favorable wind conditions and by assisting the vessel's main power plant with sails it is possible to reduce the engine revolutions over forty percent while maintaining desired ship speed. From known propeller characteristics and the appropriate Bp-δ chart it was possible to convert the RPM reductions into power reductions and then to determine range of fuel savings. Power required by the propeller at particular vessel speeds and the associated fuel consumption showed potential reductions for both at over ninety percent for a vessel shile on passage. These advantages decreased rapidly as the normal vessel cruising speed, determined as velocities in the $V/\sqrt{L}$ range of 1.1 to 1.2, was surpassed.

Winds that produced desirable sail efficiency varied from fifty to three hundred and ten degrees to the vessel heading. Setting of the full compliment of sails was limited to wind speeds of three to fifteen mph on all tacks with exception of the vessel on a run. Beyond these wind speeds reducing the sail area (reefing) is required and further study is needed to examine benefits available in this case. Winds on the beam of the vessel, especially a broad reach tack (100-129 degrees), consistently demonstrated the most efficient results, obviously producing optimum lift and drag relationships between the sails and the hull. This tack also allowed a potential sail-assisted vessel speed increase of over one-hundred percent that obtained with engine power-alone in the lower engine revolution range.

A tabulated summary of the results acquired is presented in tables 1 and 2. These agian show the considerable fuel savings available in the full range of typical vessel speeds through sail-assistance. A further analysis determined the extent of fuel savings that could be expected depending on the percentage of trip time the sails were employed. Wind polars are illustrated in figure 5 displaying the increased vessel speed on various tacks that were measured aboard the test vessel.

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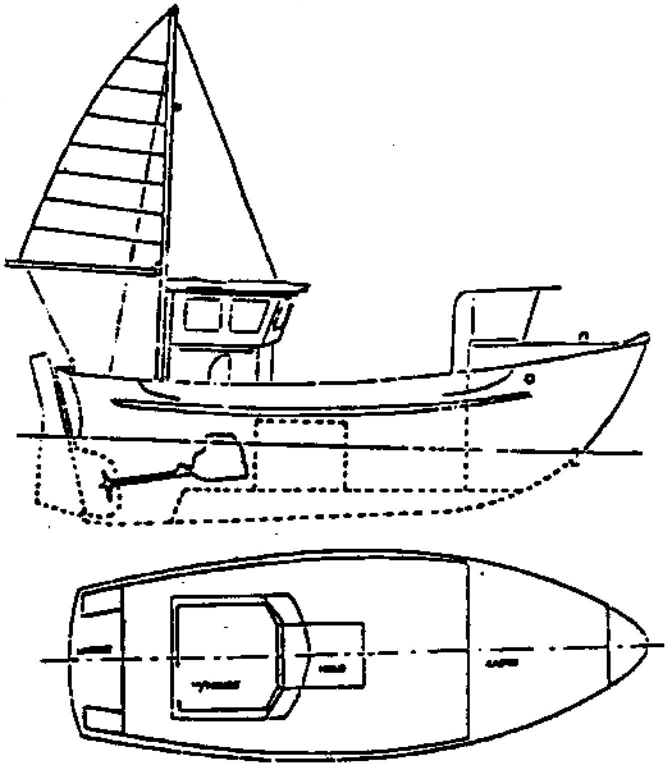
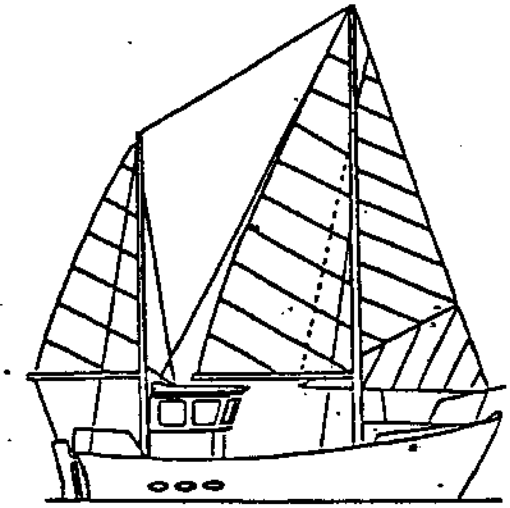
5. Lange, K., Schenzle, P.: 'Some Aspects of Sail Power Applications in the German Sea Fishery'. International Council for the Exploration of the Sea, 1981.
6. Letcher, J.S.: 'Optimal Performance of Ships Under Combined Power and Sail'. Journal of Ship Research, September 1982.



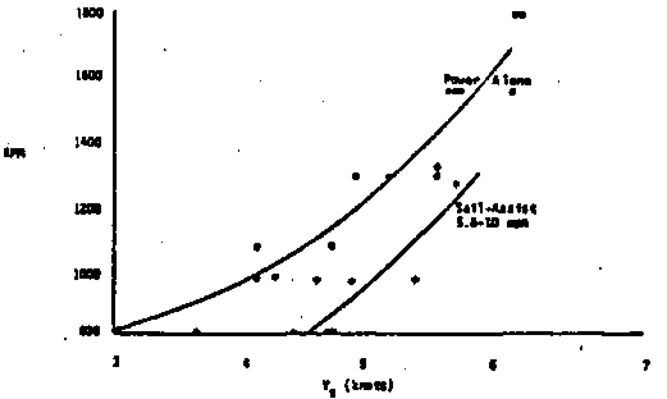
Fisher Design Fairways Potter 25, North Sea Trawler hull, LOA-25'3", Beam 9'4", Draft 3'9", Displacement 4.5; Standard package equipped with 41 sq. ft. steadying sail. Power source is a Volvo Penta, 36 hp, model MD3B, 3 cylinder, 4 -stroke diesel.

Figure 1.



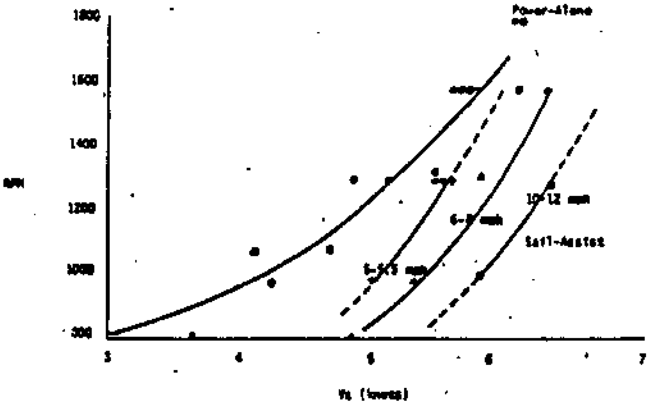
Sail-Assist Arrangement, Ketch rigged, with 245 sq. ft.

Figure 2.



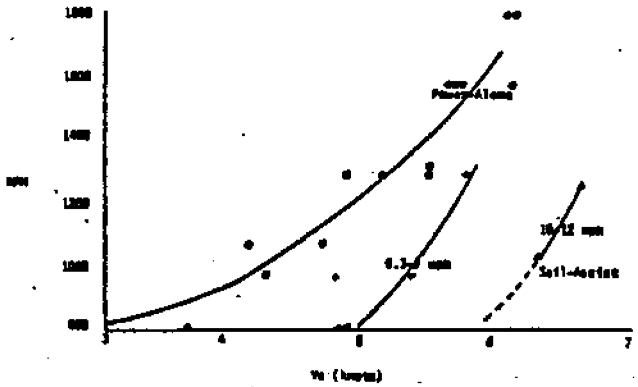
Engine RPM versus Vessel Speed (Close-Hauled)

+ Sail-Assist, V_T = 5.8-10 mph
● Power-Alone



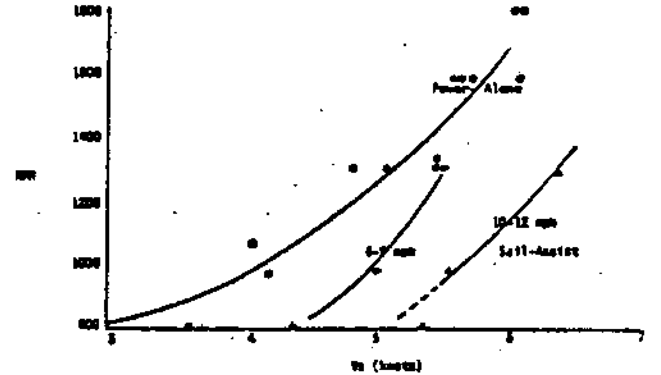
Engine RPM versus Vessel Speed (A Reach)

+ Sail-Assist, V_T = 5-5.5 mph
▲ Sail-Assist, V_T = 6-8 mph
○ Sail-Assist, V_T = 10-12 mph
● Power-Alone



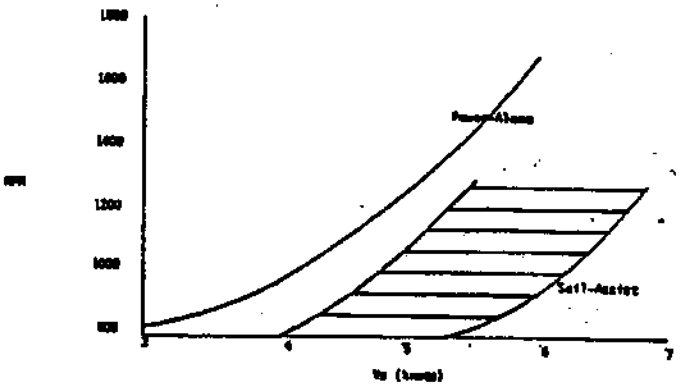
Engine RPM versus Vessel Speed (A Broad Reach)

+ Sail-Assist,V_T = 6.3-9 mph
△ Sail-Assist, V_T = 10-12 mph
○ Power-Alone



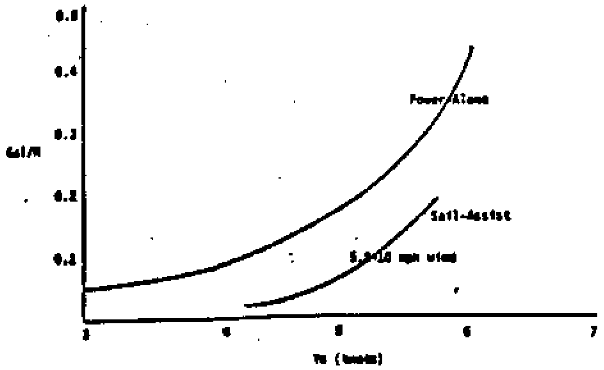
Engine RPM versus Vessel Speed (A Run)

+ Sail-Assist, V_T = 6-9 mph
△ Sail-Assist, V_T = 10-12 mph
○ Power-Alone

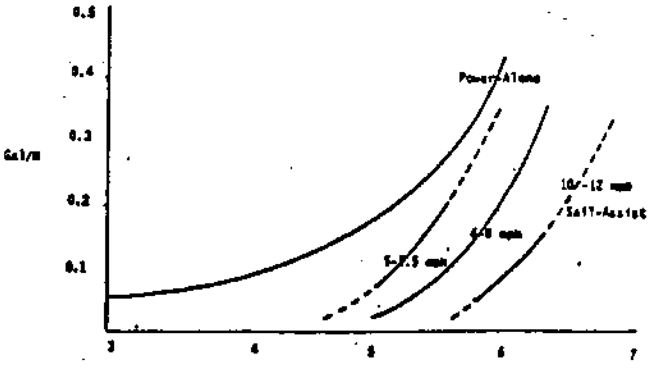


Engine RPM versus Vessel Speed - Range of all Data

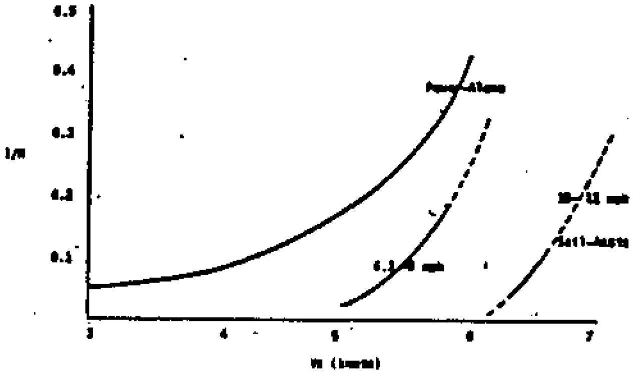
Figure 3.



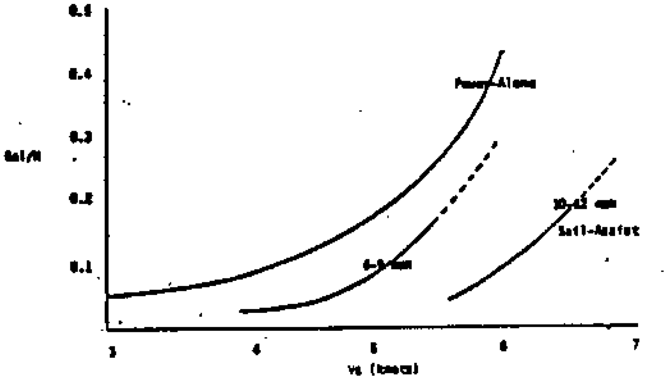
Fuel Consumption versus Vessel Speed (Close-Hauled)



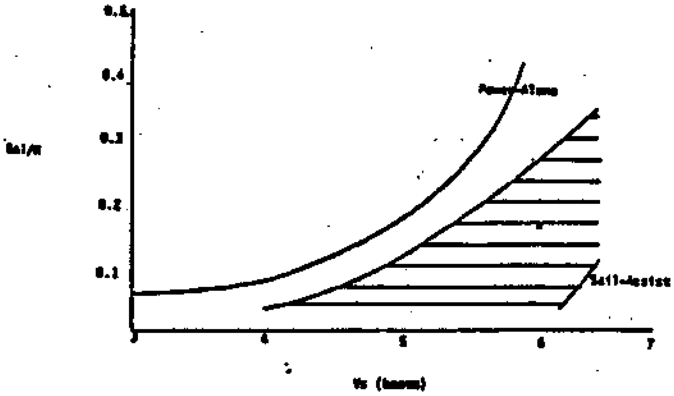
Fuel Consumption versus Vessel Speed (A Reach)



Fuel Consumption versus Vessel Speed (A Broad Reach)



Fuel Consumption versus Vessel Speed (A Run)



Fuel Consumption versus Vessel Speed - A Range of All Data

Table 1.

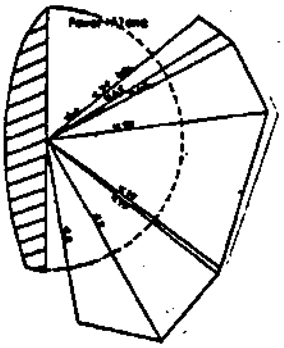
Reduction in Engine RPM's and Fuel Consumption, a Summary

	RPM's			Fuel Consumption (Gal/M)		
Vs	Power Alone	Sail-Assist	%RPM Red.	Power Alone	Sail-Assist	% Fuel Red.
4.0	952.9	779.5	18.2	0.078	0.02 - 0.032	58.9 - 74.3
4.5	1070.5	810.3-835.4	21.9-24.3	0.106	0.027 - 0.036	66.7 - 75.0
5.0	1221.1	999.9	18.6	0.167	0.067	59.8
5.5	1412.4	947.5-1290.1	8.6-32.9	0.275	0.035 - 0.180	34.5 - 88.8
6.0	1637.8	1118.0-1373.2	16.1-31.7	0.424	0.092	78.3
6.5	1890.3	1180.0-1340.4	28.6-37.5	0.606	0.080 - 0.161	73.9 - 87.3

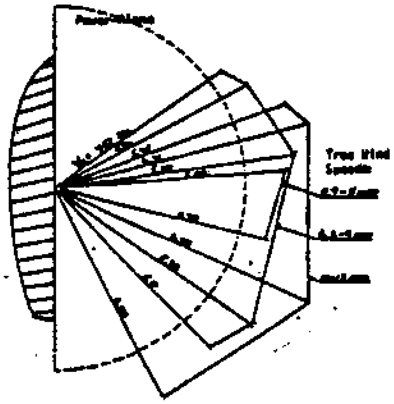
Table 2.

Reduction in Fuel Consumption, Depending on Sail Time, For Service Speed (Vs = 5.0-6.5 knots)

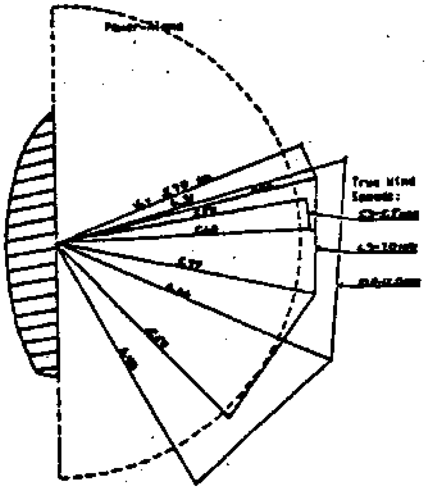
	% Reduction Fuel Consumption			
% Sail-Assist	Close-Hauled	Reach	Broad Reach	Run
100	53-63	40.7-81.4	66-84	38.5-86.0
20	10.6-12.6	8.2-16.3	13.2-16.8	7.7-17.2
40	21.2-25.2	16.3-32.6	26.4-33.6	15.4-34.4
60	31.8-37.8	24.5-48.9	39.6-50.4	23.1-51.6



Polar Curve for 800 RPM with varying Wind Speeds (5.8-10 mph)



Polar Curves for 1000 RPM with Varying Wind Speeds



Polar Curves For 1300 RPM with Varying Wind Speeds

Figure 2.

ROTOR PROPULSION FOR THE FISHING FLEET

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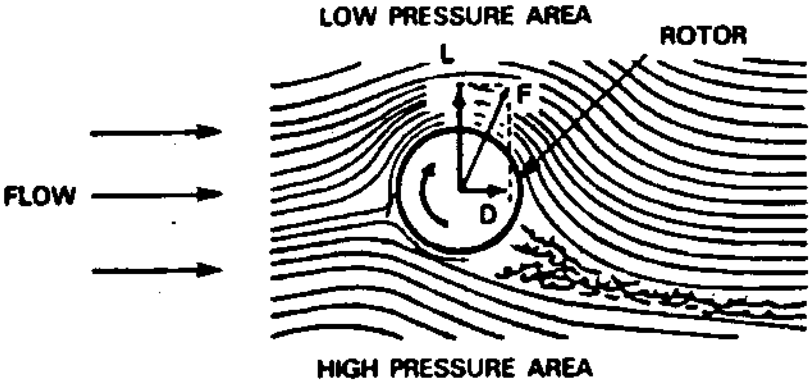
ABSTRACT

In 1924 Anton Flettner sailed the first rotor ship, Baden-Baden, from Germany to New York during which she encountered hurricane force winds with no adverse effects. Since that time, little practical application has been made of rotor sails. However, in recent years, U.S., as well as French, Swedish, German and Russian investigators, have revisited the rotor including extensive model testing and full scale testing on vessels of less than 100 feet. The relatively low cost, simplicity of operation and low maintenance cost make the rotor an ideal sail assist choice for the commercial fishing fleet as well as large and small cargo vessels.

The paper reviews the history of the rotor and its salient characteristics. In addition the following are provided: A rotor system design for a typical shrimper, a methodology for rotor sizing, and guidance on how to build a rotor system using readily available materials and tools.

What is the Magnus effect?

The first question usually asked is: "Just what is the Magnus effect?" One answer would be an accelerated airfoil. Lift is developed by spinning a cylinder at right angles to a flow (air or water stream). As the speed of the cylinder is increased, the pressure decreases on the side of the cylinder where the natural flow and spin-induced flow combine. The decrease in pressure creates a lift and the lift increases as the surface velocity increases, following Bernoulli's Therom. Figure 1 shows the Magnus effect in graphic form.



AS SOON AS THE CYLINDER BEGINS TO TURN, A LIFT FORCE L PERPENDICULAR TO THE FLOW, AND A DRAG FORCE D, IN THE DIRECTION OF THE AIRFLOW ARE GENERATED WITH A RESULTING FORCE F. THE STRENGTH OF THE LIFT DEPENDS UPON THE DIMENSIONS OF THE CYLINDER, THE SPEED OF ROTATION, AND THE SPEED OF THE WIND.

Figure 1. - The Magnus Effect[1]

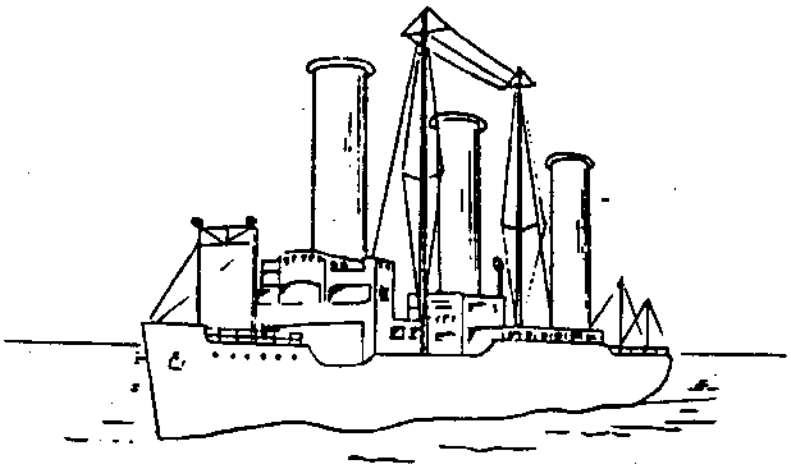
The Magnus effect is a valuable phenomenom because it can generate considerably more lift per unit of projected area than typical airfoil (i.e. wing or sail) forms. This feature permits lift systems to be developed that are typically five to ten times smaller than airfoils with equal lift. The major drawback is that external power on the order of 10 to 20 percent of the output power is required to spin the cylinder.

The History of the Magnus Effect and Magnus Effect Devices 1794 to 1970

In 1794, the Berlin Academy offered a prize for finding out why an artillery projectile would either overshoot or undershot the target when a cross wind was present. Over 50 years later, Gustav Magnus, Professor of Physics at the University of Berlin, conducted experiments with spinning cylinders in an air flow and identified the lift phenomenon which affects all spinning cylinders or spheres and which has since been known as the Magnus effect.

In 1877, Lord Rayleigh published a paper titled "On The Irregular Flight of Tennis Balls." This treatise explained why a "sliced" tennis ball will drop sharply because of the Magnus effect.

Until 1922, although a number of scientists had experimented with the Magnus effect, little, if any, practical application of the phenomenon was considered. In 1922 a German enterpreneur, Anton Flettner, set out to design an auxiliary sail propulsion system for ships. His initial efforts focused on basic airfoil technology. He later discarded the airfoil in favor of Magnus effect rotor propulsion. In the early 1920s, two vessels, were equipped with Magnus effect, wind rotor, auxiliary propulsion. The first vessel, "Baden-Baden," previously a 164 foot three-masted barkentine, was equipped with two rotors. "Baden-Baden's conversion was underwritten by Krupp. The second vessel, "Barbara", shown in figure 2, was equipped with three rotors. Both of these vessels were reasonably successful; however, fuel saving ideas in the 1920s were more technical curiosities than practical necessities.



LENGTH	300'-0"
BEAM	43'-0"
DEPTH (KEEL TO POOP DECK)	27'-0"
CARGO CAPACITY	3000 TONS
MAIN PROPULSION	2 DIESEL ENGINES, SINGLE SCREW, 1060 HP
ROTORSAILS	3
DIAMETER	13'-2"
HEIGHT	56'-1"
ROTOR SPEED	150 RPM
ROTOR DRIVES	3-36 HP ELECTRIC MOTORS (one per rotor)

Figure 2. - "Barbara with Three Rotors[2]

In 1933, Julius Madarasz, of Royal Oak, Michigan, developed a design for a massive, land based, electrical generator system which envisioned large Magnus effect rotors on railroad cars running on a circular track. One cylinder was built and tested with favorable results; however, the capital required to complete the project was not forthcoming.

From 1933 until the 1970s little work was done on Magnus effect devices. However, the fuel price crisis in the 1970s generated considerable renewed interest in the Magnus effect for applications where traditional airfoil shapes are normally used. For more historical data, see reference 2.

Marine Magnus Effect Applications

Since 1970, a number of initiatives have been made concerning the use of the Magnus effect for marine applications. Three of them are of interest to fishing fleet operators: Wind propulsion, rudders and propellers.

Wind Propulsion: Investigators in France, Germany, Russia, and Sweden, as well as the United States, have been vigorously pursuing rotors and other Magnus effect-like devices for the auxiliary propulsion of ships. The Swedish, German and U.S. investigations have been with pure Magnus effect rotors, each having built and tested yacht size rotors. The results of these tests have been lift curves that indicate that Flettner's results did not fully exploit the potential of the rotor. The recent test of a rotor by Wind Ship of Norwell, Mass has demonstrated significantly better performance than that attained by Flettner.

The rotor has distinct advantages over other wind propulsion devices. Some of these advantages are shown in Table 1. The most significant factor is the cost per unit of lift. However, other considerations such as simplicity of operation, the rotor's tendency to be storm-proof, and the relative ease with which a rotor can be telescoped to cope with bridge clearances, further enhanced its utility. Although there remain a few characteristics, which have yet to be fully explored, the rotor is currently a viable wind-assist device for the fishing fleet.

Table 1: Rig Comparision Table.[3,4]

RIG	THRUST FACTOR (CL/\$K PER FT^2)	DRAG FACTOR (% OF USEABLE WIND ANGLES)
SQUARE	9.9	75
STAYED FORE & AFT	18.5	83
SHIN AITOKU MARU	22.5	89
UNSTAYED FORE & AFT (CAT)	23.2	83
AIRFOIL W/SIMPLE FLAP	34.5	92
ROTOR	109.9	89

The Cousteau Organization, with finanical support from the French Government, has recently built and tested a device called the Aspirated Ellipse, which is similar in principle to Magnus effect rotors. This device features an elliptical mast with two slots on its after end, port and starboard, which are alternately covered or uncovered, depending on whether the wind is coming from port or starboard (i.e., on a port or starboard tack). A fan at the top of the ellipse is used to suck air through the open slot, accelerating air on one side of the ellipse and thereby creating lift. The developers claim less power is required per pound of lift as compared to the rotor; however, the ellipse needs to have about twice the projected area as a rotor for equal lift. Unfortunately, the Aspirated Ellipse on Cousteau's test vessel, "Moulin à Vent," broke off in November 1983 during an attempt to cross the Atlantic. Although the Cousteau Organization is satisfied with the performance of the Aspirated Ellipse, it is too soon to say whether the device has a serious future.

- Propellers: Patents have been in effect for Magnus effect propellers since 1912. More recent work by John Borg of Atascadero, CA, indicates that a Magnus effect propeller in conjunction with a Kort nozzle, as shown in Figure 3, could increase propulsive efficiency and have lower manufacturing cost than conventional screw propellers.[5] Rotor spin drive can be accomplished by using a rack in the shroud (nozzle) to take power from the shaft or through a concentric shaft by a separate drive. Thrust reversal could be fairly easily accomplished with either drive approach. The efficiency advantage of the Magnus propeller over other types is shown in Figure 4.

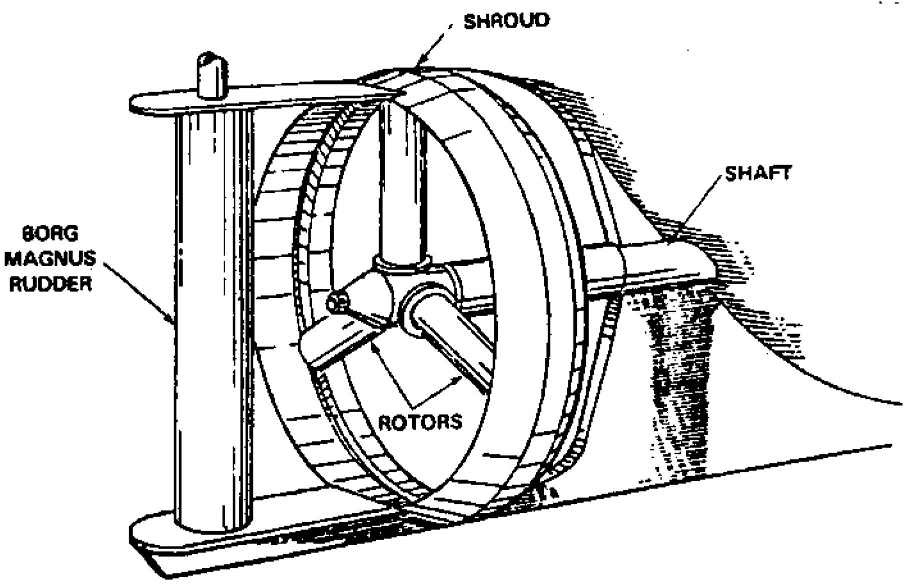


Figure 3 - Bŏrg Rudder and Propeller[3]

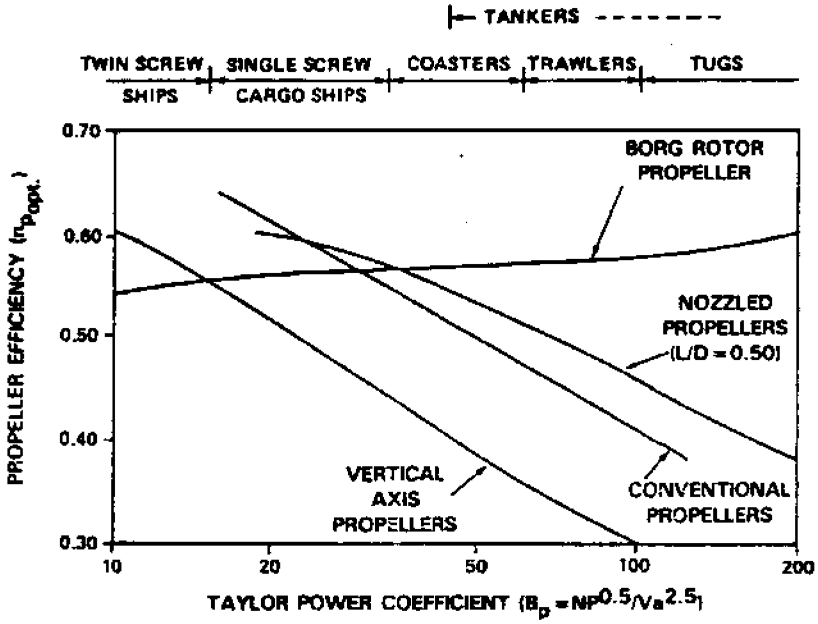


Figure 4 - Propeller Efficiency Curves[3]

Rudders: A Magnus effect rudder device was patented in 1929 by W. No evidence is available that would indicate that a Roos patent rudder was ever built. However, Borg has equipped a tow boat with a Magnus effect rudder similar to the Roos patent design (See Figure 5). The device was very successful in that steering performance was significantly improved and resistance in hard turns was significantly reduced. A Magnus effect rudder requires more power than a conventional rudder.[6] However, for vessels that frequently use large rudder angles, such as tugs, towboats and fishing vessels, the low resistance of the rotor, combined with its ability to apply thrust at the right angles to the hull centerline, makes it a most desirable system.[7]

Designing a Rotor for a Shrimp Boat

To design a rotor wind propulsion system for a shrimp boat. Let us first look at the rules.

Rules of Thumb for Rotors:

(1) The height of the rotor above the waterline is limited by the lowest fixed bridge expected to be encountered (Unless the rotor is a telescoping design).

(2) To gain maximum lift, the aspect ratio (A/R) of a rotor should be in the range of 6 to 12. Above 12, gains in lift are small and below 6 lift drops off rapidly. (Aspect ratio for a rotor = height/diameter.) See Figure 5, Curves A,B,C,E, F.

(3) Surface velocity of a rotor is a major factor in establishing lift. For rotors with A/R values above 6, a surface velocity of five times expected average wind velocity is recommended. However, for an A/R of say 2, increasing surface velocity ratio above 2 will not provide significantly increased lift (See Figure 5).

(4) Surface roughness of a rotor has some impact on lift. At least one experiment has indicated that rough cylinders provide at least 30% more lift than smooth cylinders at low Reynolds numbers (on the order of $4x10^4$). Use of a lightly sanded surface is recommended until more data, especially at higher Reynolds numbers, is available (See Figure 5, curves E & E^1).

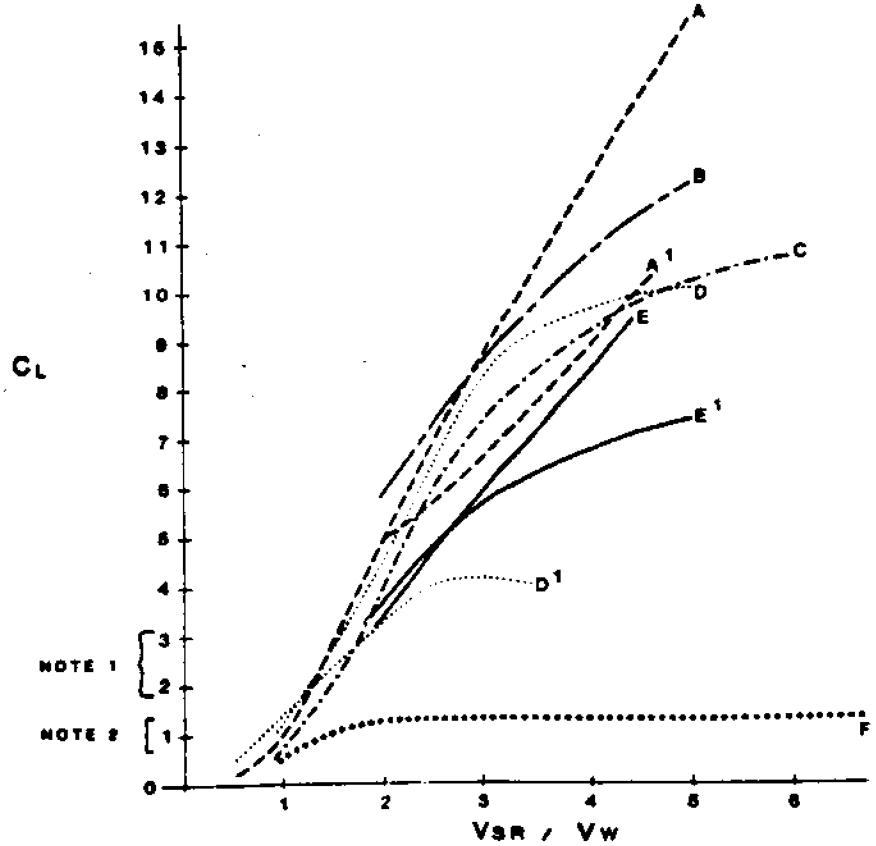
(5) End caps improve rotor lift especially for low A/R rotors. End caps or plates with diameters of 1.5 to 2.0 times rotor diameter are recommended (See Figure 5, curves A, A^1, D & D^1).

(6) If more than one rotor is used, rotors should be spaced a minimum of six diameters apart.

(7) Because upwind performance is best at a velocity ratio of 2, rotor drive should be variable speed. (See Figure 6).

	A/R	END PLATES	ROTOR FINISH	REYNOLDS #
A	12.5 & 26	3.0 X	UNKNOWN	5.3 - 6.8 X 10^3
A^1	13.3	NONE	UNKNOWN	3.3 - 11.6 X 10^4
B	6.2	1.58 X	SMOOTH	? (FULL SCALE)
*C	4.0	2 X	SMOOTH	?
D	4.7	1.7 X	UNKNOWN	6.2 X 10^4
D^1	4.7	NONE	UNKNOWN	6.2 X 10^4
E	6.7	NONE	SANDED	3 - 8 X 10^4
E^1	6.7	NONE	SMOOTH	3 - 8 X 10^4
F	2.0	WALL	UNKNOWN	5 X 10^4

* TESTED IN WATER - OTHERS TESTED IN AIR



NOTES:
1. RANGE OF LIFT FOR RIGID AIRFOILS
2. RANGE OF LIFT FOR SOFT SAILS
3. CURVES A, A^1, E, E^1, AND F ARE FROM SWANSON[8], B IS FROM WINDSHIP[9], C IS FROM BORG[2] AND D AND D^1 ARE FROM FLETTNER[10].

Figure 5. - Lift vs Speed Ratio Curves[2]

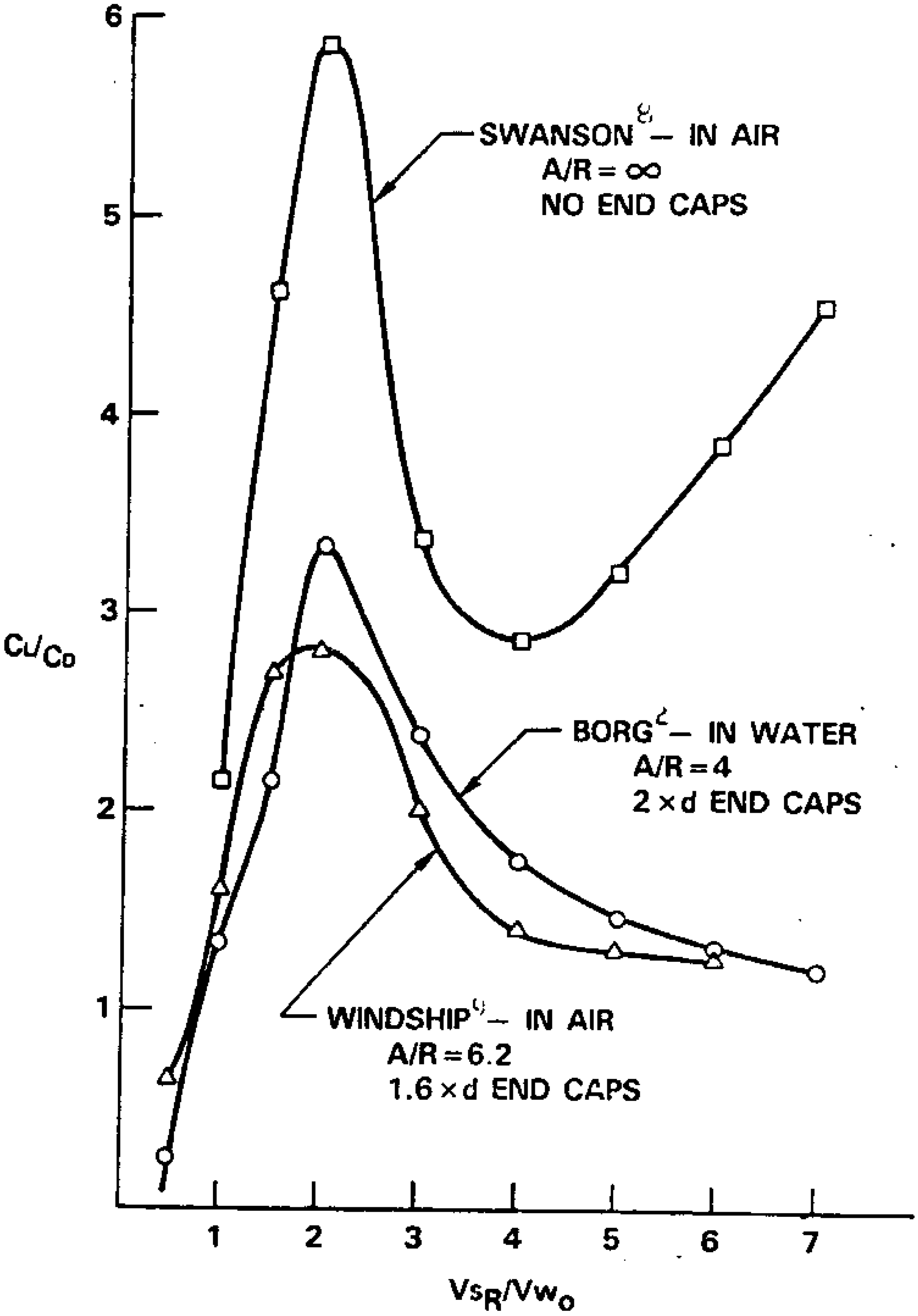

SWANSON[8] – IN AIR
A/R = ∞
NO END CAPS
BORG[2] – IN WATER
A/R = 4
2 × d END CAPS
WINDSHIP[9] – IN AIR
A/R = 6.2
1.6 × d END CAPS
C_L/C_D
Vs_R/Vw_o
1
2
3
4
5
6
1
2
3
4
5
6
7

Figure 6

Rotor Lift Requirments

Rotor lift requirements can be established in one of two ways:

(1) Replace total power capability - In a case such as replacing the sail system on a yacht with a rotor of equal lift, the area of the existing system multiplied by the ratio of sail system lift to rotor lift (A sail x C_L sail/C_L rotor) will provide the requisite projected rotor area. For example, if a catamaran, with 210 sq. ft. of high efficiency fully battened sail with a lift coefficient (C_L) of 1.5. were to be replaced with a rotor with an aspect ratio (A/R) of 8, end caps of 2 times rotor diameter, and an estimated C_L of 13 at a surface velocity to apparent wind velocity ratio of 5, the resulting projected rotor area would be 26.25 sq. ft (210x1.5/13).

(2) Augment existing propulsion plant - In the case of a large ship, the maximum practical lift is usually the goal. Optimization of cost is required which makes the problem more difficult.

Rotor Sizing & Arrangements

Data Required:

(1) Allowable rotor height with and or without telescoping (Bridge clearance - weather deck height above the DWL).

(2) Selection of the number of rotors and the rotor arrangement - Typically, small vessels (100 ft long or less), need only one rotor. However, large vessels usually need 2 or more rotors to gain a reasonably significant effect.

Example

Ship selected: Florida Shrimper

Reasons: High fuel consumption per pound of catch and relatively large open weather deck area forward.

Power requirement: 75 hp to permit 8 knot speed under optimum rotor performance and 12 knots of apparent wind (12 knots is the average wind in the Florida area.

One rotor has been selected as one can provided enough power.

A rotor with a height of 42 feet and a diameter of 6 feet for an A/R of 7 to comply with Rule (2) is planned.

The rotor is planned to be telescoping to permit passing under bridges with 40 foot clearance to comply with Rule (1).

Rotor end caps, 10 feet (6x1.67) in diameter have been provided to comply with Rule (5).

A variable speed drive has been provided to comply with Rule (7).

CALCULATIONS

Lift Calculations

Given:

Rotor height (L): 42 feet

Rotor diameter (d): 6 feet

Optimum apparent wind speed (^{V}wo): 12 kts or 20.0 ft/sec

Velocity of rotor surface ($^{V}s_R$): 20.0 x 5 = 100 ft/sec (Complies with Rule (3).)

Projected area (Ap): L x d = 42 x 6 = 252 sq ft

Mass density in slugs/cu ft = 0.0023 (air) (or 1.9875 in salt water) ( ).

Then:

Wind Speed (Vw) Knots	ft/sec	Ratio (Vsr/Vw)	C_L[#]	Lift (L)(Pounds) ($C_L A_p \rho Vw^2/2$)	HP Lift x.05
10	16.9	5.9	(14.0)	(1149)	57
12	20.0(Vwo)	5.0	13.0	1,500	75[@]
15	25.3	4.0	11.5	2,116	106
20	33.8	3.0	9.0	2,956	148
25	42.2	2.4	6.8	3,481	174
30	50.7	2.0	5.0	3,695	185
40	67.6	1.5	3.0	3,941	197
Collapse Rotor					
50	84.5	1.2	1.3	2,818	141
100	168.9	0.6	0.3	1,299	65

#Numbers in brackets are extrapolations of existing test data in Figure 5 and may not be accurate. Other numbers are interpolated from Figure 5.

@ 75 HP will drive a medium sized shrimper at eight knots.[11]

Input Horsepower (HPin) Calculations

Given:

Length along rotation axis in feet (L) - 42

Diameter of rotor in feet (d) - 6

Diameter of endplates in feet (d_E) - 10

Optimum apparent wind speed in feet/sec (V_{Wo}) - 20.0

Optimum surface velocity to V_{Wo} ratio - 5

Friction factor (f) - 0.0000121 (air)(or 0.01 (water))

Then:

Projected area (Ap) = Ld = 42 x 6 = 252 square ft

Rotor surface velocity (Vs_R) = V_{Wo} x 5 = 20.0 x 5 = 100 ft/sec

Rotor surface area, (As_R) = Ap π = 252π

Rotor RPS = $Vs_R / \pi d$ - 100/6π = 5.31

Rotor RPM = 60 RPS = 318

Surface area of rotor end (As_{EPR}) = $(d/2)^2 \pi$ = 9π square ft

Surface area of rotor end cap outside surface (As_{EPO}) = $(d_E/2)^2 \pi$
= 25π square ft

Surface velocity of rotor end plate (Vs_{EPR}) = RPS x π x $^2/3$d
= 66.7 ft/sec

Surface velocity of rotor end cap outside surface (Vs_{ECO})
= RPS x π x 2/3d_E = 111 ft/sec

Rotor Torque (T) = $f[(As_R \times d/2 \times Vs_R{}^{1.825}) +$
$2\ (As_{EPO} \times d_E/3 \times Vs_{ECO}{}^{1.825}) + [(As_{EPO} \times d_E/3 \times Vs_{ECO}{}^{1.825})$
$- (As_{EPR} \times d/3 \times Vs_{EPR}{}^{1.825})]]$

$T = 0.0000121\ [(252\pi \times 6/2 \times 100^{1.825}) + (25\pi \times 10/3 \times 111^{1.825})$
$+ 2\ [25\pi \times 10/3 \times 111^{1.825}) - (9\pi \times 6/3 \quad \times 66.7^{1.825})]]$

T = 177 ft pounds

HPin = T x RPM/5252 = 177 x 381/5252 = 10.7*

*Authors Note: The input horsepower is very conservative. Actual input horsepower may be as little as 50% of the calculated value.

The rotor is shown installed on a typical shrimp boat in figure 7, and is designed to be constructed from readily available materials. The rotor itself should be made of fiberglass, the mast of standard aluminum tubing and pipe and the hydraulics system should use stock hydraulic components available from a number of hydraulics manufacturers. Based on costs the author is currently incurring for a smaller rotor, a cost figure of $100 per square foot of rotor projected area, half for materials and half for labor, is considered reasonable. Therefore, the cost of the rotor system shown in figure 10., should be about $25,000 (Ap x Cost/sq ft = 252 x 100 = $25,200).

In 1970, the fuel bill for a typical shrimper was on the order of $10,000 to $12,000 a year, however by 1980, fuel cost rose to between $80,000 to $100,000 a year. The expected fuel savings with the proposed rotor are on the order of 25%. Therefore the cost of rotor system can be written off in about one year and when the cost of system installation (which will vary from vessel to vessel) is added, a writeoff period of one to two years is anticipated.

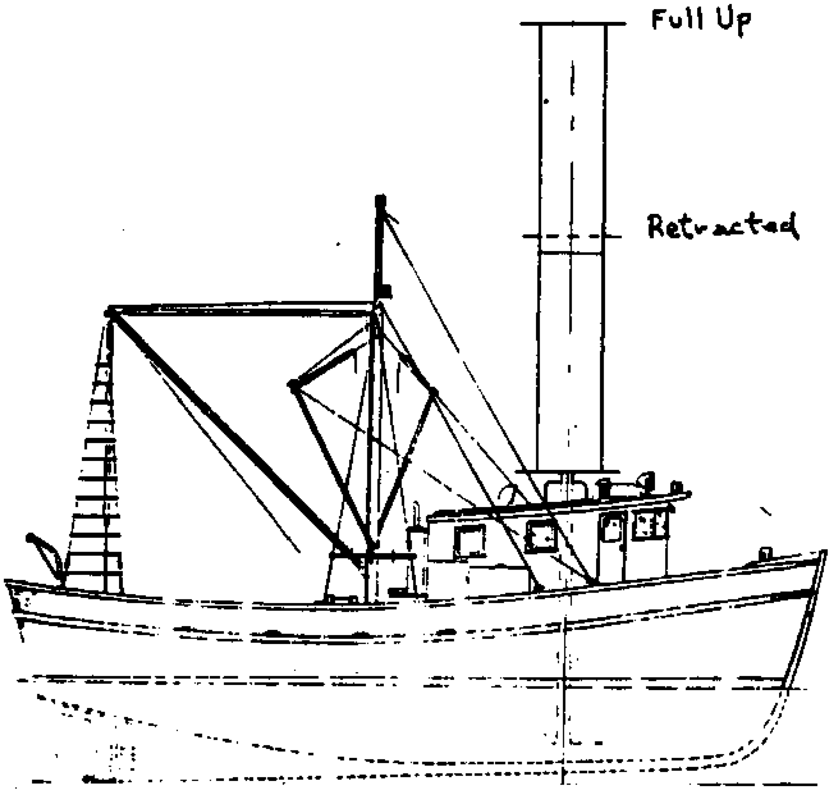


Figure 7. - Magnus Effect Rotor for a Shrimp Boat[13]

Conclusions and Recommendations

The Magnus effect is a significant phenomenon which has been overlooked to a great extent by the marine community. With fossil fuel energy in short supply and at a high cost, we should be considering those changes that either eliminate or reduce fuel requirements. The Magnus effect is clearly a phenomenon that should be used. Therefore, the fishing fleet owners and operators should seriously consider the following:

- Fitting fish boats with Magnus effect rotors,
- Fitting fish boats with Magnus effect rudders,
- Fitting fish boats with Magnus effect propellers

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FURTHER DEVELOPMENT OF THE TUNNY RIG

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1. INTRODUCTION

The idea of using a wing sail is not new, indeed the ancient junk rig is essentially a flat plate wing sail. The two essential characteristics are that the sail is stiffened so that it does not flap in the wind and attached to the mast in an aerodynamically balanced way.

These two features give several important advantages over so called 'soft sails' and have resulted in the junk rig being very successful on traditional craft and modern short handed cruising yachts.

Unfortunately the standard junk rig is not every efficient in an aerodynamic sense, due to the presence of the mast beside the sail and the flat shape which results from the numerous stiffening battens.

The first of these problems can be overcome by using a double skinned sail; effectively two junk sails, one on either side of the mast. This shields the mast from the airflow and improves efficiency, but it still leaves the problem of a flat sail.

To obtain the maximum drive from a sail it must be curved (or cambered), an effect which can produce over 50% more force than from a flat shape. Whilst the performance advantages of a cambered shape are obvious, the practical way of achieving it are far more elusive.

One line of approach is to build the sail from rigid components with articulated joints that allow the camber to be varied (Ref 1). This approach is claimed to give a very good performance, but suffers from the practical disadvantage that the sail area cannot be reduced by reefing.

(On large vessels, where the rig provides sail assistance contributing only a fraction of total power, this may not be a limitation, but for small vessels designed to exploit sail power to the full, some form of area reduction appears to be vital).

The alternative of a sail which has both camber control and provision for reefing has proved to be a very difficult specification to meet with a practical seagoing solution.

Many extremely complicated systems have been proposed, which whilst they may have worked for their enthusiastic inventors, have not been sufficiently practical to attract wider attention.

The first practical solution to this problem has come from an unexpected quarter. The Combewrights had a simple objective, to build the cheapest boat that would take them safely across the Atlantic. They started with no preconceptions and little knowledge of boats or the sea. They talked to people, listened and learnt. Their solution was a 32' catamaran with a wing sail - a wing sail with camber control which they called the Tunny Rig. They realised their ambition, crossing the Atlantic last year without problem. Since then they have proceeded via the West Indies to the Panama canal and so to the West Coast of America.

Their voyage has proved the practicality and seaworthiness of the rig in its prototype form, and their basic concepts are embodied in new versions of the rig being developed by Gifford Technology. This development has taken place over the past 18 months and has now reached a stage where a production prototype is being tested and commercial applications are under serious consideration.

2. PRINCIPLES OF THE TUNNY RIG

The Tunny Rig is an aerofoil wing sail made from a set of shaped battens connected by panels of sail cloth (Fig 1). The battens wrap around the mast, thus forming the aerofoil section, and can be collapsed one on top of the other either for reefing or clearing away the sail.

For almost two thirds of their length the battens are rigid frames, made from thin section timber. The final third of the length comprises detachable, flexible extensions to the main frame, interconnected only by a single tensioned line.

The camber control comes from two lines, one from the end of each flexible section, which are fastened to a cross member in the front part of the batten. By differentially tensioning these lines the batten is warped one way or another, the degree of warp being limited by the tension line joining the flexible extensions.

In the full rig these control lines are tensioned by a common line running down through the sail to over centre levers on the lower batten (or boom).

3. TUNNY RIG DEVELOPMENT

Building on the Combewrights' original concept the rig has been developed in two phases, comprising the manufacture of a 28m^2 sail for initial practical appraisal followed by an 18.2m^2 sail for direct comparative evaluation against other rigs.

Much of the first stage was reported in Ref 2, notably the selection of the aerofoil section for the sail and the simplification of the internal mechanism.

The section chosen was the NASA GA(W)1, a recent product of research into aerofoils suitable for light aircraft. It proved ideally suited to the Tunny Rig for several reasons:

Relative insensitivity to surface roughness

Thick section with thickness carried well forward

Camber largely concentrated in small proportion of the trailing edge
Good lift to drag ratio coupled with high maximum lift coefficient

Good aerodynamic properties retained in low wind speeds, i.e. low Reynolds numbers

Figure 1 (from Ref3) shows the basic lift characteristics of the section at typical sailing boat Reynolds numbers. Figure 2 shows its superiority in terms of maximum lift over other typical sections, in the practical condition of aerodynamically rough surfaces.

The first development rig of 28m^2 area was built using this section and fitted to a catamaran trials boat (Fig 3). Extensive trials were carried out last summer in a wide range of wind strengths and a great deal was learnt about the general handling characteristics and practicality of the rig.

The positive features demonstrated are:

Low sheet loads. A direct result of the sail area being aerodynamically balanced on the mast as in the junk rig.

Evenly distributed loading. Since the sail does not rely for its shape on being set between attachment points where loads are highly concentrated, the sheet and aerodynamic loads are evenly distributed so reducing the weight and quality of sail cloth required as well as the complexity of manufacture.

Insensitivity to sheet position. Because the sail is self supporting it does not rely upon the direction of the sheet lead for its shape. Consequently the sheet can be taken to points which do not obstruct operation of the boat and which fit best with the overall deck layout, rather than dictating the layout as with many soft sail rigs.

Absence of flapping. The sail cloth is supported by the Tunny battens so does not flog when being raised or as the sail is tacked.

Practicality of Reefing. Again in common with the junk rig reefing is a quick and effective process.

Ease of handling. The light sheet loads mean that the $28m^2$ sail can be tended single handed without the need for a winch. Its docile behaviour when hoisting or lowering also makes life much easier for small crews.

Power tacking. An unexpected feature was that because the sail retained drive even when very close to the wind, tacking was a much more reliable process than is normally the case on a catamaran. Provided the sheet was freed off (contrary to normal practice) as the boat was put into the wind the sail would drive the boat round and through the wind.

Close winded. The lower drag of the rig coupled with its stable shape allows it to be carried very close to the wind.

Camber effective. Trials conducted with and without camber in the sail demonstrated an improvement in drive resulting from the application of camber.

Stern power. By pushing the sail out into a reverse mode the craft can be sailed astern quite simply.

Life is not all gain however and there are of course some negative points about the rig. Compared to, say, a Bermudian rig, the Tunny Rig will always be heavier, though further development and the incorporation of more refined structural details could well reduce the margin considerably.

The other problems which were found on the first rig were largely specific to that particular rig and were rectified for the next $18.6m^2$ version.

Excessive structural weight. In the absence of reliable loading data the Tunnies and the sail cloth were made greatly over strength and thus the rig was much heavier than necessary.

NASA GA(W)-1 Section

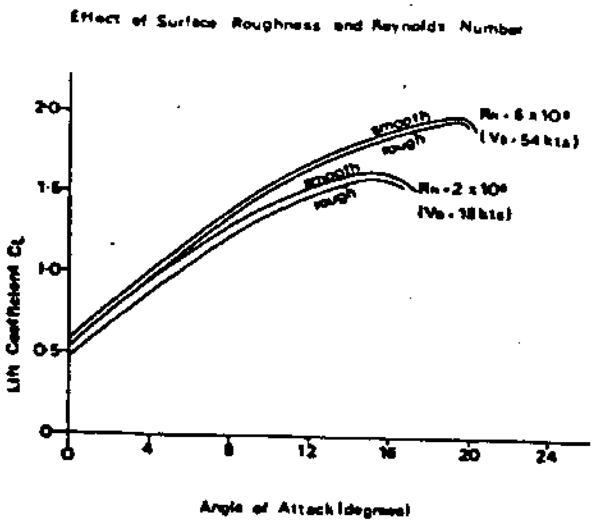

Effect of Surface Roughness and Reynolds Number

Figure 1

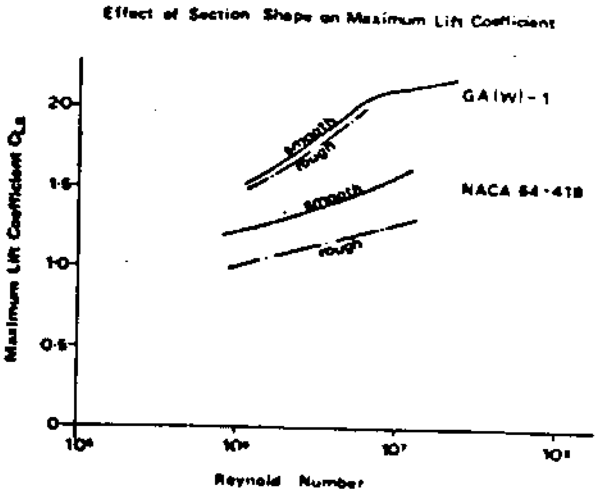
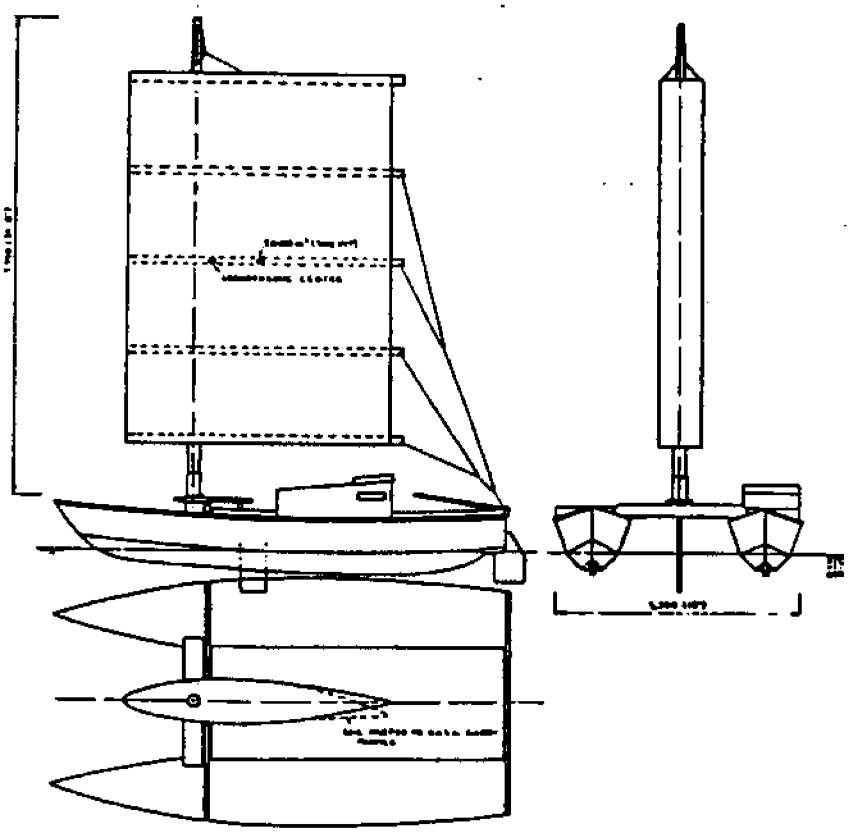

Effect of Section Shape on Maximum Lift Coefficient

Figure 2



TUNNY RIG ON SANDSKIPPER

Figure 3

Creasing of sail cloth. Due to deflection in the top batten (gaff) the top panel set badly towards the trailing edge. Also it proved impossible to completely eliminate creases around the leading edge, even when extreme halliard tensions were applied.

Over complex lacing. The sail was laced to the Tunnies far more extensively than was necessary, making removal and adjustment very time consuming.

High camber control forces. This problem was linked to the overstrong structure which resulted in the Tunnies being stiff and difficult to deflect.

Damage to control mechanism. The camber lines were tensioned by means of levers on each Tunny. These worked well but suffered damage when the sail was lowered.

Limited rig rotation. Due to the halliard arrangement and the down haul on the boom the rig could not swing out freely to large angles. Since it was finely balanced about the mast the aerodynamic forces were insufficient to force the sail out and thus it was difficult to reduce the drive by simply freeing the sheet.

Lack of visual clues to sail setting. Since the sail did not flap when not set correctly it was difficult to determine visually the best sheeting angle.

No reliable measure of performance. Whilst the sail appeared subjectively to perform well it was not possible to obtain any valid comparison with other rigs by which its true performance could be judged.

4. THE 18.5m^2 RIG

Choice of Area

As part of a contract with the Commissioners of the European Communities (CEC) Gifford Technology had carried out a series of comparative sailing trials using four different rigs, each of 18.5m^2 area (Ref4). Accordingly it was decided to build a Tunny Rig of this same area which could be evaluated in the same way, thus giving a measure of its performance relative to other well known rigs.

Aspect Ratio

Having fixed the area it was also decided to increase the aspect ratio by a moderate amount to a value of 2.0. This is still far from the most efficient shape for the very best performance to windward, but a reasonable compromise for all round performance when heeling moment and structural consideration were taken into account.

Planform

Theory indicates that the elliptical planform is the best from the point of view of minimising the induced drag of the rig. Practically it is far from ideal since each Tunny must be a different length, thus greatly complicating manufacture. Similar considerations apply to a triangular shape, which is also a bad shape aerodynamically.

These lines of reasoning led to a rectangular planform, which particularly when associated with a squarely cut off tip, gives virtually the same performance as the elliptical shape. The final refinement was the cutting back of the top batten to give a small amount of taper and to eliminate the creasing in the top panel.

Using information in Refs 5 and 6 predictions were made of the induced drag for different planforms and it was found that the penalty associated with adopting the practical rectangular planform was negligible, as compared to the "ideal" ellipse.

Detailing

In addition to these basic design considerations a number of detail changes were made, mainly as a result of the negative features found on the 28m^2 rig.

Lightweight Structure. The size of the timber used for the Tunnies was reduced from 90mm x 18mm to 40mm x 12mm. For reasons of economy solid redwood timber was used and further weight savings could be made by using laminated timber construction.

Lightweight sail cloth. Cloth weight was reduced to 4½ oz.

Lacing arrangement (Figure 4). The number of lacings on the intermediary Tunnies was greatly reduced and a revised system adopted for the top and bottom which makes adjustment of the sail much easier.

Detachable battens. The flexible sections of the Tunnies were made detachable so that they could be readily replaced if damaged. They were sleeved into the sail in the same manner as conventional sail battens, so further reducing the need for lacing.

Harp String Warping (Fig 5). Instead of using levers on each Tunny for tensioning the warping lines a harp string approach was adopted. This does not give quite such a good mechanical advantage, but offers significant gains in simplicity and a reduction in weight.

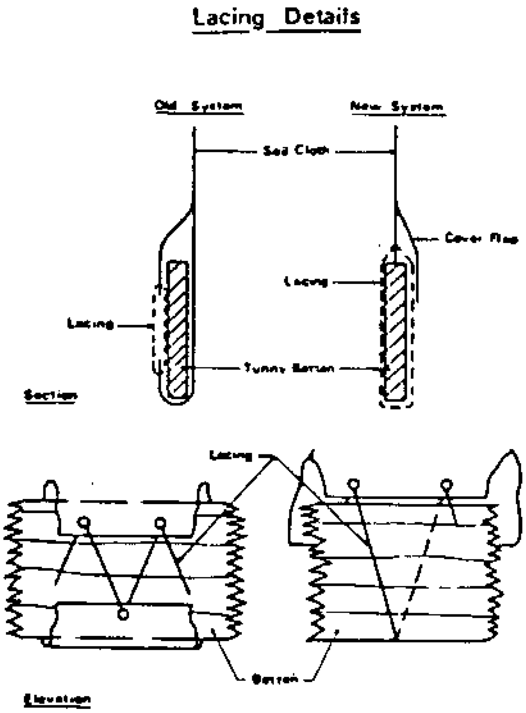
Floating Ribs. To give a smooth shape around the leading edge intermediary 'floating ribs' were fitted to the sail.

Mast Crane and Collar (Figure 6). To ensure that the sail was free to rotate the halliard was lead from the mast head via a rotating crane and the boom restrained by small wheels bearing against a collar on the boom.

5. THE TWIN TUNNY RIG

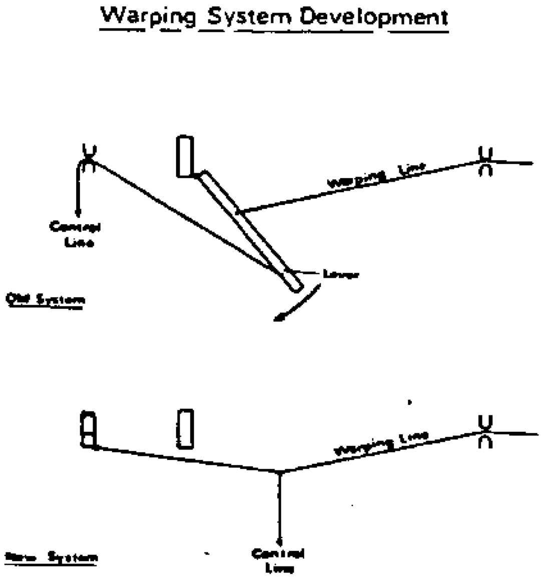
In parallel with the development of the Tunny rig a 28' double hulled fishing boat was being developed for a beach fishing project in Sri Lanka (Ref 7).

For a good sailing performance in tropical conditions a sail area of 37m^2 was considered necessary and a design exercise was undertaken to examine the possible rig alternatives.



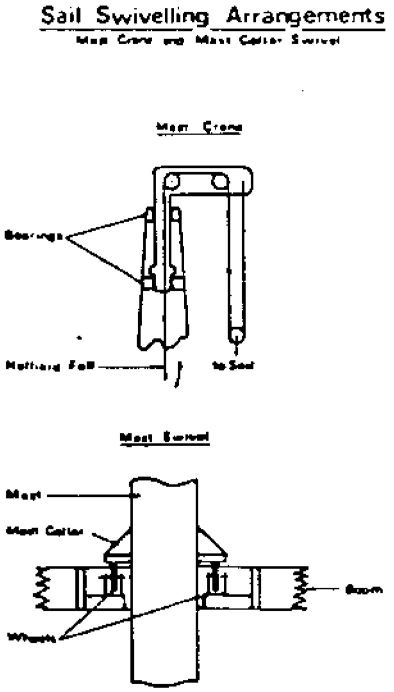
Lacing Details

Figure 4



Warping System Development

Figure 5



Sail Swivelling Arrangements

Mast Crane and Mast Collar Swivel

Figure 6

SANDSKIPPER 28

LOW ENERGY FISHING BOAT
WITH TWIN TUNNY RIG

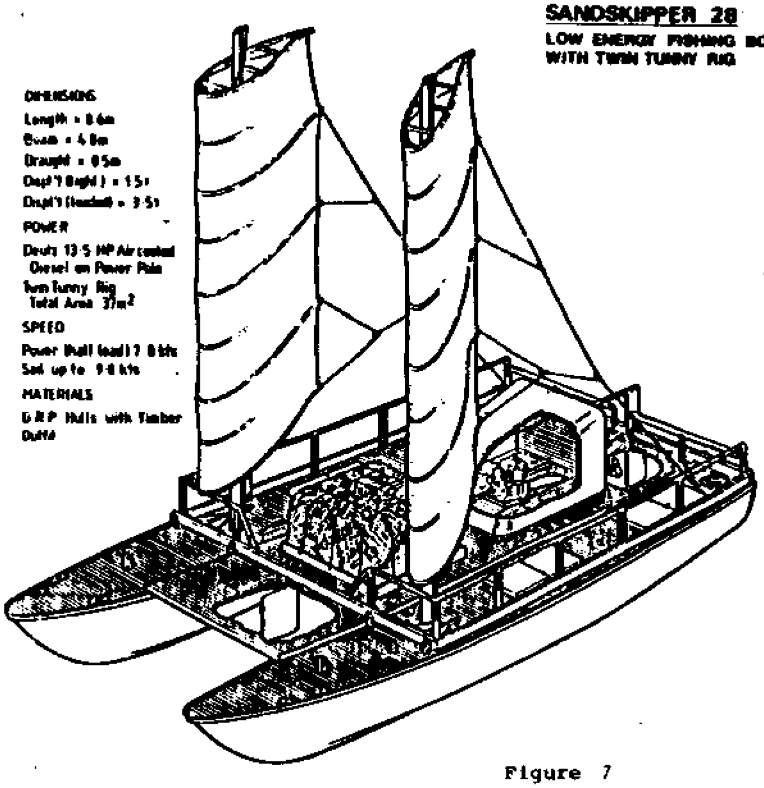
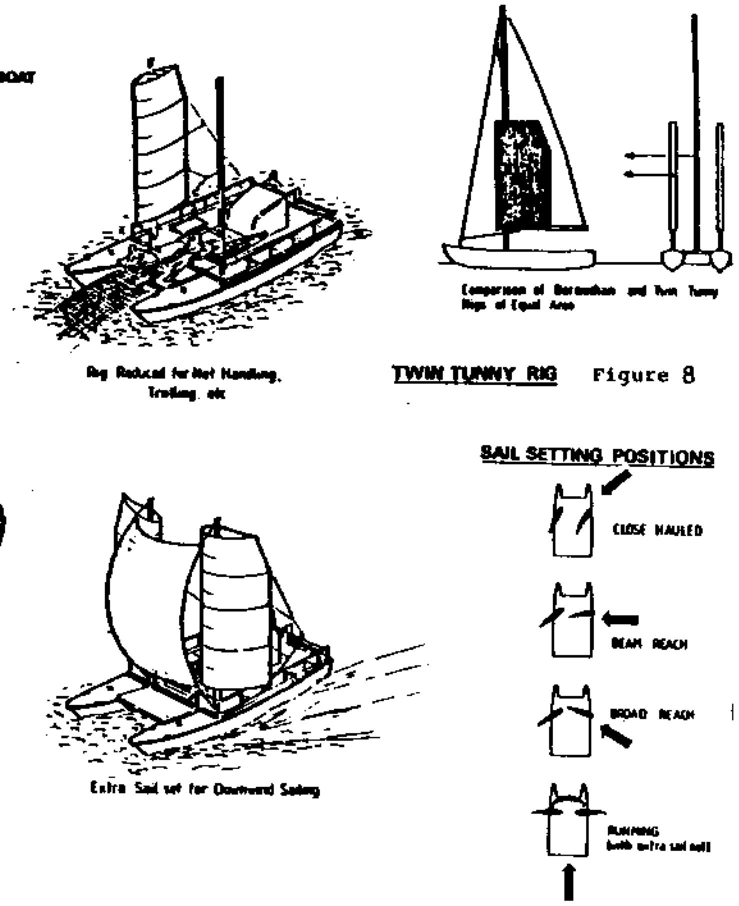


Figure 7


Rig Reduced for Net Handling, Trawling, etc

Comparison of Bermudian and Twin Tunny Rigs of Equal Area

Extra Sail set for Downwind Sailing

TWIN TUNNY RIG Figure 8

The traditional rig that could be used was the lateen, which though a practical and culturally acceptable option, was not expected to give a very good performance. Also the size of the spar required would become difficult to stow on the boat and vulnerable in surf. Similarily there were likely to be problems with spar size if other rigs such as the sprit or bermudian were carried on a single mast. Within the length of the boat the idea of splitting the rig to give a ketch or schooner layout proved impractical - the rigs would be so close together as to interfere a great deal and much of the double hulled advantage of deck area would be lost. Finally a centre line position for a mast introduced additional structural loads on the cross beams and practically precluded the possibility of using unstayed masts.

The solution adopted was to place the masts side by side, one in each hull. It was immediately apparent that this gave many worthwhile structural and layout advantages and potentially good aerodynamic performance.

Figure 10 shows that 8.5m boat fitted with two 18.5m^2 Tunny rigs (though the layout is not rig specific and could be made to work well with other single rigs suitable for unstayed masts).

The advantage of this layout over a single central rig are considerable:

Lower centre of effort height.

Less weight aloft giving lower centre of gravity and lower pitch inertia.

No obstruction to bridge deck for handling nets.

Absence of rigging and the associated point loadings.

Cross beams do not have to be reinforced to take mast compression.

In Figure 8 the two rigs are overlaid and the differences are visually very apparent. The other appealing features of this layout when compared to a fore and after disposed twin rig are:

Masts can be buried into the hulls and thus unstayed if required.

Close hauled performance is likely to be much better (though not quite as good for a single rig).

Reaching performance is potentially very good due to the slot effect that can be created by working the two sails together.

Down wind additional sail area can be set between the two masts.

The boat remains balanced with only one rig set, so reefing can be readily achieved by dropping one complete sail. Also the single sail rig is useful where only low thrust and fine control are required.

Most of these points are illustrated by the various insets in Figure 7 and 8.

Performance Predictions

Using a computer program specially developed for evaluating the effects of different rig and hull characteristics a series of polar performance predictions were made. Figure 9a shows three of the results obtained. Figure 9a illustrates the correlation between the theory and experimental results obtained for the bermudian rig tested on a Sandhopper trials craft during the CEC trials. It can be seen that the prediction is within the scatter of the full scale results and can thus be considered an adquately reliable tool, particularly where emphasis is placed upon relative performance comparisons.

For Fig 9a the hull data was obtained directly from trial results and rig data from wind tunnel tests at Southampton University (Ref 8).

For the predictions of the SK28 performance no direct test data was available for the hull. However, tank test data for the 20' craft of very similar proportions was available, as was full scale thrust data for the SK24, a craft even closer to the proposed SK28. Figure 10 shows these two results plotted non-dimensionally and Figure 11 is the prediction made from them for the SK28. Finally a check was made using the excellent prediction method of Gerritsma (Ref9). Also on Figure 11 is the thrust curve measured during the CEC trials for the 13.5 HP Deutz diesel, the engine proposed for the SK28.

Rig data could not be established quite so reliably and various assumptions had to be made so that conventional aerodynamic theory could be applied.

The first stage was to construct a polar curve for an AR=2 Tunny rig. For operation up to the point of stall standard aerodynamic theory was used to correct the wind tunnel data of Ref 3 to finite aspect ratio and the planform and tip shape of the proposed rig. Then using other rig data and information from Ref 6 a drag coefficient of 1.2 was deduced for the foil set normal to the flow and a smooth curve fitted between that point and the stall point. This approach may not give a very reliable estimate of the foil performance just beyond stall, but the prediction program indicated that the optimum angle for the rig seldom exceeded the stall point except on direct down wind courses. Thus the errors resulting from the inprecision are likely to be very small. The resulting rig polar is shown in Figure 12.

For the twin tunny rig a biplane correction factor was applied to the results up to the stall point (this was applied as a virtual reduction in aspect ratio and hence increase in induced drag). Beyond the stall point it was assumed that the sails were set to give beneficial mutual interaction which at large angles of attack could give a peak lift coefficient similar to that of a flapped aerofoil, i.e. around 2.4. In the down wind condition the drag coefficient was corrected to incorporate the effect of adding extra area and the vague intermediary part of the data faired in. This result is also shown on Figure 12.

Finally using similar methods a rig polar for a Tunny sail fitted with a notional flap was also constructed.

Figure 16 shows various rig polars from other sources with the twin Tunny prediction overlaid for comparison.

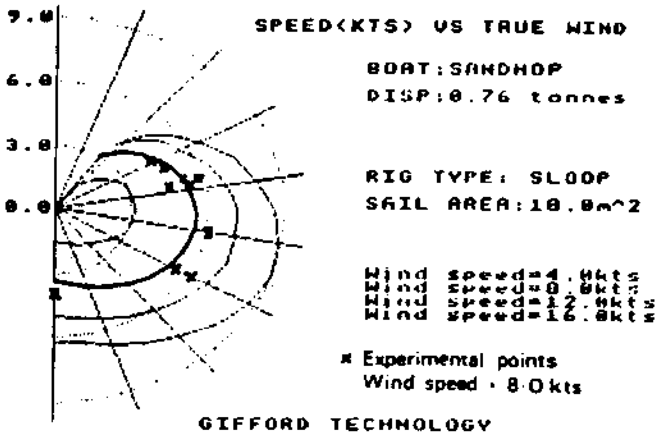


Figure 9 a

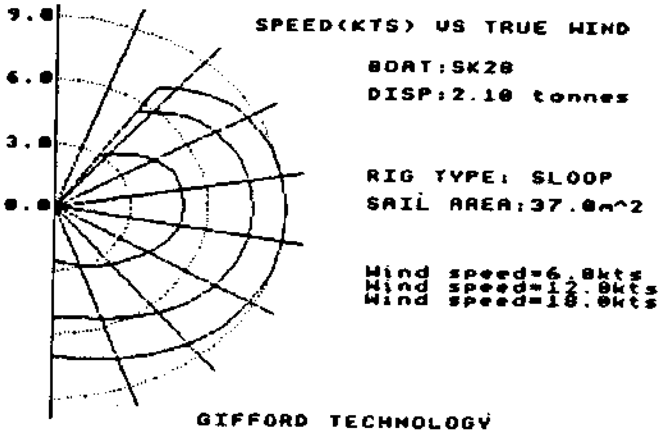


Figure 9 b

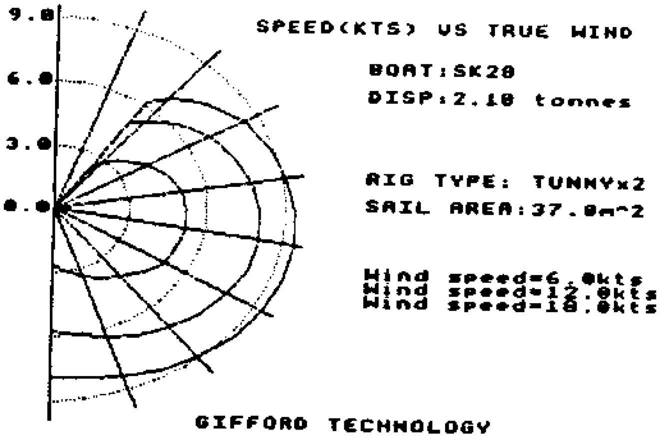


Figure 9 c

Figure 10

Comparison of Specific Resistance

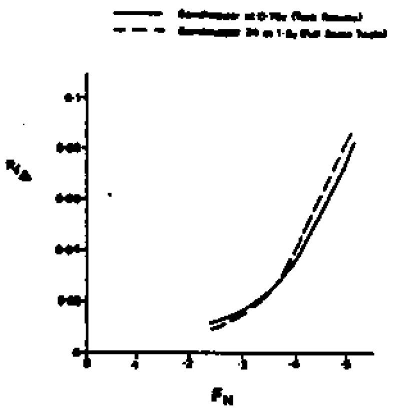


Figure 11

Sandskipper 28 Resistance Predictions

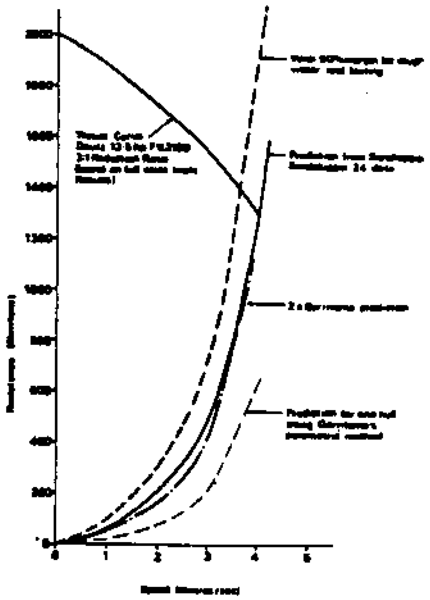


Figure 12

Polar Diagrams of Different Tunny Rig Configuration

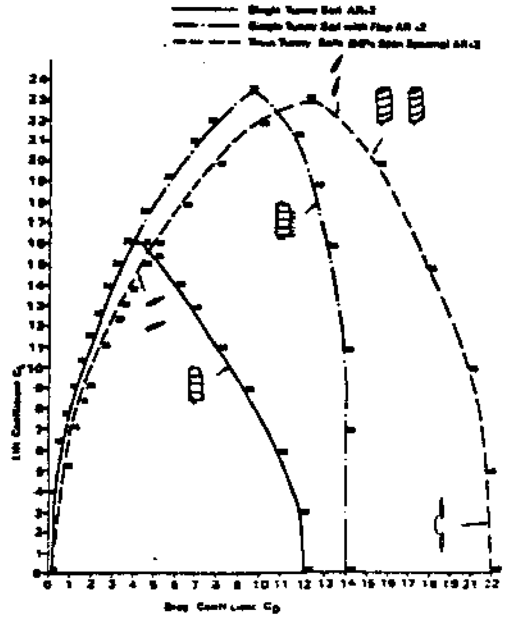
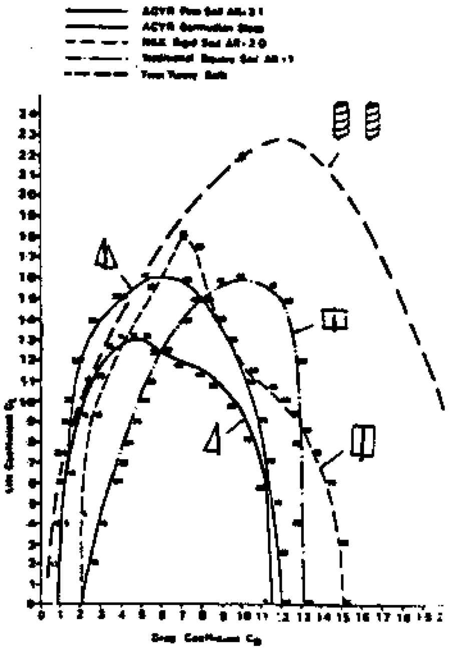


Figure 13



Using this data and Southampton University data for Bermudian sails (Refs 8 and 10) a range of performance predictions were made. Two of the polar curves are shown in Figure 9b and 9c and the results of the full set of runs summarised in Figures 14 and 15.

Figure 14 compares the performance of the four different rigs in 12 knots of wind (a typical tropical breeze). For each rig three different speeds are shown:

Speed made good to windward.

Maximum reaching speed (generally with true wind angles of 90° to 100°).

Down wind speed (true wind angle of 180°).

To windward the Bermudian sloop has a slight advantage over the others, showing a Vmg of 4.5 knots against 4.2 for the twin tunny and 4.3 for the single tunny.

When reaching the twin tunny and the tunny with flap gave the best performance and downwind the twin tunny is fastest of all, due to the additional area assumed to be set between the masts.

On all points of sailing the speed through the water exceeds 5 knots, and when reaching it is greater than 7.5 knots. This compares favourably with the maximum speed under power of 7.8 knots and suggests that lack of speed will not be a large disincentive to the use of sail.

The results in Figure 14 are for a wind strength where even for the tall single rigs an adequate stability margin remains.

With increasing wind speed the sail area must be progressively reduced - the limiting stability point of flying a hull occurring in 19 knots of wind with the single sail rigs and 21 knots with the twin tunny.

Figure 18 compares the performance of reefed rigs. First; in 18 knots of wind a single small Tunny (i.e. twin rig with one sail dropped) is compared to the sloop rig with a deep reefed mainsail. The step function change in tunny rig area results in a slightly inferior performance, but the difference is not more than 10%.

In stronger winds (25 knots) a comparison is made of the single Tunny with a mainsail only Bermudian rig. Under these conditions the Tunny rig is superior on all points, but again the margin is small.

The single Tunny can theoretically be carried in winds of 30 knots, but the option of reefing that sail remains if continued operation under sail is required in still stronger winds.

From these comparisons it can be seen that twin Tunny rig has a good performance potential, which coupled with its convenient layout and ease of handling makes it a very attractive rig for small commercial boats.

6. ECONOMIC ANALYSIS OF SAIL ASSISTANCE

Particularly at an early stage in a project it is very difficult to make valid economic forecasts of sail assistance projects. There are two distinct (but in some senses connected) sides to the equation; on one side the capital investment and running costs for the rig, on the other

Figure 14

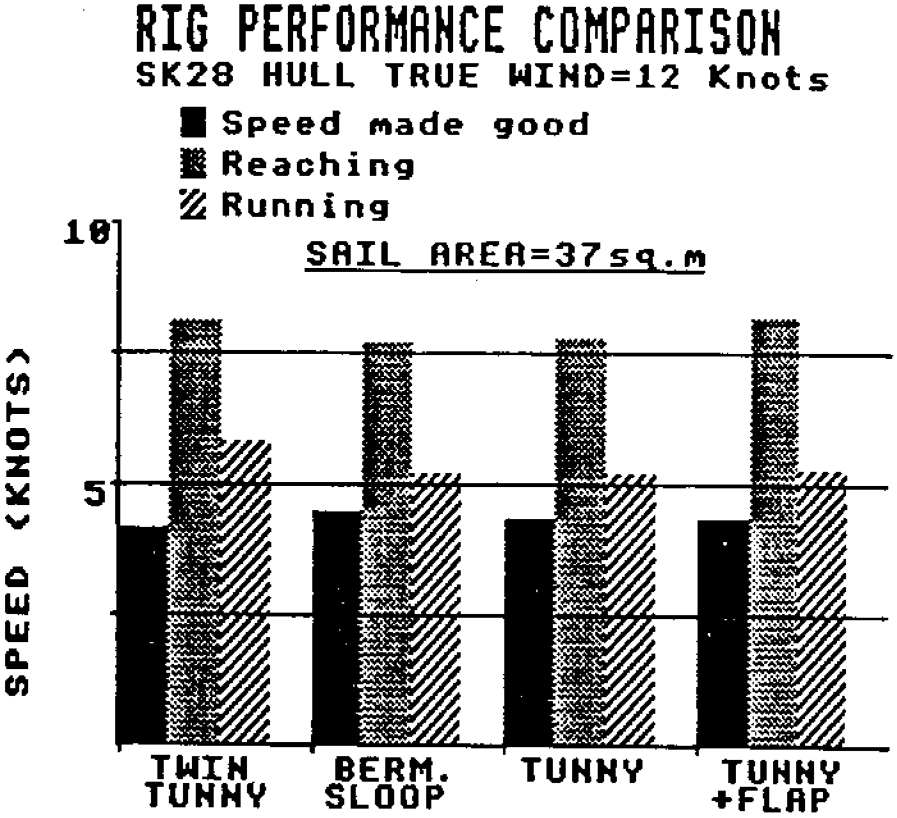
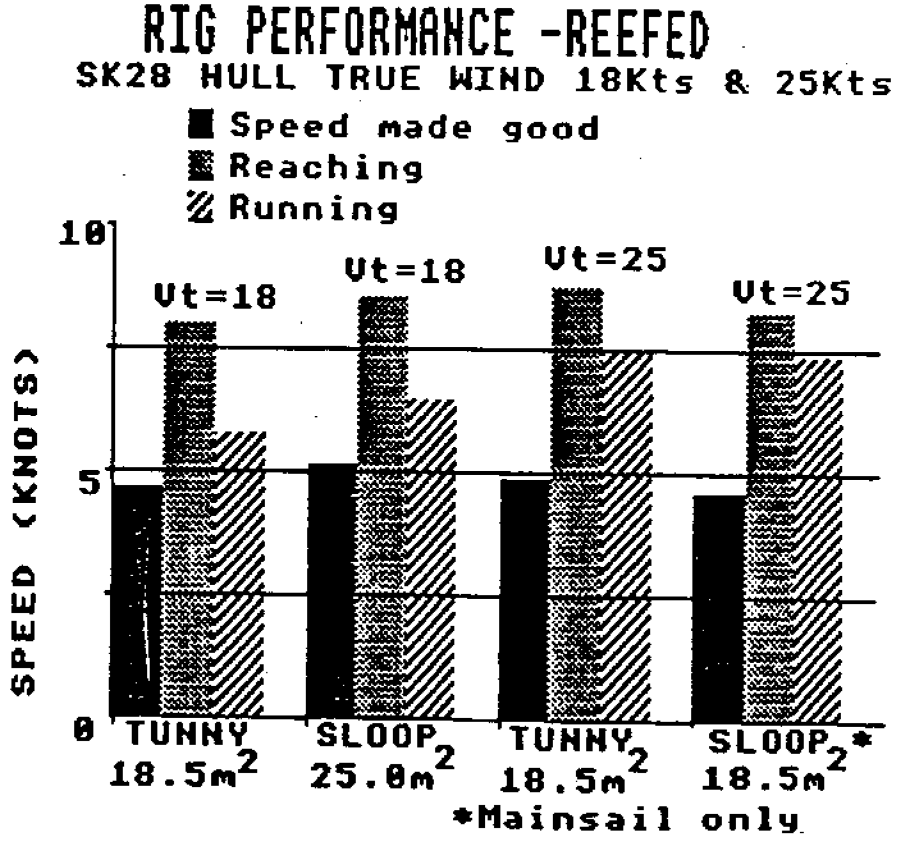


Figure 15



the fuel saving that will result from the use of the rig. The capital side is relatively straightforward, though maintenance costs will tend to be very dependent on actual operational conditions. The problems come when the other side is tackled. Difficult questions immediately arise:

How many hours per year will be rig be used

What is the minimum speed acceptable

Will motor and sail be used together

Will the use of sail influence the fishing strategy

Does sail allow more fishing trips to be made (i.e. say cases where fuel was previously unavailable)

The combination of all these unknowns must make very unreliable any attempt at a single answer for the amount of fuel saving that will result from the use of the rig.

Fortuntely if computers are available to do the donkey work it is a simple matter to examine a range of different cases and so get a feel for the sensitivity of the economics to the different variables. From this it is possible to make reasonable judgements and answer "what if" questions.

To demonstrate this approach the ITIS/Giffords Sandskipper project in Sri Lanka is taken as a basis. Two 7.3m double hull Sandskipper fishing boats are currently working in Sri Lanka, soon to be joined by a larger (8.5m) boat. Using data from the present operation a prediction of the economic performance of the new boat was made. The method used is a full discounted cash flow over a ten year period (a conservative life for the boat). Figure 16 shows the result for a datum condition. An internal rate of return (IRR) of 96% is indicated. For more details of this prediction and the assumption with it, see Ref 7. The techniques used for the analysis are well described in Ref 11.

To investigate the potential for sail assistance the analysis was rerun for three different conditions to give a range of rates of return. In each condition the capital cost of the rig was adjusted to maintain a constant IRR.

Figure 17 shows that at IRR of 108% an expenditure of roughly RS 500 is justified for each percentage point of fuel saving. Since the operational experience reported for other sail assist project indicates that fuel savings in the range of 10% to 30% are practical, the capital outlay on a rig can be Rs 8000 to Rs 20000. (Rs 2000 being already allowed in the datum condition). For projects where a lesser rate of return is acceptable greater capital expenditure can be justified.

No cost for the Tunny rig in Sri Lanka is currently available, but it is known that a lateen rig of similar area would cost only Rs 2000. Even assuming a fourfold price differential the Tunny rig would only have to provide a 10% fuel saving to earn its keep. Experience suggests that this will be an easy target to meet, thus giving great confidence that the twin Tunny rig could provide economic sail assistance in this project.

Figure 16

SANDSKIPPER 28 in SRI LANKA
PREDICTED ECONOMIC PERFORMANCE

ENGINE:12.5HP DEUTZ DIESEL

RIG 37sq.m LATEEN

NETS:30 UNITS ,120 MESH DEEP

REVENUE DERIVED FROM SK24 DATA

MEDIAN VALUE FROM 4 BOATS

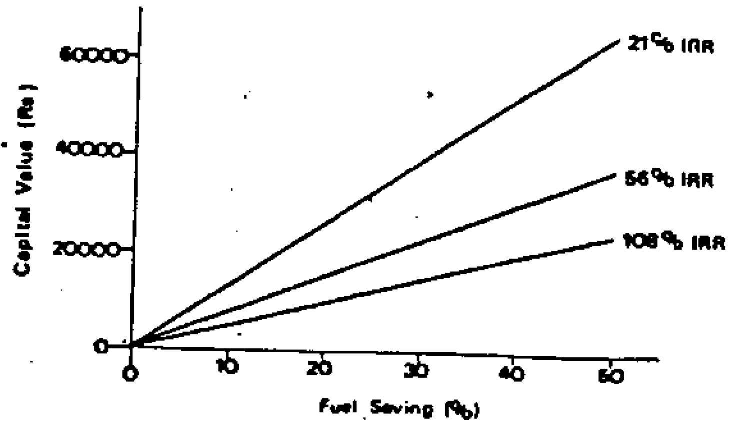
DISCOUNTED CASH FLOW ANALYSIS GIFFORD and PARTNERS 1984

	YEAR 1	2	3	4	5	6	7	8	9
CAPITAL COSTS									
Boat+sail	95000.00	.00	.00	.00	.00	.00	.00	.00	.00
Engine	35000.00	.00	.00	.00	35000.00	.00	.00	.00	.00
Nets	64000.00	.00	.00	.00	64000.00	.00	.00	.00	.00
Equipment	3000.00	.00	.00	.00	.00	.00	.00	.00	.00
Beach winch	9500.00	1500.00	1500.00	1500.00	11000.00	1500.00	1500.00	1500.00	.00
Sail replacements	.00	.00	2000.00	.00	2000.00	.00	2000.00	.00	.00
TOTAL	206500.00	1500.00	3500.00	1500.00	112000.00	1500.00	3500.00	1500.00	.00
OPERATING COSTS									
Fuel (l)	1875.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	625.00
Fuel rate	8.30	8.30	8.30	8.30	8.30	8.30	8.30	8.30	8.30
Fuel cost	15562.50	20750.00	20750.00	20750.00	20750.00	20750.00	20750.00	20750.00	5187.50
Consumables	10500.00	14000.00	14000.00	14000.00	14000.00	14000.00	14000.00	14000.00	3500.00
Labour	36000.00	48000.00	48000.00	48000.00	48000.00	48000.00	48000.00	48000.00	12000.00
Maintenance	7500.00	10000.00	10000.00	10000.00	10000.00	10000.00	10000.00	10000.00	2500.00
Winch fuel	994.00	1325.00	1325.00	1325.00	1325.00	1325.00	1325.00	1325.00	331.25
TOTAL	70556.50	94075.00	94075.00	94075.00	94075.00	94075.00	94075.00	94075.00	23518.75
TOTAL COSTS	277056.50	95575.00	97575.00	95575.00	206075.00	95575.00	97575.00	95575.00	23518.75
REVENUE	161250.00	215000.00	215000.00	215000.00	215000.00	215000.00	215000.00	215000.00	53750.00
NETT CASH FLOW	-115806.50	119425.00	117425.00	119425.00	8925.00	119425.00	117425.00	119425.00	30231.25
DISCOUNTED CASH FLOW	-59081.60	31087.30	15595.20	8092.33	308.35	2106.50	1054.75	540.35	70.82
DISCOUNTED BALANCE									-218.79
INTERNAL RATE OF RETURN	96.00								

Figure 17

Capital Value of Fuel Saving

At Constant Internal Rate of Return



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A MICROCOMPUTER-BASED INSTRUMENTATION PACKAGE FOR ON-BOARD WIND-ASSIST MEASUREMENTS

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ABSTRACT

Described is an inexpensive instrumentation and data logging package applicable to performance measurements of wind-assisted commercial vessels as well as sailing yachts. Power is supplied by small dry cell batteries to a CMOS technology lap computer: The Epson HX-20. Data are logged on the self-contained microcassette tape. Sensors are used to measure and log data on: apparent wind speed, apparent wind direction, vessel speed through the water, hull leeway or yaw angle and angle of heel. Experimental runs are plotted and errors discussed.

INTRODUCTION

This work was preceded by the earlier, development of a microcomputer data acquisition package using an Apple II+ microcomputer, 12 volt marine storage battery, inverter and a $4000 Cyborg Isaac plus sensors for a total instrumentation expense of about $10000. The work was concluded successfully and has been reported in detail.(1)(2) Earlier work in this field is described in References (5) and (6).

With little time or monies available, it was decided to attempt to develop an inexpensive set of instruments to accomplish the same functions. The object was to measure the following parameters aboard sailing vessels and yachts: boat speed through the water, apparent wind speed, apparent wind angle, yaw or leeway angle and heel angle. In addition, earlier work used a roll monitor from Ocean Motions Company to record metacentric height - changes in vertical center of gravity. A diesel fuel flow sensor was also obtained from that company.

This series of experiments was conducted to measure the first five parameters listed above and do so inexpensively with low battery drain equipment. The CMOS technology Epson HX-20 notebook computer with rechargeable, low current drain batteries proved ideal. It is thanks to Chet Swenson, Marine Engineer, whose idea this was and the generosity of Epson America Corporation, that we were provided with this beautiful small computer. Total cost including computer, analog to digital converter box and sensors was about $ 1600. This could be reduced to $ 1300 by successful junkyard scrounging to measure the first five parameters listed.

There was only time for one day of data acquisition with this new package. As in almost all experimental projects, one problem occurred which was not evident until after the experimental day: boat speed was measured but not recorded! There was no time for a repeat series of measurements, and the data presented here are thus incomplete.

EXPERIMENTAL EQUIPMENT

The series of four photographs in Figure 1 show the 19 foot modified Cougar catamaran used as a test bed for these and earlier experiments. The sensors are mounted on an aluminum pipe well forward of the sail plan in clean air and water on all points of sailing except running. See Reference (2) for a thorough discussion of this arrangement. The wind sensors shown here are from Kenyon Corporation. The wind was heavy that day - hence the double reef in the mainsail and the generally poor sail set. Figures 2 and 3 show the instruments. The computer and analog to digital box were housed in an attache case. The coffee can contains the joystick with a plumb bob mounted on the stick. The aluminum pipe pierces the water and holds a hydrofoil-shaped foam and fiberglass instrumentation housing for the water speed paddlewheel sensor. A two-dimensional hydrofoil blade of five square inches senses leeway angle by turning in a set of bearings and shafts taken from an old bicycle steering column. The instrumentation plate is a steel bookshelf with holes for mounting all equipment including the roll monitor not shown here.

Figure 4 presents a block diagram of the instrumentation package showing the general arrangement of computer, analog to digital box and the sensors. Note that two heel angle inputs were taken. There were two potentiometers on each of the two axes of the joystick, and signals were taken from two of these for comparison.

EXPERIMENTAL PROCEDURES

Data was acquired at 4800 baud, and it was tentatively found impossible to increase the rate. Three hours of data fit on one side of a 30 minute microsette tape, and 2400 sets of data were logged in that time. Thus, each set of data took approximately 4.5 seconds. Of that time, the one digital input: boat speed, took 3 seconds. Yaw angle, wind speed, wind direction and the two heel angle measurements each took about 0.3 seconds. Courses were sailed from zero to 180 degrees, on port and starboard tacks. Wind was gusting to 18 knots or so.

Unfortunately, there was not time to back-calibrate the wind speed sensor, so that raw data has not yet been analyzed. As noted above, Boat speed was not logged due either to corroded or loose electrical contacts or to salt water entering the potted knotmeter sensor case. Cracks were noted in the potting compound after the test. In the Reference (1) and (2) tests, a strobe light was used to pre-calibrate the paddle wheel sensor.

COMPARISON WITH ANALYTICAL METHODS OF PERFORMANCE PREDICTION

There have been various successful attempts to measure or predict sailboat performance. See references (4) through (15) for examples. Data were taken from the earlier experiments (1)(2), and are plotted in Figures 5 and 6. A towing test of the catamaran test bed gave good drag figures for the fouled hull used during those experiments. A quadratic equation was fit to the data and provided resistance information for entering the performance prediction computer program.(3) Results from the computer program are shown as crosses and crosses in circles in

Figures 5 and 6. The experimental data clearly show the differences in sailing this particular craft on port and starboard tacks. Most sailing vessels will sail better on one tack than the other. Although measured, leeway angle was not included here. That would shift the early part of the experimental curve to the left. Leeway angle directly reduces apparent wind angle and thus true wind angle is also reduced by the usual trigonometric relationship.(10)(11) Another shift to the left would occur from a cleaner hull bottom and more experienced helmspersonship. Pinching is clearly evident when close-hauled. Thus, the analytical and experimental curves would then tend to agree even more than is shown in the figures.

The experimental points in Figure 5 are the result of intensive statistical analysis as discussed in detail in Reference (2). Each point represents many individual measurements. The vertical bars are 95% confidence limits which give a measure of consistency in the data. In Figure 6, a sixth order polynomial was fitted to the mean data after some experimentation.(2) A spline fit might have been better. This emphasizes even more clearly the differences in the close-hauled condition as discussed above.

EXPERIMENTAL PROBLEMS AND RECOMMENDATIONS

The computer and analog-to-digital box required protection from salt water while in an open boat on a windy day. The attache case used was helpful, but a permanent case with transparent window and an RS-232 socket in the case would be preferable. Perhaps a sealed, membrane keyboard would be an improvement. Much salt water corrosion of the mechanical parts was noted a few days after the test run. Salt water resistant materials should be used. Voltage dropping resistors on the instrumentation platform were also corroded and need to be in water tight enclosures. There was considerable noise in the signals leading to variations in data obtained when such should have been constant. The reason for this is as yet unknown to the authors. Speed of reading the digital data inputs was limited by the use of BASIC and could have been much improved had there been time to write that routine in machine language. A much preferable solution would be to ensure that all inputs are analog. Simple, rugged and accurate boat speed sensors have been constructed with Simerl generators and paddle wheels.(15)

There were serious problems with battery drain, although one 6 volt lantern battery was used. The analog-to-digital box uses only 50 ma. and the port will handle 6 to 24 volts. The sensors were relatively heavy current users, especially the boat speed paddle wheel, and battery voltage dropped from 6.7 to 5.35 volts after five to six hours use. The voltage dropping resistors used with the analog sensors also consumed power. The voltage drop led to erroneous readings. A separate battery pack will be necessary perhaps with a zener-regulated supply.

CONCLUSIONS

This brief project was successful in terms of proving the practicality of inexpensive computer-aided data acquisition for performance measurements of sailing craft. It is unfortunate that data was not recorded from the key sensor: that for boat speed. It was very easy to program and use both the Epson HX-20 and the ADC-1. Technical advice from both companies was excellent as was their willingness to discuss problems and suggest solutions. The package ran at sea for over five hours with no problems.

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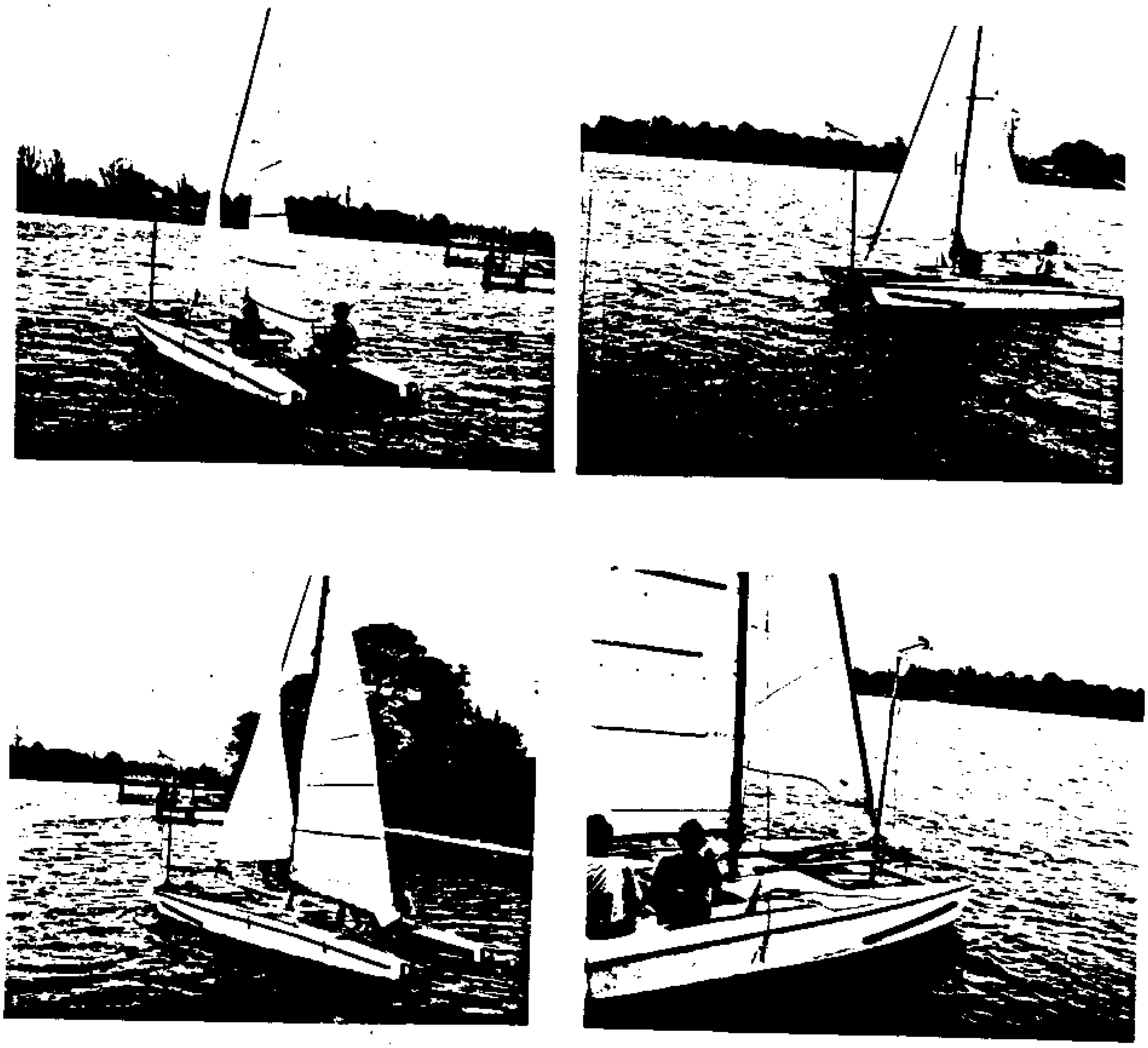


Figure 1.

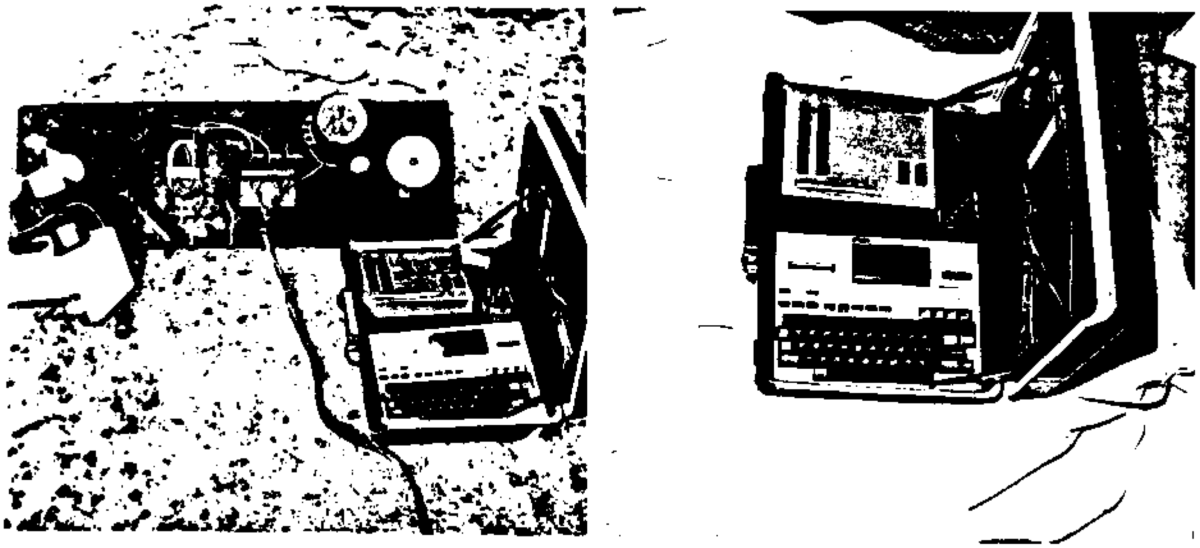


Figure 2.

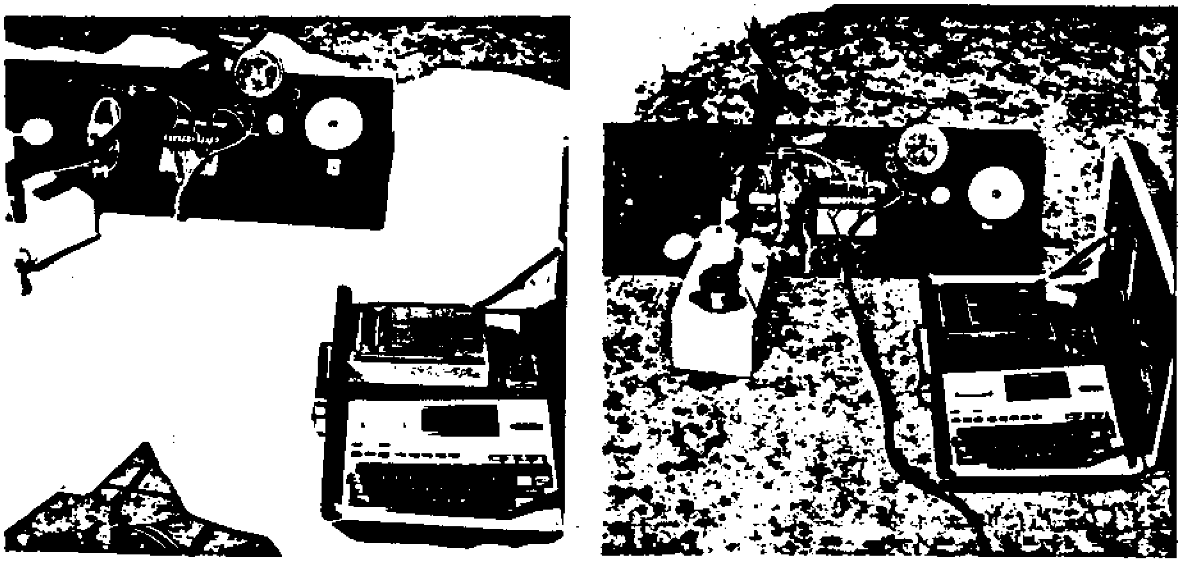
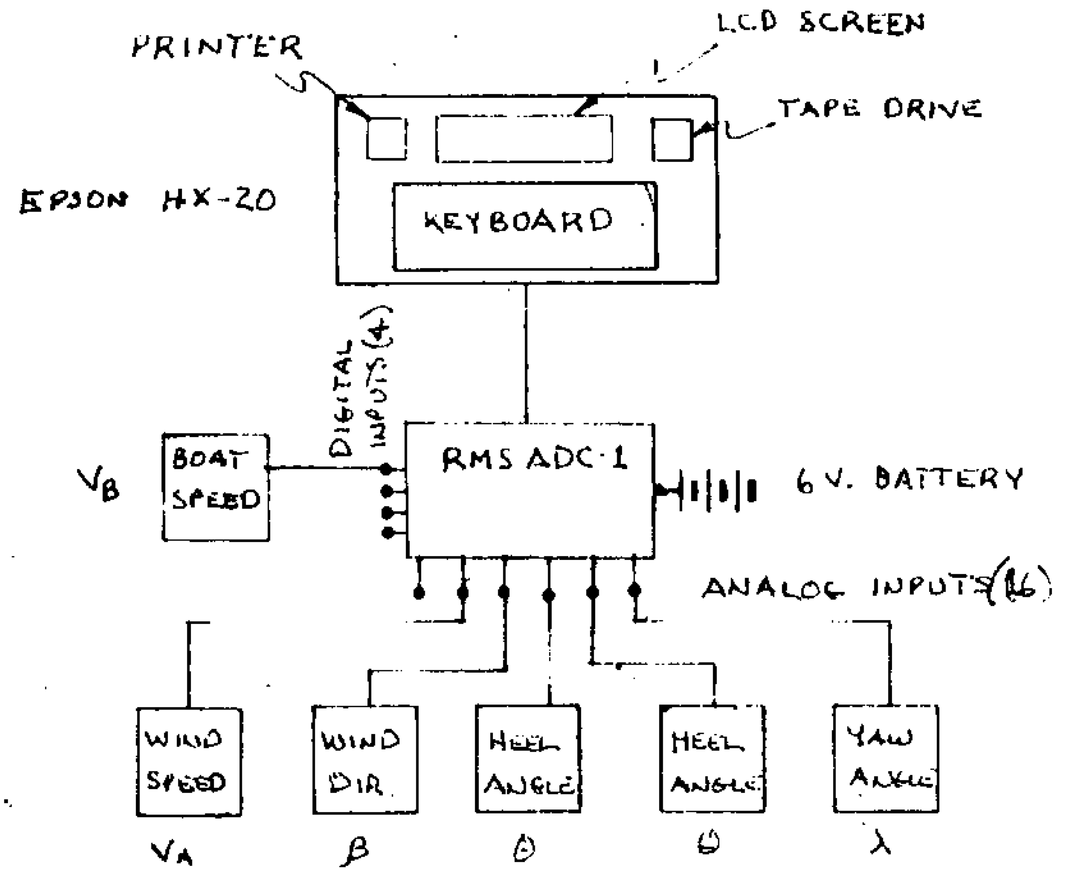


Figure 3



BLOCK DIAGRAM
INSTRUMENTATION
PACKAGE
FIG. 4

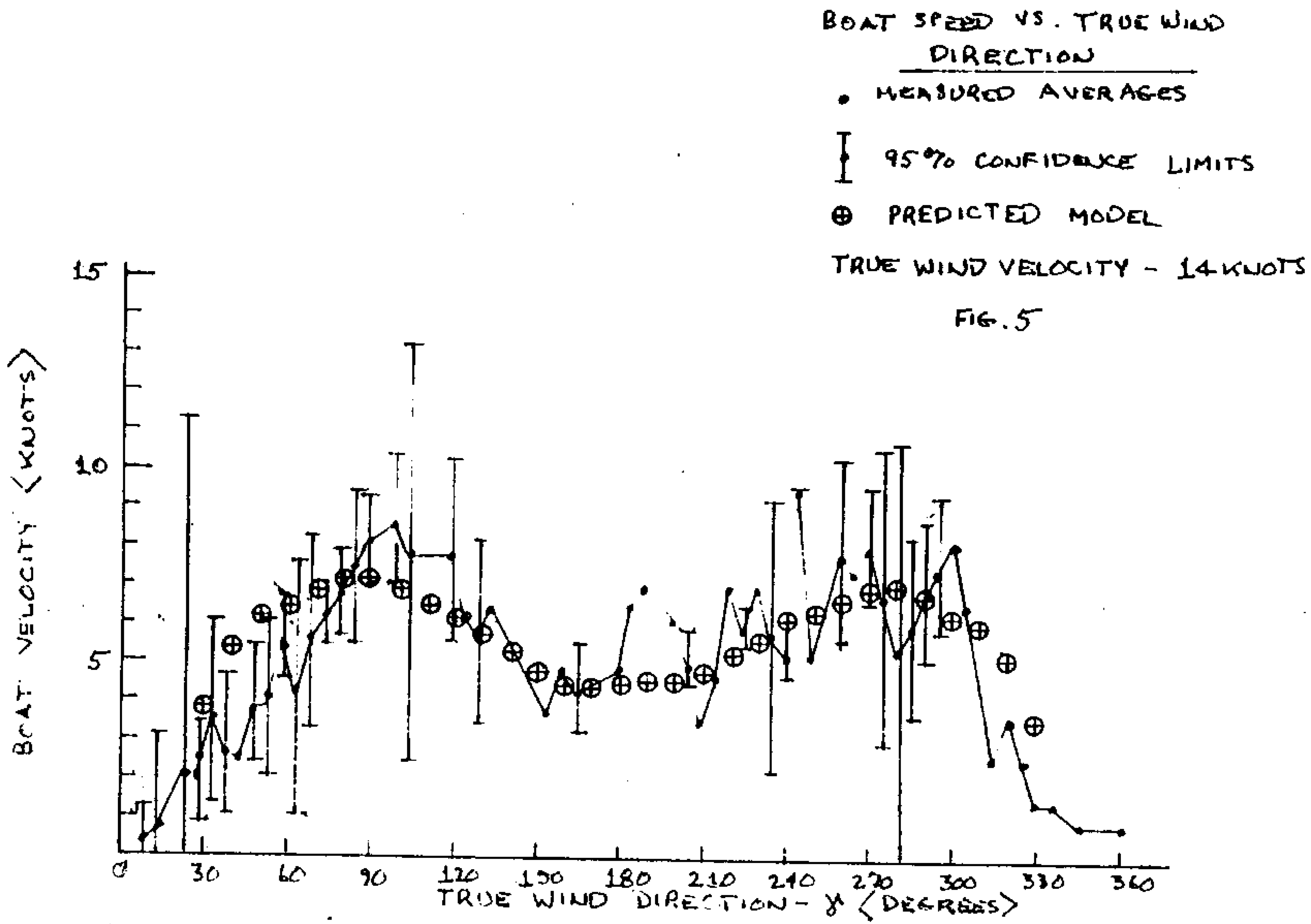

BOAT SPEED VS. TRUE WIND DIRECTION

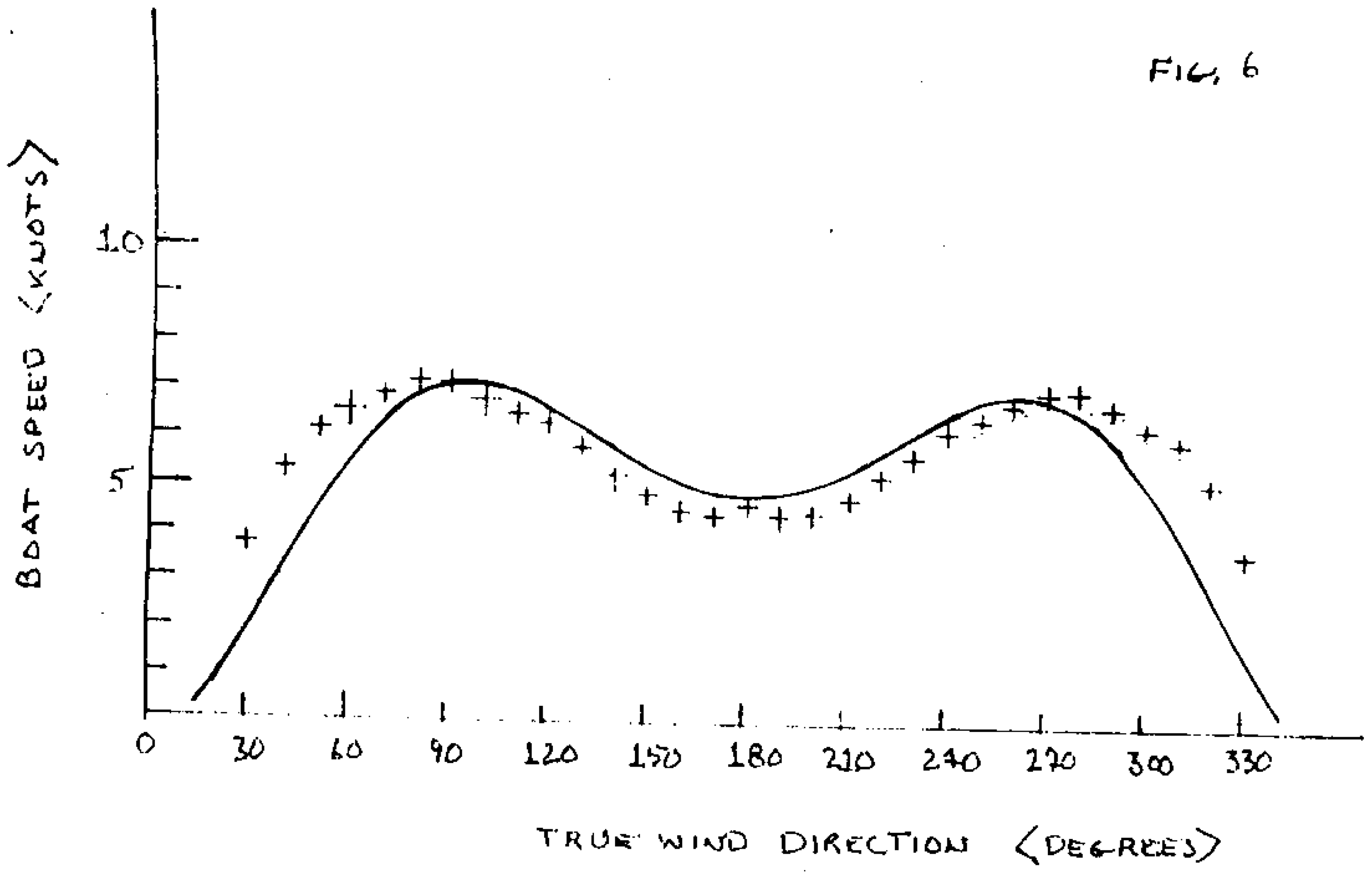
FIG. 5

BOAT SPEED VS. TRUE WIND DIRECTION

SOLID LINE - CURVE FIT TO MEAN DATA
6TH ORDER POLYNOMIAL

\+ ANALYTICAL MODEL PREDICTIONS

FIG, 6



DESIGN AND TESTING OF A FISHING VESSEL WITH COMBINED MOTOR/SAIL DRIVE FOR THE ARTISANAL SMALL SCALE FISHERY OF SIERRA LEONE

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Abstract:
Rising fuel prices during the last twelve years made it necessary to reduce the fuel consumption of fishing vessels especially in developing countries. To replace gasoline outboard engines used in the artisanal fishing fleet of Sierra Leone, by diesel inboard engines of considerably lower fuel consumption, major changes in design and construction of the traditional fishing crafts were needed.
An 11 m V-bottom boat driven by a 30 HP diesel engine and a standing lug rig of 49 m^2 was designed and tested. From the results it can be seen that the new type of fishing boat can be used successfully in the small scale fishery of Sierra Leone.

Since 1980, the Fisheries Pilot Project Tombo (FPPT), a German development aid project of "Deutsche Gesellschaft für Technische Zusammenarbeit"(GTZ), is engaged in developing the artisanal small scale fishery in Tombo, Sierra Leone.

There are two main fields of activity in this project:

1. Improvment of the fishing technology of local fishermen.

2. Introduction of improved types of fishing boats.

Local traditional fishing crafts cover a wide range from small 6m-dugouts driven by paddles to 18 m Ghana-type planked canoes driven by gasoline outboard engines with a power up to 40 HP(29.8 kw).
High fuel consumption, frequent repairs, and short life time make these outboard engines one of the most expensive parts of the fishing equipment. All these problems could be solved by introducing inboard diesel engines, but this cannot be done without major alterations in hull shape and constructional details of the traditional canoes.

For these reasons, the FPPT started the design of a V-bottom boat, based on a type which was developed by the FAO in 1974. /1/

The main dimensions of this new design are:

Length over all	Lo.a.	=	10.90 m
Length between perpendiculars	Lp.p.	=	9.85 m
Beam moulded	B	=	2.80 m
Depth	D	=	1.10 m
Draught	T	=	0.60 m
Displacement	Δ	=	6.60 m^3

The body plan is given in Fig. 1.

The boat ist driven by a Yanmar Diesel Engine 3 QM 30 of 30 HP (22,4kw)at 2600 rpm with a reduction gear of 2.21 : 1. The 3 blade propeller has a diameter of 460 mm and a pitch of 330 mm.

For the rig a standing lug sail of 34 m^2 was chosen with an additional jib of 15 m^2 (Fig. 2).

Calculating the sail area/displacement ratio

$$T_s = \frac{\sqrt[2]{A}}{\sqrt[3]{\Delta}}$$

A = sail area

Δ = displacement

we find T_s = 3.73. This corresponds to values of T_s published by Timmermann /2/.
Mast, boom, and yard were made out of bamboo giving high strength at low weight.

Speed tests were performed in November '83 at the Yawri Bay, south of the Freetown peninsula. The boat's speed was measured by means of a Dutchman's log and a stop watch. To measure the wind speed, a cup anemometer was available.

At full engine power, a mean speed of 8 kts was obtained at a displacement of 5.1 m^3. With less displacement (3.0 m^3) the speed increased to 8.6 kts. Speed measurements when sailing were more difficult because of the low winds prevailing during this time (October - December) at the coast of Sierra Leone. Only few data could be obtained which should be completed at times of better wind conditions. Sailing on a course broad reach to down wind, a boat's speed of $v = 2.4$ m/s (4.6 kts) was measured at a wind speed of $u = 5$ m/s.

Rolling tests were performed to check the initial stability of the boat. With a rolling period of $T_R = 2.5$ s, a metacentric height of GM = 0.56 - 0.65 m was calculated according to the formula

$$GM = \frac{(\pi \cdot B \cdot C_r)^2}{T_R^2 \cdot g}$$ /3/

Cr = 0.79 - 0.85
B = 2.34 m (beam at waterline)

The last test of this series was a one day fishing trip with a local crew of 15. The crew experienced in using an 18 m outboard powered canoe, had no problems to handle the gear, a traditional ring net, aboard this new type of boat. Shooting and hauling the net was done in about the same time needed on a traditional Ghana-canoe for these operations.

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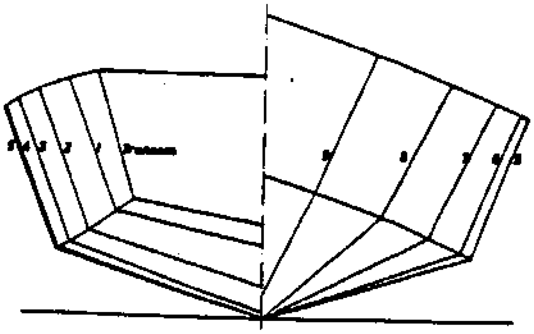

Figure 1. V-Bottom Boat, bodyplan

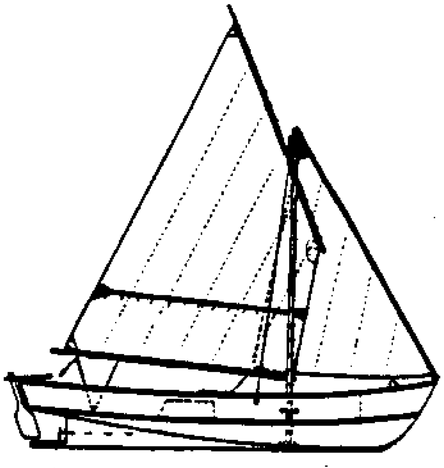

Figure 2. V-Bottom Boat, sail arrangement

A PROGRESS REPORT
WITH SPECIAL REFERENCE TO
A 30.5 M FISHING CATAMARAN

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1. Introduction and background

An earlier paper presented to last year's International Conference on Sail-Assisted Commercial Fishing Vessels in Florida, USA in May 1983 gave the overall background to our project, some historical information, and a broad overview of what we are trying to achieve with respect to commercial craft of all types. Since then we have consolidated rather effectively, obtained major venture capital funding and filled in many of the grey areas in our technological knowledge. We have completed the design of a nominal 100 sq. m. triplane thrust unit, which we call the Module 2, and we are in the exciting early stages of manufacture of the first prototype.

2. Aerodynamics

We decided that, even though we had a very great deal of aerodynamic information about both single and triple aerofoil combinations, we needed to build a 1/10 scale wind tunnel test model of the actual Module 2 complete in every detail. This quite large device was too big to fit into the Markham tunnel at Cambridge University Engineering Department, which we had used so effectively for most of the earlier runs, and we therefore made arrangements to test it in the large 2 Megawatt No. 4 wind tunnel at British Aerospace at Filton near Bristol. By an odd coincidence this was where I had started my aero-engineering career some 15 years before.

The model is suspended in the conventional way, Fig. 1, horizontally in the tunnel and lifting downwards, with an upstream projecting sting linked to a vertical wire controlling the model in angle as well as measuring the turning moments about the main suspension points. We have now done three separate series of tests at Filton on this model with both zero flap and full flap conditions, tail on and tail off and over a wide range of Reynolds numbers, and feel that we are getting to grips extremely well with the complex aerodynamics of high performance triplanes.

The last time that any work was done on triplane aerodynamics in any serious way was of course before the 1st World War, and we have found that to be only of marginal relevance in 1984.

The tests have confirmed that we can control the triplane very satisfactorily in the CL = 2.3 to 2.5 range, accepting of course that the triplane CL max. will inevitably be lower than the CL max. obtainable on a single aerofoil. This is because of interference between the three planes, which in some ways is beneficial, delaying stall and therefore improving the trim tolerance of the device; in other respects being detrimental, in that the actual overall thrust per sq. m. is marginally degraded. We have also simulated the storm survival case where the wingsail must weathercock in winds up to 75 m per sec, something of the order of 150 knots, neither damaging itself nor putting the vessel into any kind of danger or discomfort. The results are all quite favourable, and no modifications to the design as it progresses have been forced by aerodynamic constraints.

FIG. 1.

3. Structures

The design of the triplane has moved ahead very strongly in the months since we were writing last May, and we now have a very clear idea of the best way to build the type of structure envisaged. We have settled upon a standard 400 mm square rolled hollow section steel central mast up the middle of the centre wing, carrying the entire bending moment of the sailset, and mounting the entire triplane down onto a standard off-the-shelf slewing ring type rotating bearing, very similar to the ones which give extremely reliable service underneath small and medium-sized earth movers and diggers. This central spar is clad with streamlined reinforced plastic aerodynamic surfaces, and supports a pair of cross-arm structures which carry the outer wings, Fig. 2.

The outer wings are of identical profile to the central wing, but have a much lighter spar structure, since they only have to support their own bending moments and communicate their thrust onto the central spar through the cross-arms. Hinged onto these three wings, the outer ones of which are set slightly forward in a symmetrical staggered configuration, so as to trim the nett centre of pressure to a suitable position, are three flaps. These are all very nearly identical, although the centre one has slightly stronger hinge structures which react the hydraulic loads from the hydraulic actuating cylinders and also drive pushrods which cause the outer flaps to move in unison with the centre flap. We have two hydraulic cylinders operating the centre flap working in harness, arranged so that in the event of one failing due to a burst hose or a leaky seal the other will still be able to carry the load.

Supported downstream of this high thrust triplane is the tail vane, which is full height and is slightly longer in chord than either the main wing or the flap. It is again supported at two points by triangulated bracing struts and is again operated by two hydraulic rams for safety and reliability, even though they are considerably smaller in size and thrust rating than the ones which operate the main flaps.

The general structure of the triplane, apart from the steel central spar which we have already mentioned, consists of a reinforced plastics D-box leading edge with moulded ribs leading back from the D-box to a trailing edge in a fairly conventional aircraft style of construction, with all the lightly loaded panels covered in a specialist American aircraft covering material called Ceconite. This is a heat shrinking self primed material and we expect it to give extremely good performance at moderate cost, with the ease of repair of minor rips and defects using readily available on the spot methods.

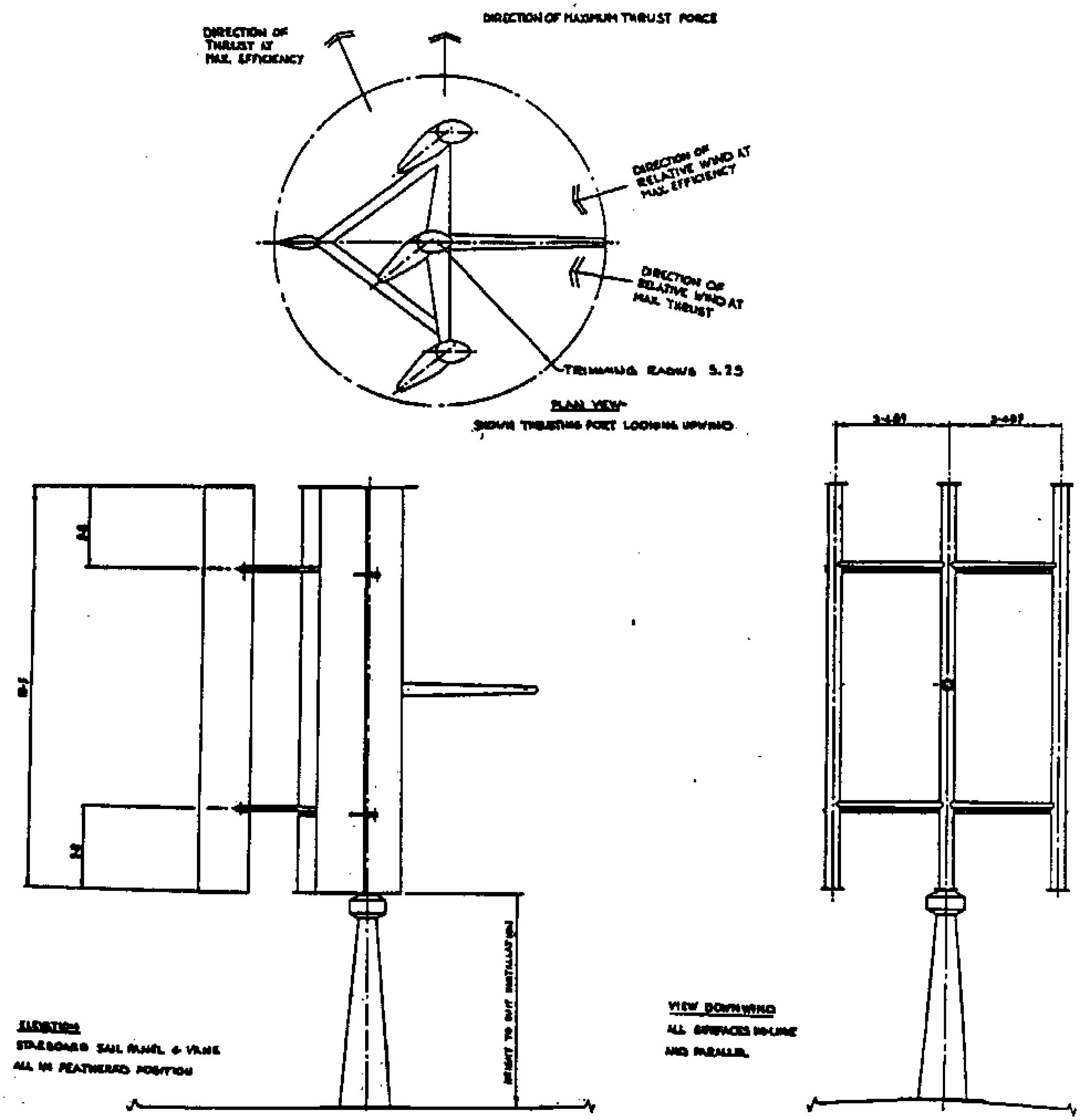
DIRECTION OF THRUST AT MAX. EFFICIENCY
DIRECTION OF MAXIMUM THRUST FORCE
DIRECTION OF RELATIVE WIND AT MAX. EFFICIENCY
DIRECTION OF RELATIVE WIND AT MAX. THRUST
TRIMMING RADIUS 5.25
PLAN VIEW
SHOWN THRUSTING PORT LOOKING UPWIND
ELEVATION
STARBOARD SAIL PANEL & VANE
ALL IN FEATHERED POSITION
HEIGHT TO SUIT INSTALLATION
VIEW DOWNWIND
ALL SURFACES IN-LINE
AND PARALLEL.

FIG. 2.

Ahead of the triplane on a short stub pointing upstream from the central steel main spar is a balance weight which provides static and dynamic balance for the rig as it pivots and weather cocks on its free bearing, using a balancing principle pioneered by Utne in Norway in the late 1930's. Earlier in the project we believed that we would need an "upstream stalling vane" which was shown on our earlier illustrations and photographs. This is because when you fully stall this type of triplane with a downstream tailvane the vane finds itself in a low energy eddying wake from the central sail. In the wind tunnel we had found that a very slight and mirror imageable asymmetry was perfectly capable of giving us reliable in-stall drag moments as soon as the peak of the lift curve was passed, and we have therefore eliminated, at least from this particular design, the upstream stalling vane.

4. Computer System

The computer system is necessary for any economy device such as our own because for a vessel to work, whether she be a tanker or a fishing boat, she must have her crew available to do the work for which they are trained and needed. If we instal onto such vessels an economy device which increases the crew loading, either in terms of overtime payments or actual extra crew members, then we are taking away at one sweep much of the benefit which we are able to bring to the owner and operator of the craft by fitting the economy device in the first place.

We therefore equip the wingsail with a 3 position switch for "On, Ahead"; "Off, locked and isolated" and "On, Astern" and 3 lights, one being green for "all systems on and operating"; one being amber for "a fault has occurred but redundancy is coping and the unit is still in operation"; and a red light indicating that there is a fault present which the redundancy system has not been able to correct and that therefore the system has been shut down, all vanes brought to central, and is weather cocking with small drag and no cross wind force.

The computer is based upon the well known Zilog Z80 chip, made most famous perhaps by Sir Clive Sinclair with his ubiquitous range of computers for the everyman. Fig. 3. The unit takes information from several transducers. It receives information from a vane mounted on an instrument strut extending upstream of the unit, as to the angle of the wind to the centre axis of the wingsail, and it takes information from a potentiometer which reads the angle of the sailset to the centre line of the ship. From those two inputs the computer can know at all times both what the angle of the apparent wind to the sailset is and what the angle of the apparent wind to the vessel centreline is. From the angle of the apparent wind to the vessel centreline the computer decides whether to set the unit into one of four modes:-

- If the wind is coming from within 20 deg. either side of the vessel's centreline from ahead, the computer knows that there can be no benefit obtained by switching the sailset on and it therefore either maintains the weather cocked state, or it brings it to the weather cocked state if this circumstance has just arisen.
- If the apparent wind direction is between 20 and 60 deg. to the vessel's centreline, the computer knows automatically upon which tack the vessel is and it sets the sailset up correctly for that tack and for optimum lift/drag ratio.
- If the wind is between 60 and 135 deg. to the vessel's centreline the computer sets maximum thrust.

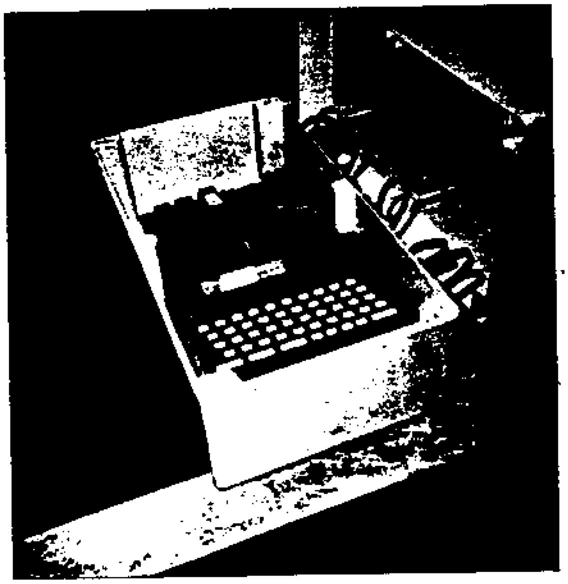

FIG. 3.

- If the wind is between 135 and 180 deg. the computer stalls the rig to produce the maximum drag and the worst lift drag ratio that it can achieve.

The computer also receives information from an anemometer which tells it whether the wind is too low to provide significant energy, at 5 m/sec. or approximately 9 knots; whether it is in the working range of 5 to 20 m/sec.; and whether it is in the over 20 m/sec. range, where there is statistically very little energy available on a year round basis from these higher winds. It therefore, for a further 5 m/sec., platforms the output thrust, and after 25 m/sec. it reduces the output thrust down to zero by going to mode 1.

So we have mode 1 which is the wind ahead case, mode 2 for close reaching and close hauled on either tack, mode 3 for reaching and broad reaching on either tack, and mode 4 stalled, the spinnaker mode, for running downwind.

The next set of activities which the computer performs is the monitoring of the relative health of the various items in the system, so as to check whether the electrics, hydraulics, structure or computer are functioning as they should. In some cases there is available redundancy and the unit may still continue in the presence of a fault either at full or reduced efficiency. In the event of a fault which cannot be coped with by redundancy the computer will return all vanes and flaps to central, close everything down electrically, spring loaded pins locking all in-line for safe long-term weathercocking.

We at first thought that all this was going to be quite a difficult set of problems for a computer to solve. However, due to the advanced state of knowledge in the modern micro computer field, we have discovered that a simple low cost unit can be made up which will perform all of these functions, and which would probably have spare time to play Space Invaders in between whiles, if it felt so inclined.

5. Hydraulic System

The vanes and flaps are all moved from the central positions to their operating positions and back again, as we have already mentioned, by modestly sized hydraulic cylinders, some single acting with spring extension for locking purposes, and some double acting. These are all controlled by a set of solenoid operated spool valves arranged in a fairly complicated circuit designed to provide fail safety.

The logic of the circuit has been sketched out briefly above in the section on the computer system. The system receives a set of impulses and commands from the computer system:- for example when starting up with the wind between 20 and 60 deg. on the starboard bow the hydraulic motor provides a pressure of 200 bar, approximately 3000 p.s.i., charging up 2 small hydraulic accumulators. The system will then hydraulically withdraw all the locking pins from the flap and vane systems and push both flap and vane over to their full angles of action. Then it will allow re-entry of the flap locking pins to hold the flaps at full extension, and respond to the commands of the computer in controlling the vane so as to maintain the optimum lift/drag ratio of the main triplane, with CL about 1.2.

If the wind direction relative to the centre line of the ship changes into the 60 - 135 deg. sector, the computer causes the tail vane to look for a different operating position, seeking now for maximum thrust.

6. Installation on 30.5m Catamaran fishing craft

We are very interested indeed in fitting an early standard Module 2 unit to an approximately 30.5 m catamaran fishing craft. Fig. 4.

Such a vessel would have about 14 metres of beam, and provide an excellent stable working platform with large deck area for handling nets and catches, a high sprint speed for getting to and from fishing grounds with minimum loss of time, while capable of completely silent and vibrationless trawling and line hauling.

We are convinced that a quiet fishing boat must be likely to catch more fish than a noisy and vibrating fishing boat beating the water with propellors, although we should of course be very happy to receive the opinions of the experts on this subject.

The installation should present no particular problems:- we have taken a standard "Fjellstrand" design as used extensively around the Scandinavian coast and as oil rig service boats in the North Sea, and simply widened it by the addition of a parallel section to take the beam out from approx. 9 m to approx. 14 m.

We believe that this would not in any way degrade the seakeeping and other characteristics of the vessel as a power craft, be fitted with twin diesel engines and fully feathering controllable pitch propellors (a system which is already well proven in this class of vessel in any case). We believe that she will represent, either in steel or in welded light alloy a practical and cost effective vessel for medium offshore fishing purposes. We shall be very happy indeed to take this proposal, which at the moment is only at the scheme design stage, through to a stage of full analysis with any seriously interested parties.

If the wind goes round to astern, then the vane will extend to full angle to stall the main sailset.

At any time if there is a fault, or if the wind drops too low, or if it rises too high, then the electrical power to the hydraulic system is cut, main system pressure drops and hydraulic reservoir pressure is used to pull out the flap lock pins, return flaps and vane to centre, finally allowing the re-entry of flap and vane pins.

All this must be completely automatic since in storm conditions the last thing needed would be a human input to the system, especially on a small craft.

Because of the way that the system has been optimised, all the valves and components are small, at international CETOP 3 size, with 1/8 inch bore piping in most cases. The repair and replacement of elements which may have failed is therefore a modest and economical task.

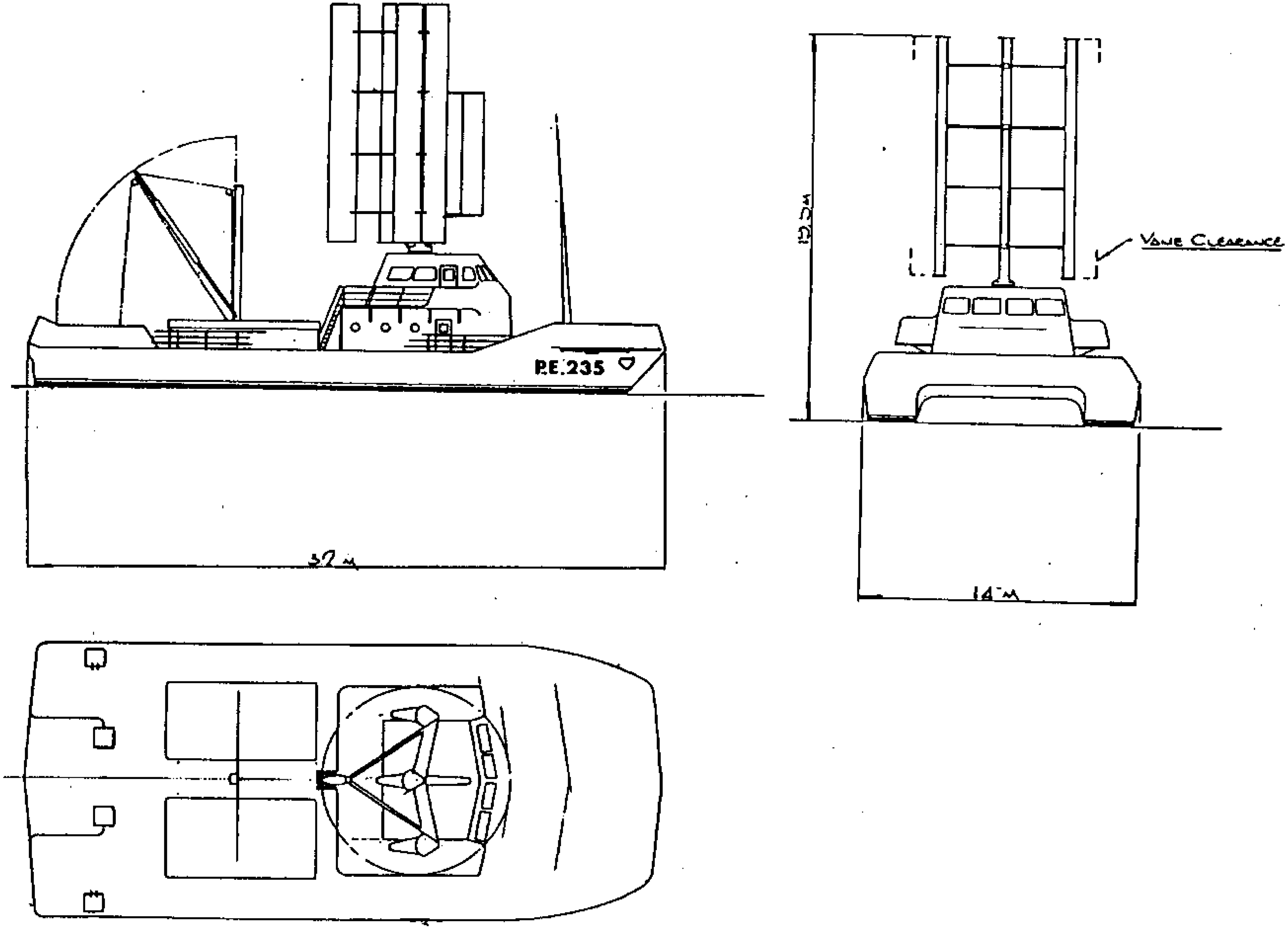

FIG. 4.

7. Summary

We are at the moment building, as we have said, the first prototype Module 2, and by the time that this paper is presented the main triplane wing elements will be complete and the first wing ribs will be coming through from the sub contractors ready for installation and fabric covering. Figs 5 and 6.

Completion and first trials are scheduled for Mid-September 1984, and we hope that by that time we shall have made not one but many contacts among people who either attend The International Conference or who read the proceedings, so that if our products could make a cost effective contribution to their activities, then we shall not miss any opportunity to put forward proposals, have discussions, and eventually negotiate contracts.

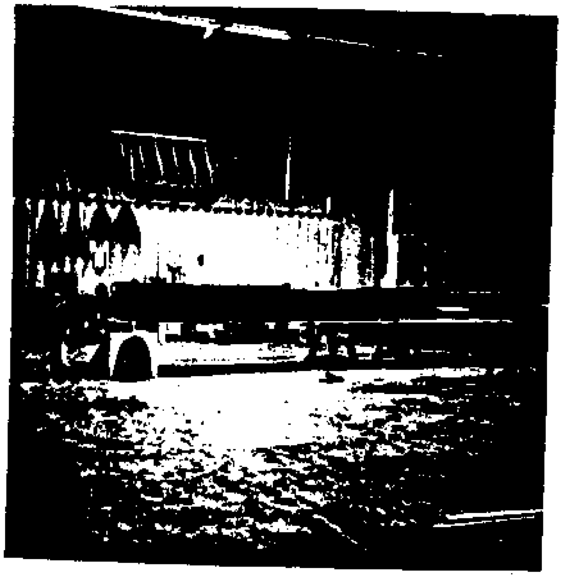
FIG. 5.


FIG. 6.

EASILY HANDLED RIGS FOR FISHING VESSEL SAIL ASSIST

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ABSTRACT

Once one has successfully mechanized sail handling it is relatively easy to automate it electronically.

Luff roller-furling of soft sails is the best way to set, reef and furl sails. A crane-boom whose rotation is cantilever controlled (that is not free pivoting as all booms have been) is the best way to sheet sails. Crane-booms free the rails and deck and are safe and very easy to handle.

INTRODUCTION

If the average fisherman is asked: do you want sails on your fishing boat most will answer "No!". Their negative answer is because they have never seen a truly well equipped modern sail assisted fishing boat. They do not realize how extremely little sail handling would be required. The truth is that none yet exist although right now such a vessel could be built. This paper tries to explain some important rig, sailplan, and deck machinery features of such a super-modern fishing vessel.

ADVANTAGES

A fishing vessel with sails would roll much less and thus could catch more fish in higher sea states. It could stay out in worse weather and be more comfortable and less tiring for the crew. In addition to these multiple advantages, it would save fuel and often travel faster and thus be a considerably better money earner.

MECHANIZED OR AUTOMATED

Mechanized sail handling is the first step towards automated sail handling. Once sails are mechanized it is relatively easy to automate them if one wishes. Whether or not to automate is primarily a consideration of equipment costs and maintenance versus a small amount of labor saving and medium amount of judgement obviating.

RIG AND SAIL CONFIGURATION

The basic sail system that I favor is a luff roller-furling sail whose clew is controlled by a crane-boom which may also be called a cantilevered-boom, davit-boom, or controlled-boom. As the name implies, this device is a long armed crane whose rotation is controlled at its foot or forward end so that it can never flog or get out of control. The clew of the sail is controlled by a "sheet-outhaul" which can be a single part or a multiple part tackle that goes from the clew to the tip of the boom davit. There are thus no sheets to pay out and no sheets to gather in when one tacks or jibes, for one simple rotates the crane by push botton or turning a dial to the new setting. Such a device obviates overlapping sails such as genoas which really have very little place on a working sailboat because they are, first: labor intensive, second: dangerous, and third: cause a clutter.

MAINSAILS AND MIZZENS

This writer believes that it is very important that roller mainsails and mizzens roll up in the open rather than rolling into a cavity in the mast. The absence of lips and cavities in the mast assures reduced friction and wear and obviates the possibility of jamming in the mast slot or cavity since they do not exist in my systems. The rolled up sail is a few inches abaft the mast, and never out of sight or out of reach.

MAST AND RIGGING

The mast may either be cantilevered or stayed. Cantilevering permits lowering the mast in tabernacles if it is important to have this capability because of limited air-draft.

CARGO AND BOAT HANDLING

It should be noted that a really good and powerful crane can serve multiple purposes in addition to trimming the sails. While under power or while in harbor, such a crane can make substantial lifts from the water or from the shore. It can thus on and off-load gear or cargo of all kinds, and boats of any size that might be desired.

SAILS

Sails would be soft and made of the best sailcloth material available and they could be single ply or multiple ply in order to achieve the desired strength.

Sails are triangular and can be of jib, staysail, mainsail, or mizzen configuration.

Ideally all sails on a given vessel might be the same size so that the craft can carry a spare on board or shore based one for quick replacement when necessary.

ASPECT RATIO

Since the sails that are trimmed by a crane-boom cannot overlap a mast, the most convenient way of having sufficient sail area is to design and specify a tall mast or masts from the outset.

ROLLER STAY

It is our practice to use stainless steel tubing with a stainless steel groove to support and roll up sails. Aluminum is considered much less suitable because of its softness, which cannot resist scratches, or deep scoring which create stress raisers. Because of aluminum's low Young's Modulus of ten million compared to stainless steel's thirty million, an aluminum tube will either stretch or twist more per given tension or torque than stainless steel. Such a stainless steel tube stay is ideally configured to take torque, as well as tension. It can be shipped in 30 foot lengths and joined together by pinned spigots as is the practice of the British Rotomarine when they are supplying roller furling gear to larger vessels. We have employed more than sixteen such units and we have found them to be sound and durable on ocean going vessels.

We do not consider a combination of one by nineteen wire with sections of aluminum extrusion outside separated by plastic sleeves durable enough or of sufficiently long life. Two of these materials (aluminum and plastic) are too soft and in combination steel and aluminum invite corrosion, particularly in the tropics. The above admittedly has a price advantage for pleasure craft but lacks the durability required by workboats, that have along season, hard use and require minimum overhaul time.

THE CRANE BOOM

The crane boom can be actuated at its base by cogged gears, worm gears, hydraulic rack and pinion and various other systems. It could also be controlled by timing belt, roller chain, or cable and quadrant. Since it is a question of controlled rotation any device similar to steering gear can be applied to control the rotation of the vertical post on which the crane boom rotates. Actually, most crane technology and best practice is applicable to this fairly standard crane application.

ADVANTAGES OF CRANE BOOM

In addition to the already mentioned advantages of the crane boom, this type of sheet control greatly reduces the need for sheet winches, since a multiple part tackle of say four parts can be used, this means that a winch of one quarter of the power will do the job. Usually a headsail requires two sheets and two winches whereas the crane-boom can have one light winch of one eighth the cost of the two bigger winches.

This is of particular advantage to a commercial fishing boat for it frees the decks of headsail sheet winches and the lines at the rail that cross the deck as in older sail trimming systems.

SAFETY FEATURE

Loose-footed luff roller-furling jibs usually have a high clew to minimize re-leading the sheet as the clew moves forward during reefing. This high clew creates a dangerous flogging sheet problem. On the davit-boom the sheet is very short and does not flog since it is never loose. As reefing commences the lead slides forward automatically thus keeping the sheet lead relatively short and free from flogging and being a danger to the crew. The long lead from a high clew to the rail of a normal completely loose footed sail creates a true hazard and people's heads and other parts have been flogged by these high clewed jib sheets. I have seen a man with glasses get a bad cut across the face as his glasses were ripped off him by a flogging sheet and it could have been considerably worse. I am sure that this hazard repeats itself a great many times per year as high clewed jibs are becoming more and more popular for the above named reasons.

Thus the benefits accrued by crane-boom include: lower costs, because of substantially reduced number and power of winches, much less cluttered decks, safety from flogging sheets, substantial labor saving, and mechanization which is easily automated.

COSTS

In fairness it should be admitted and stated that the crane boom is itself expensive, requires deck reinforcing and takes up some space on deck a few feet aft of the mast or stay that it serves.

It is hard to say whether the initial cost of a crane boom cancels out the cost of the multiple sheet winches needed for a loose-footed headsail but I would estimate that they fairly well cancel each other out. On the other hand, there can be no doubt that a cantilevered boom would quickly amortize its initial costs and maintenance because of the substantial labor saving.

BREAKTHROUGH IN SAIL HANDLING

The crane boom is surely the one single greatest contributions to commercial sail since luff roller-furling and reefing which has been progressing for over 70 years. Crane-booms are still extremely rare but in this writer's estimation they are very promising and they have an excellent future. We should be seeing a great many more crane-booms on larger vessels. They are so convenient and safe that they will probably also be used on moderate sized, and large yachts to minimize crew requirements. The idea of fixing a point in space by the tip of a crane is certainly not new, but up to this point it was considered too expensive and too much weight compared to restricting swing by a sheet at the after end of booms.

The main reason that this author recommends cranes now is that labor is expensive and a swinging boom, free pivoting at this forward end is dangerous and labor intensive to control by conventional sheet, not to mention foreguys, boom vang, and topping lifts.

CONCLUSION

A sail assisted fishing boat that is equipped with external luff roller-furling sails and crane booms would be so easy to handle that fishermen would accept them. They would soon see the great anti rolling feature of sail and the increase profit from fuel saving.

We propose to offer commercial fishermen air foils that appear and disappear by push-button and that are oriented by turning a dial.

Handling sail of this kind would be so simple that one would not need to go further, but making the entire system automatic has already been done by the Japanese and is relatively easily accomplished if one desires to automate electronically.

Such a mechanized sail handling vessel, be she automated or not, would eventually be popular with the fisermen and would more importantly be a more profitable craft. She could catch more fish in worse sea conditions and have fuel as well.

It is this writers opinion that if one good one were built, that many more would follow.!

WRITTEN CONTRIBUTION

Gunnar C.F. Asker

It is evident from the number of papers on this subject that the interest is considerable, but the acceptance for commercial operations with wind-assist systems has been slow.

This is partly due to the fact that the economic benefits to most operators are marginal today.

We must however be ready with fully developed systems, which are practical and easy to operate, when the oil prices again will rise. The increased costs of drilling and extraction will soon neccessitate higher oil prices.

20th century technology is available to apply new practical techniques to utilize the oldest method of ship propulsion, which was oars and wind power. Human oar power plus wind power was the first motor sailer.

A wind-assist system saves fuel, but equally important it provides the fishermen with a stabler platform from which to work. The reduced rolling and pitching means less energy lost in proceeding through the seas and this results in more fuel savings.

After development and design of several different approaches, which included rotor thrusters as well as wing-sails I have become convinced that for smaller vessels the combination of a roller furled Genoa side sail combined with a small permanently installed wing-sail provides a practical means of wind-assist system.

For larger commercial vessels, such as general cargo ships and tank ships my patented "Twin-Rotor" wind-assist system would be preferable, because of reduced overall size. This system is described in US Patent #4,398,895, which issued August 16, 1983.

The "Flexi-Sail" wing-sail was first tried in a small lake with a radio controlled 3 foot long model shown in figure 1.

Then my 24 foot Ulrichsen Sea skiff was equipped with a 36 ft^2 wing-sail after two parallel plate keels and a sailboat type rudder had been added. The metacentric height was unchanged and also the displacement, because a 660 LB 100HP gas engine was exchanged for a Volvo-Penta 23 HP diesel of approximately half the weight. The Volvo engine was kindly furnished by Volvo-Penta.

The experience with the 360° fully rotatable wing-sail system is that the hinged jib-sail to the main wing-sail enables the boat to point higher than what is possible with a soft sail of equal area. The wing-sail is easy to control and can, because of the modest sail area, remain mounted in all kinds of weather.

The same boat with a flying bridge installed would have less stability and considerably higher drag.

Rolling and pitching was reduced and the boat goes through waves more smoothly, in fact higher speed can be maintained in choppy seas without discomfort compared to a boat without wind-assist system. This was evidenced by observing the inclinometer onboard.

At 5 to 6 knots boat speed in a 20 to 25 knot wind speed the fuel saving approximated 50% with the engine delivering about 4.5 to 5 HP and the propeller shaft running at 1050RPM. It is impossible to obtain such fuel savings with a 36 ft^2 wing-sail from wind propulsion effort alone. Therefore the energy saving from smoother travel through the seas accounts for the difference. This has also been observed by the Japanese in comparing identical vessels, where one was equipped with wind-assist sytem and the other ship was identical, but without sails.

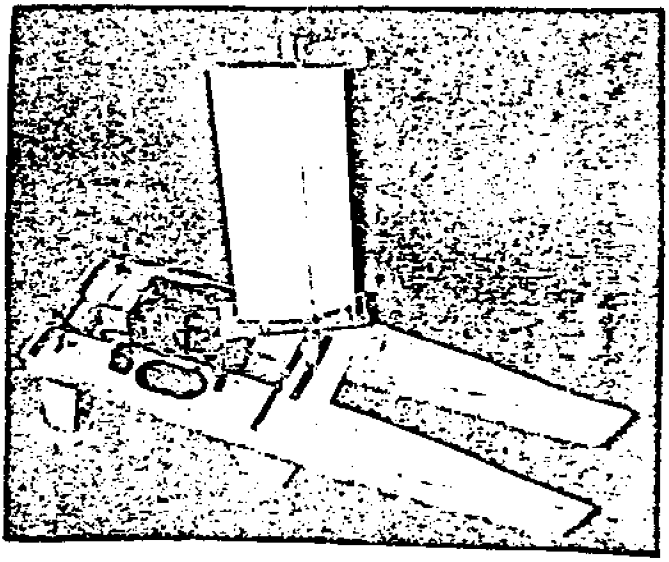


Figure 1.

3 foot long wing-sail equipped test model

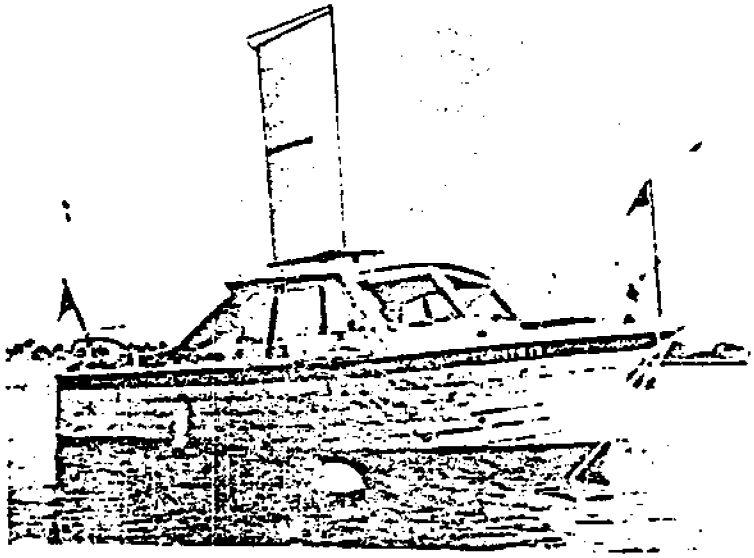


Figure 2.

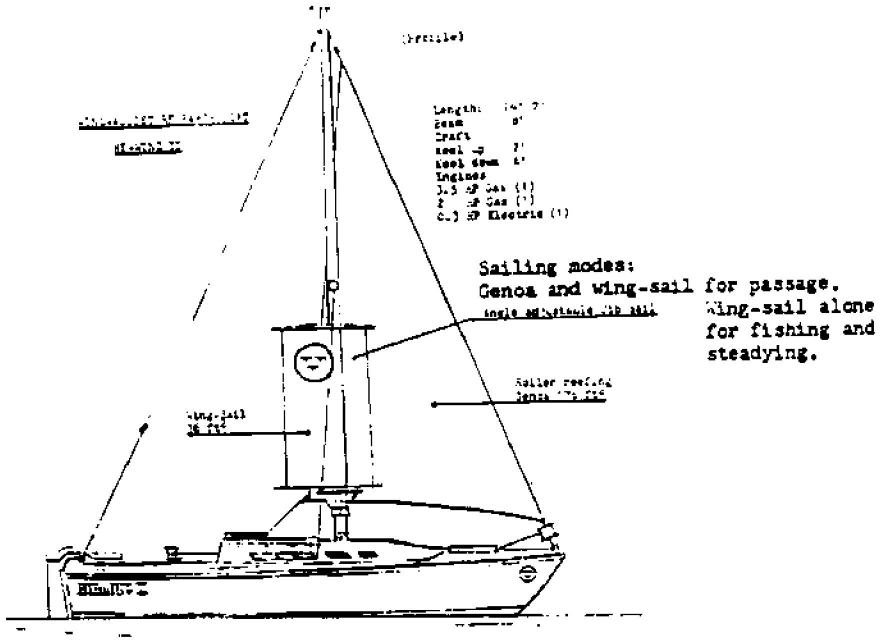
24 foot Sea Skiff equipped with "Flexi-Sail" wing-sail.

After tests with the wing-sail and also "Twin-Rotor" systems on the boat shown in figure 2, I have developed the approach for commercial fishing vessels and other smaller vessels, to use a combination of furled soft sails and a permanent wing-sail system.

For transport to and from fishing grounds both the Genoa side sail and the wing-sail system would be used. A conventional sail boat will sail as fast with the Genoa and the wing-sail as is possible with the standard main sail and jib of about 10% larger sail area.

Figure 3 shows a 25 foot test boat with which I will conduct tests during the summer and fall 1984 to determine the interaction between the wind-assist system and the side sail in various combinations with and without engine power.

In figure 4, I show the relative position between the Genoa and the wing-sail system. The air pressure decreases in three steps. The highest pressure is on the windward side of the wing- sail and the air pressure is further reduced on the lee side of the wing-sail and lowest on the lee side of the Genoa.



The coefficient $C = \frac{SA}{\sqrt[3]{D^2}}$ is indicative of propulsive capacity and Sail area SA in ratio to displacement D.

For the wing sail C is [illegible], which is low, but average for wind-assist systems.

For the Genoa C is 16, which is average for sail boat

For Genoa and WS C is 19, which is relatively high

Figure 2.

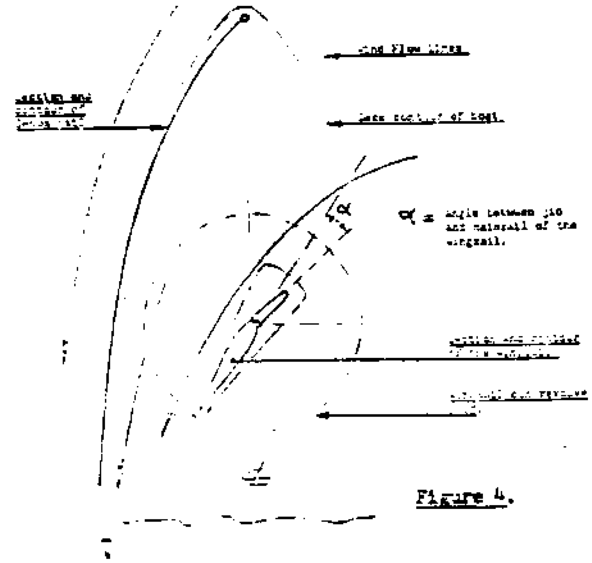


Figure 4.

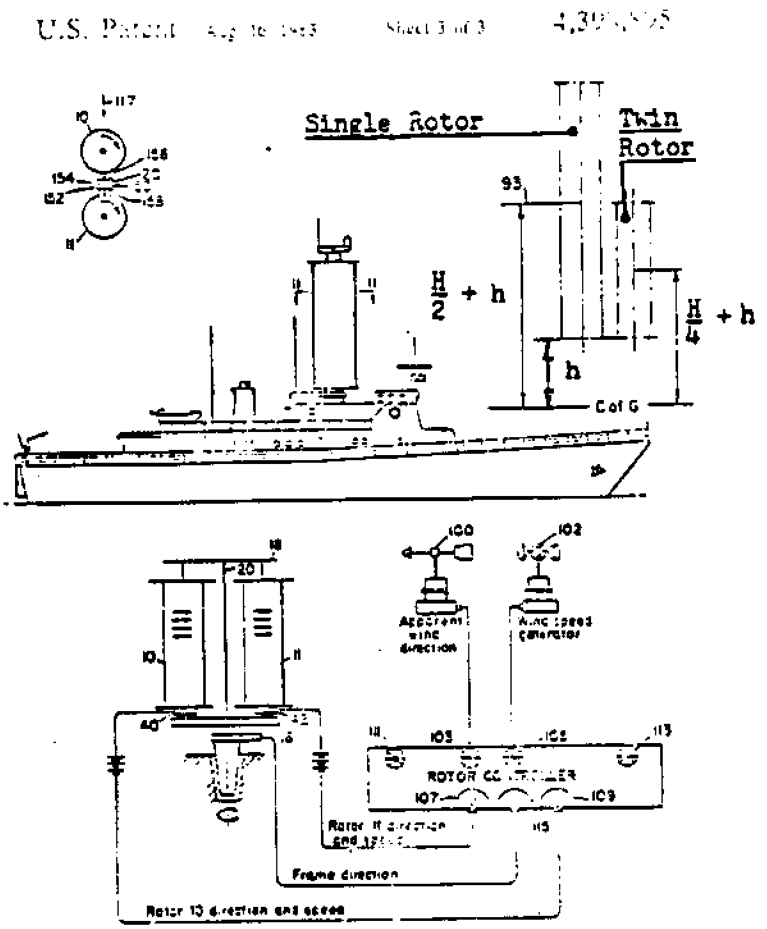


Figure 5.

This testboat is equipped with a swing keel and can be operated as a motor sailer with the keel up and as a full rigged sailing ship with the keel down.

One should also keep in mind that for commercial use motor sailing is safe and makes sense. 3 HP of engine power added to 210 ft^2 of sail is equal to a sail area of some 330 ft^2 for this test boat. With 330 ft^2 of sails this boat is unsafe in winds over 8 m/s, but would be safe and fast with 3 HP or more plus the wing-sail and the Genoa, which will furl as determined by wind strength. The side drift is also reduced by the addition of engine power.

A Fisheries Inspection Vessel of 150 feet length shown in figure 5 is large enough to take advantage of the reduced overall size of the "Twin-Rotor" system, which I have also tested on the Sea Skiff shown in figure 2.

In conclusion today I want to mention that we have seen many different approaches to wind-assist systems. It is apparent that quite a few researchers work independently without much knowledge of each other.

If we all work together- Researchers, Designers, Operators, Financiers and Fishermen in at least exchange information, we will be able to reduce net cost of the catches.

The upcoming WIND-TECH meeting in 1985 in Southampton, England, is a step in the right direction.

PART VI

INTERNATIONAL AND SMALL-SCALE FISHERIES

COMPARISON OF OPERATIONAL ECONOMICS FOR VESSELS WORKING FROM THREE WEST AFRICAN PORTS

By

John C. Sainsbury
Florida Institute of Technology

Abstract

It has not been uncommon among some developers to assume that a vessel which is profitable in a fishery, when working from a particular country and port, will prove equally successful when operated from a different country or port. There can however be a considerable difference in operating costs for the same vessel when working under such a situation, and this difference can have a marked effect on the unit cost of landing a particular species. Predicted costs for a similar vessel working from three West African ports are used to provide an example of how differences in operating, crew, and investment expenses lead to considerable change in landed unit cost of a directed fishery.

Introduction

Operational economics is always important, and often the critical factor determining profitability of a fishing vessel's operation. In Developing Countries both operating costs and earnings from catch sales can vary widely as a result of local conditions, yet it is not uncommon for developers to assume that a vessel which has proved profitable in a fishery when working from a particular country will be equally successful when based elsewhere.

While considerations other than pure profitability may be involved. e.g. the need to provide for fish protein needs of a population, it is obviously necessary that any vessel selected be appropriate for the facilities and services available locally, and have the ability to land fish at an acceptable cost within the particular situation. In some countries the need is critical that this be achieved at lowest possible cost with the least possible drain on foreign exchange needs, or in order to boost foreign currency resources.

The differences which can arise, even among neighbouring countries within the same area and working on essentially the same fish stocks, can be demonstrated by considering the West African countries of Senegal, The Gambia and Guinea Bissau, see Fig.1, (Fig.1.7.1 of Reference 1) and the case of a similar vessel working out of their principal fishing ports of Dakar, Banjul and Bissau.

Summary of Country Fisheries Situations

As shown in Fig.1, the three ports are situated between 11 deg. to 15 deg. North and 16-40 deg. to 18-30 West and have direct access to the ground fisheries of the West African continental shelf and the pelagic fisheries offshore.

Senegal is a net fish exporting nation, (mainly high value species to Europe), Ref.1, with the industrialized sector consisting of over 150 trawlers, shrimpers, purse seiners, gillnetters and handliners (Ref.2). A major new (and expandable) fishing port at Dakar, is well situated, and is a center for all services needed by a sophisticated fishing fleet, with daily air service to European, U.S., African, South American and Eastern destinations, and worldwide container services by sea. Both Senegalese and Foreign vessels (under license) land in Dakar, and there is an energetic private sector involvement in the industry, with a free market situation prevailing. Fishing Agreements are in force which allow Senegalese vessels to fish waters of The Gambia and Guinea Bissau.

The Gambia (Ref.1) is situated around a large river, with Senegal to the North, South and East. The country has virtually no industrialized fishing sector; several foreign purse seiners land fish for export to Ghana, and a local private company is commencing the use of vessels larger than the traditional canoes to further develop a successful export trade to Europe. The port of Banjul is conveniently situated for direct access to the offshore fishing grounds, with facilities for landing fish available; ice and processing plant capacity is not fully utilized, vessel maintenance and repair facilities are situated in the Port, with other services and supplies available locally or from Dakar (30 mins. by air, and some 2/3 hours by road). Private Sector involvement in the industry is developing, and public sector declining, with a free market situation prevailing. A Reciprocal Fishing agreement is in force allowing access to Senegalese waters.

Bissau, the principal port of Guinea Bissau (Ref.1) is situated a short distance up the mouth of a large river, and an offshore group of islands must be cleared in order to reach trawling grounds. Principal fish landings from the Industrialized Sector arise from a mixed enterprise company (with USSR) "Estrello do Mar" which operates several vessels under the country's flag. Other mixed companies are no longer operating at sea, although a small private sector company is preparing to expand to offshore fishing. Fish unloading facilities are available and an extensive new freezing/cold store facility is complete. Vessel maintenance and service facilities are available from both the public (military) and private sector (restricted) although specialized services and supplies must be imported. There are daily flight connections by propeller aircraft to Dakar and several times a week by jet to Portugal and USSR. Fish prices are established by the Government as are the prices of fuel (rationed) and other supplies (all imported) which are often difficult to obtain or unavailable. The presence of offshore oil resources is expected to ease both fuel and import problems in due course. An extensive series of fishing agreements are in effect with some 15 countries including Senegal, EEC, USSR, Algeria and Portugal; private sector activities are becoming important in many areas of the economy.

Components of the Total Cost/Ton of Landing a Particular Species :

In order to make a realistic cost comparison it is necessary to have data available for the operation of the same vessel type from the three ports of Dakar, Banjul and Bissau. Reference 1 provides estimated figures for a 78 foot vessel to work the stocks of Balistes (Trigger Fish) which have recently developed within the 200 mile zones of West Africa, and which appear to be available almost year round from these ports (see Fig.1). Typical catch rates obtained by the Norwegian Research Vessel R/V Fridtjof Nansen are also shown in Fig. 1; these, together with data from Reference 5 led to catch rates for the economic analyes contained in Reference 1 being set between 0.5 - 1.25 tons per fishing hour.

<u>Projected Landings</u> :The analysis also assumed a total capacity of 60 tons for the vessel, and that fishing continued until full. The number of days fishing per year and per trip therefore varied with catch rate, with the total days at sea (Fishing and on passage etc.) set as 225 per year for vessels working from Bissau and Banjul, and 240 days per year from Dakar; this difference was to allow for increased down time and more difficult working conditions anticipated from those ports. A summary of the projected landings is shown in Table 1 (extracted from Table 1.8.1 of Ref.1), which reflects the difference in steaming time and hence trip length necessary to land 60 tons/trip.

Data from Reference 5 indicated that when undertaking a directed fishery for Balistes, high value food fish aproximated 8% of the catch, and this was included in the analysis.

<u>Vessel Operating Costs</u> :In calculating vessel operating costs, both fixed and variable costs were added to give the total annual operating cost. The following were included:

Fuel: consumption estimated from Fig.2, (compiled from Average fuel consumption figures given in Reference 3); Prices: Bissau: $1.90/US gal; Banjul: $1.22/USgal; Dakar: $0.77/USgal.

Oils: 8% of fuel cost (ref.3)

Gear repair & Replacement: Reference 3 indicates an average cost for West African trawlers as $57,000 for an average of 180 fishing days per year.This figure was adjusted for number of fishing days per year, to give Assumed Cost = $57,000 x No. of Fish.days/180. As no operations are presently taking place from Banjul and Bissau, no reliable data was available, and costs for vessels working from those ports was assumed to follow the above formulation.

Ice: (If used) based on a 1:1 Fish:ice ratio plus 20% allowance for loss. (Ref.3). Prices: Bissau $50/mt; Banjul $21.20/mt; Dakar: $40/mt.

Insurance: Taken as 3.5% of investment cost (Ref.3).

Fishing License: Varies with country: Bissau: not required for G-B Registry. Banjul: $24/grt vessels over 400 hp, $12/grt vessels less than 400 hp. Dakar: $25/grt.

Port Taxes: Varies with Port; Bissau: $1 + $12.75/mt landed; Banjul: private mooring $6/month, freshwater $5.20/thousand gals.(25000gals/yr assumed); Dakar: 1% value of landed catch.

Maintenance and Repair: For new vessel: 5.5% of investment cost per year (Ref.3). For used vessel 3.5% of investment cost per year. (Based on data of Ref.3).

Crew Expenses: Expenses for local crew usually include food and transportation allowance. Bissau: $1/day at sea plus 24 kilos of rice/month/person. Banjul: total of $12000/yr allowed for local crew costs; Dakar: $12,000/yr for local crew. Also included here is the cost of housing and transportation of expatriate skipper, based on situation prevailing in each port.

Crew Costs : Crew requirements vary according to regulations and the practices common in each port. An expatriate skipper was assumed in all cases, with an annual pay of $3,400 per month plus $6/ton of fish landed. Pay of local crew was based on a scale and bonus common or recommended by the industry in each port. Bissau: Engineer (expatriate) $20000/yr + bonus, mate $1850/yr + bonus, Assistant Engineer $1860/yr + bonus, Cook $1650/yr + bonus, Deckhands (5) 5 x $1500/yr + bonus; Bonus shared by local crew: 1% of catch value over 10 tons/day. Banjul: Engineer $3360/yr + bonus, Mate $2640/yr + bonus, Assistant Engineer $1680/yr + bonus, Cook $1200/yr + bonus, Deckhands (4) $3840/yr + bonus; bonus shared by local crew: $4/ton. Dakar: Engineer $3600/yr + bonus, Mate $3200/yr + bonus, Assistant Engineer $2400 + bonus, Cook $2200/yr + bonus, Deckhands (4) $8000/yr + bonus; Bonus: $10 per ton.

Administration Cost : The average administration costs for West African vessels is approximately 4% of total operating and crew costs (Ref.3).

Vessels and Investment Cost : The vessels used in the analysis were assumed to be of US origin and representative of those suitable for the fishing operations involved. A number of similar vessels have been sold to West African Interests during the past five years, and Reference 4 provides typical first cost for vessels equipped for shrimp trawling and stern trawling on the African West Coast: first cost (investment cost) for such vessels is summarised in Fig.3 (for the year 1982). As a basis for the cost calculations the principal particulars of the vessels were assumed as: Length 78ft., HP 415, GRT appr. 84. In order to illustrate the effect of using Ice versus RSW storage, and the difference between new and used vessels a range of vessel investment costs were used:

RSW storage:	New vessel	$675,000
	Used vessel	$474,000
Ice storage:	New vessel	$600,000
	Used vessel	$400,000

In each case, financing arrangements typically available in the United States were assumed: one third down payment with fifteen year amortization at 15% interest on the annual declining balance. Capital costs for the first year of operation were then calculated as:

Investment cost:	First year int. + amortization:
675,000	$322,500
600,000	$286,667
475,000	$226,944
400,000	$191,111

Note that first year capital costs only were included, so that second and subsequent years would reduce considerably. No attempt was made to carry the analysis over years beyond the first.

Earnings from High Value Food Fish By-catch : As mentioned previously, approximately 8% of the total catch for trawlers pursuing a directed Balistes fishery consists of high value food fish which may be used for export earnings or for internal consumption in the country. The earnings from sale of these fish was considered to act as a credit, hence reducing the actual cost of landing the Balistes. Applicable fish prices for the various ports were found to be:

Bissau (official maximum prices): Class 1 fish: $0.75/kg ex-boat. $1.25/kg/ retail; Class 2 fish: $0.56/kg ex-boat, $1/kg. retail; Class 3 fish: $0.38/kg ex-boat, $0.75/kg retail: Class 4 fish: $0.34/kg ex-boat, $0.56/kg retail. For the analysis an average figure of $891/tonne was assumed as potential retail earnings to be set as a credit against cost of landing Balistes.

Banjul and Dakar: Assumed Average price received for exported food fish was $720/mtonne. (Refs. 2 & 3)

Cost of Landing Balistes : The total cost of fishing for any particular combination of vessel and port was therefore given by:

Total Cost = Vessel operating cost + crew cost + Administration cost + capital cost.

The actual cost of landing Balistes was then calculated as:

Total cost of Balistes = Total Cost - Value of Other Fish

From this the unit cost of landing Balistes is calculated as $/mtonne (assumed equivalent to long ton), and in cents/lb..

Cost Comparison

Reference 1 provides tables showing estimated landed cost of Balistes for the range of catch rates selected and for the various vessel types and investment costs outlined previously. Table 2 shows a comparison of landed cost/mtonne for a used vessel utilizing RSW holding, when working from the three ports, and demonstrates the development of the figures for landed cost/mtonne. Table 2 is extracted from data presented in Tables II.A.5.2, II.B.4.2, and II.C.4.2 of Reference 1.

As may be seen from Table 2, there is a spread of more than 22% in total operating cost between Bissau, the most expensive, and Dakar, the least expensive. When the value of local high value catch is included the spread rises to some 24%, and in terms of Cost in $/tonne of Balistes landed, the difference rises further to 32% although the costs for both Banjul and Dakar are almost identical. It is interesting to note that the difference in operating expenses (other than crew and capital costs) is 54%.

The Effect of catch rate on the landed cost can be seen from Fig.4, which was compiled using data from the same Tables, Reference 1. The increase in landed cost per unit weight increases sharply as the catch rate decreases, although the percentage difference between costs decreases. As the catch rate increases the cost per unit weight landed tends towards a constant value.

The effect of applying new or used vessels, together with the method of hold storage can be seen from Fig.5 which shows the results for the port of Banjul (Fig.II.B.4.6 of Reference 1). In general, the costs for both methods of storage approximate one another, with RSW having an advantage above a catch rate of about 0.6 tons/hour, which reaches about 10% at a catch rate of 1.25 tons/hr. As might be expected, operation of a brand new vessel results in increased capital costs; in practice, of course, it may well be possible to close the gap between the new and used vessels through judicious use of financial manipulation.

The effect of vessel investment cost for the Port of Banjul is shown in Fig.6 (Fig.II.B.4.7 of Reference 1) for a catch rate of 1.25 tons/hr. Under the capital cost situation used in Reference 1, it may be seen that an RSW vessel of approximately $150,000 greater initial investment is predicted to produce Balistes at the same unit cost as an Ice storage vessel.

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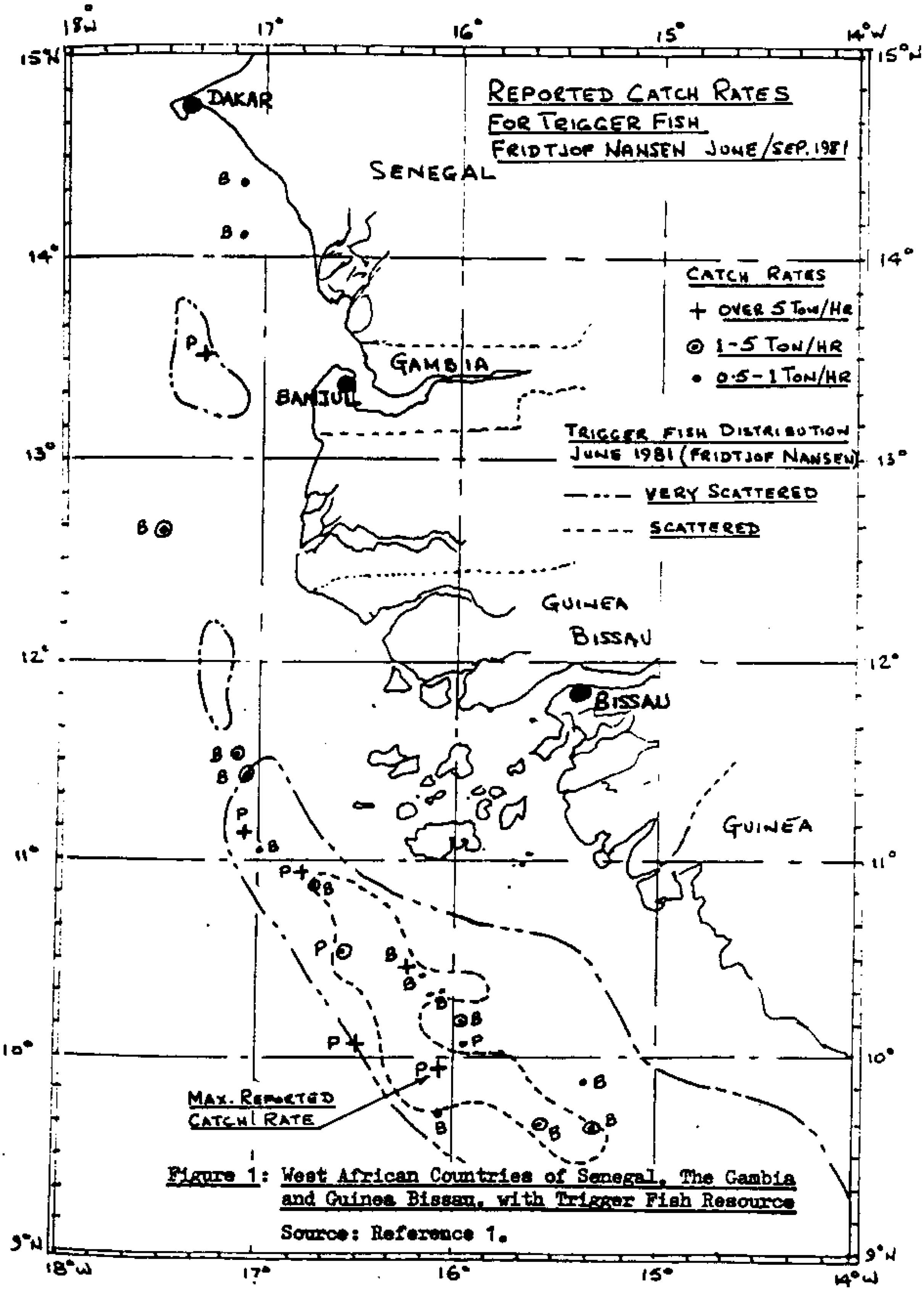

Figure 1: West African Countries of Senegal, The Gambia and Guinea Bissau, with Trigger Fish Resource

Source: Reference 1.

Figure 1

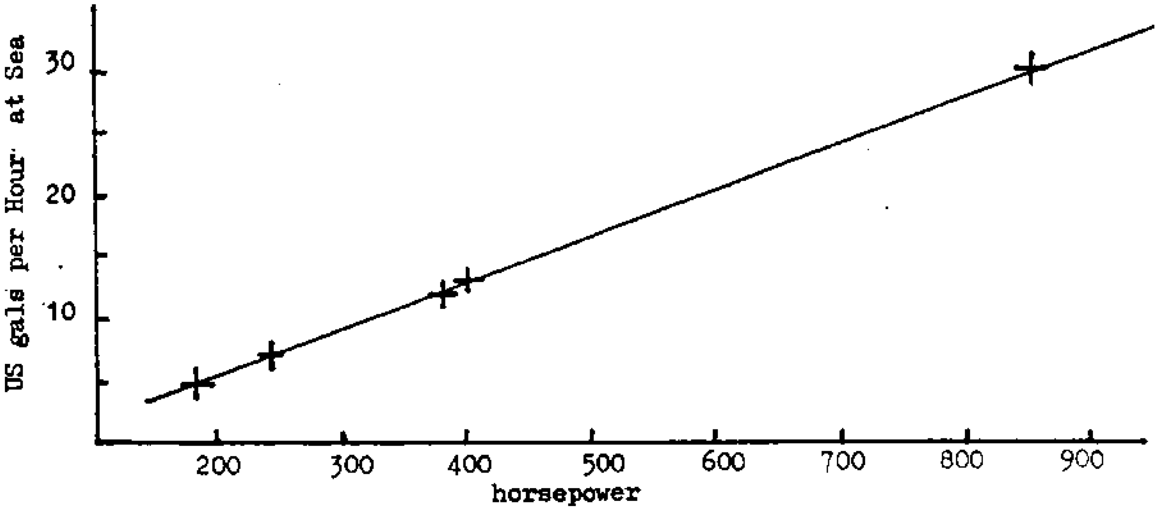

Figure 2: Average Fuel Consumption For West African Trawlers

Compiled From Data Included In "Some Observations on Formulation of Alternative Strategies for Development of Marine Fisheries" (Ref.3)

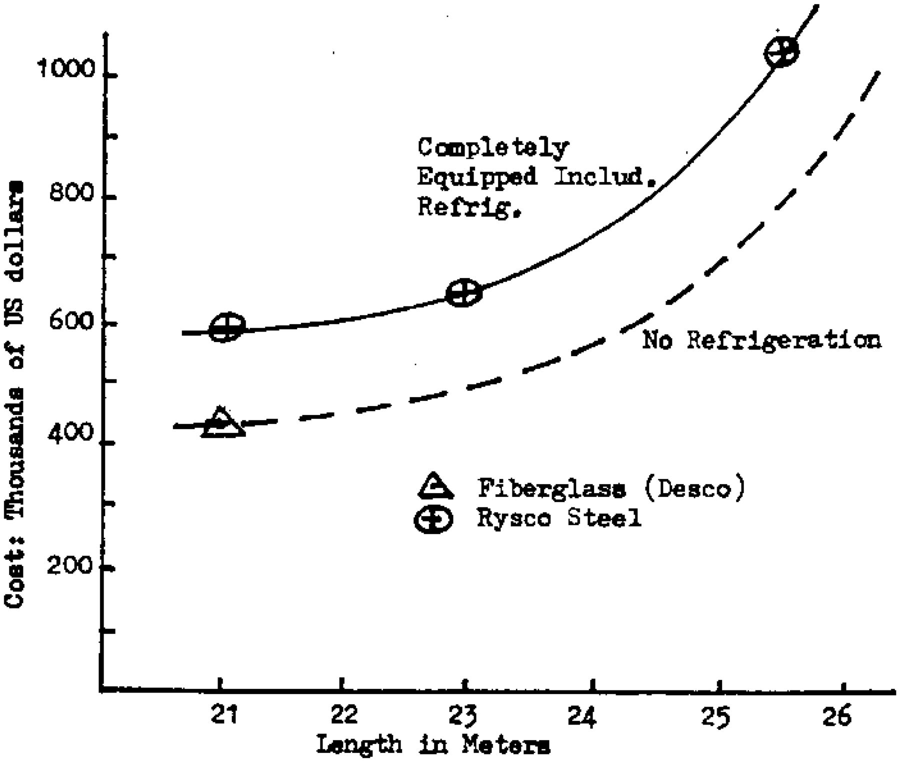

Figure 3: Assumed First Cost for Vessels Equipped For Shrimp Trawling and Stern Trawling, African West Coast (1982)

Compiled From data Included in "Bioeconomic Analysis of a CECAF Shrimp Fishery" (Ref.4)

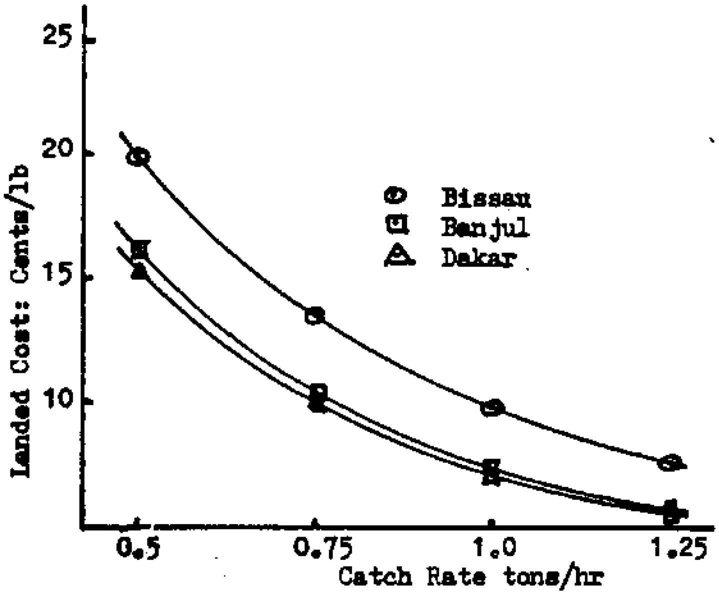

Figure 4: Effect of Catch Rate on Landed Cost

Compiled from Data included in Ref.1

Table 1 : Projected Landings for Economic Analysis

Port	Catch Rate Tons/hr.	Catch tons/ 12hr.day	Fishing days per year	Trips per yr	Landings tons/yr
Bissau	1.25	15	180	45	2700
	1.00	12	187.5	37.5	2250
	0.75	9	195.6	29.4	1760.4
	0.5	6	204.5	20.5	1227
Banjul	1.25	15	200	50	3000
	1.00	12	204.5	40.9	2454
	0.75	9	209.3	31.4	1883.7
	0.5	6	214.3	21.4	1285.8
Dakar	1.25	15	192	48	2880
	1.00	12	200	40	2400
	0.75	9	208.7	31.3	1878.3
	0.5	6	218.2	21.8	1309.2

Table 2 : Comparison of Landed Cost/mtonne of Balistes for Used RSW vessel working from Bissau, Banjul and Dakar at catch rate of 1 tonne/fishing hr. Vessel Cost: $475,000

Port	Bissau	Banjul	Dakar
Catch rate; tonnes/hr	1.0	1.0	1.0
Trips/year	37.5	40.9	40
days/trip	6	5.5	6
days at sea/year	225	225	240
days fishing/year	187.5	204.5	200
catch: Balistes:tons/yr	2250	2454	2400
Other:tons/yr(8%)	180	196.3	192
total:tons/yr	2430	2650	2592
Ice: tons/yr	-	-	-
cost:$/yr	-	-	-
Fuel: Gal/hr	15	15	15
Gal/yr	81,000	81,000	86,400
Cost $/yr	153,900	98,820	66,528
Oils cost $/yr	12,636	7,906	5,322
Gear cost $/yr	59,375	58,283	63,333
Insurance $/yr	16,625	16,625	16,625
Fishing License $/yr	-	2,016	2,100
Port taxes $/yr	30,911	171	5,400
Maint. & Repair $/yr	40,375	40,375	40,375
Crew Expenses $/yr	12,000	12,000	12,000
Total Oper. Cost $/yr	325,822	236,196	211,683
Crew Cost $/yr	85,979	80,023	86,120
Total: Oper. + crew	411,801	316,219	296,803
Admin. cost $/yr	16,472	12,649	11,912
Capital cost: 1st.yr.	226,944	226,944	226,944
Total Cost $/yr	655,217	555,812	536,658
Value of by-catch $/yr	160,380	141,336	138,280
Total Cost: Balistes	494,837	414,476	398,379
cost:Balistes $/mtonne	219.9	168.9	166.0
Cents/lb.	9.8	7.5	7.4

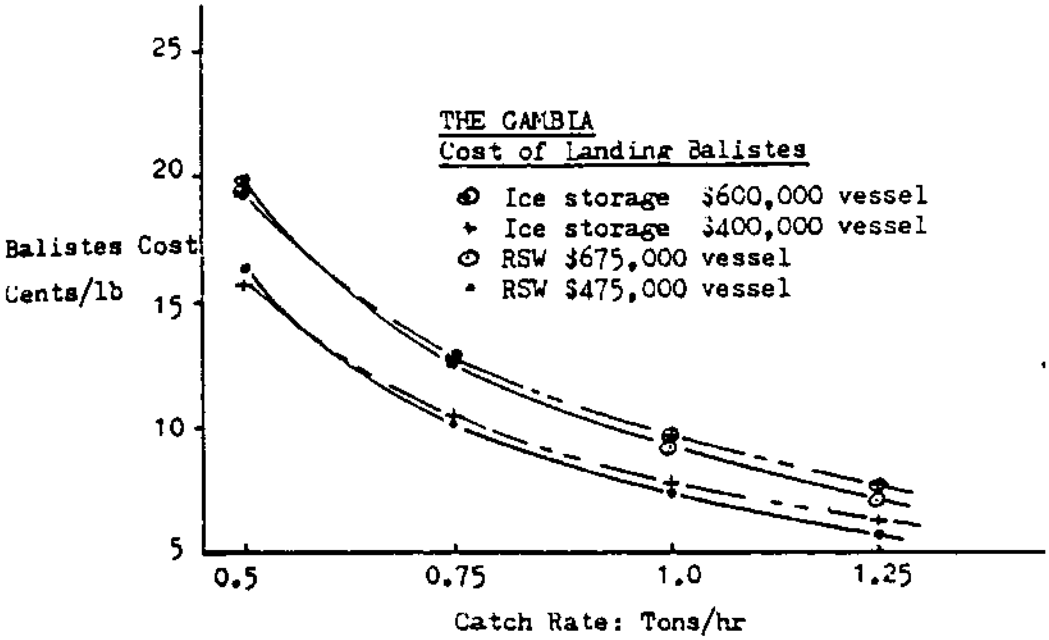

Figure 5: Effect of Storage and New/Used Vessel on Landed Unit Cost: Port of Banjul

Source: Reference 1.

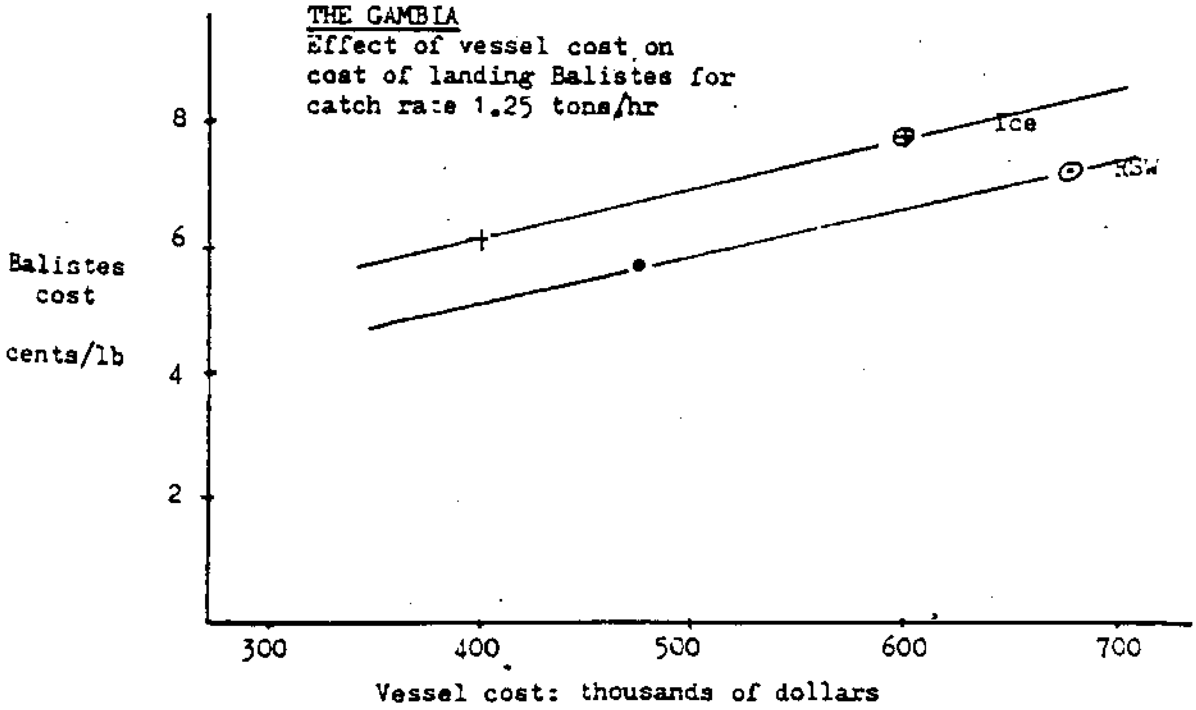

Figure 6: Effect of Vessel Investment Cost on Landed Unit Cost: Port of Banjul

Source: Reference 1.

APPROPRIATE DESIGNS

FOR

FISHING CRAFT IN BANGLADESH

by R.G. MacAlister, C. Eng. Mrina

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1. SYNOPSIS

In 1982, MacAlister Elliott and Partners were asked by the Bangladesh Krishi Bank (BKB) to design and organise the construction of a number of small fishing boats for the Asian Development Bank (ADB) Bangladesh Fisheries Credit Project (BAN 420). This paper describes the design thinking for a series of gillnetters and trawlers to be built in local yards with local materials and facilities but optimising vessel efficiency and safety.

2. BACKGROUND

Bangladesh is a deltaic country with a population of some 100 million people. The marine fisheries have been largely confined to coastal activities but in recent years, efforts have been made to exploit the deeper Bay of Bengal waters within their Exclusive Economic Zone. The traditional fisheries for Hilsa, jewfish, pomfret and Bombay Duck, as well as sharks, skates and rays, are facing increasing competition from high value catches of shrimp for export. Although the value of annual catches has risen by as much as 26% per annum since independence, in 1972, landings of fish for local consumption have dropped over the years. Various projects are in hand to try and increase domestic landings, including the A.D.B. project with which we are involved.

The A.D.B. Bangladesh Fisheries Credit project aims to provide 200 mechanised wooden gillnetters and 100 small wooden trawlers plus associated back up and distribution infrastructure such as landing places, workshops, ice plants, cold stores and refrigerated trucks. The project is being coordinated by the counterpart agency, the Bangladesh Krishi Bank.

In 1982, MacAlister Elliott and Partners were asked by the A.D.B. to review some fishing boat designs which had been prepared in Bangladesh for the project. As is so often the case when drawings are required for an activity which has been carried on for generations by skill and experience without drawings, the results were sadly lacking and did not define realistic or safe vessels.

MacAlister Elliott and Partners were consequently retained by the Bangladesh Krishi Bank to research and prepare suitable designs, to identify appropriate yards and to provide the necessary master boatbuilder to guide construction.

In December 1982, the writer visited Bangladesh to glean information for the design process. This involved understanding the fishermen's preferences, the fishing methods and conditions, the facilities and materials available and the skill and techniques of traditional boatbuilders.

3. TRADITIONAL BOATBUILDING

With the myriad of estuaries, rivers, canals and waterways in Bangladesh, the number and variety of vessels is enormous. Most of the inshore fishing however, is carried out by Balams or similar craft. Gill netting and drift netting has been practiced in the Bay of Bengal for generations, and the Balams were traditionally sailed, drifted or rowed. Their fine lines and low profile are well suited to gill netting and their shallow forefoot and keeless section allows beaching and settling in the mud which is prevalent along the coast.

The building method is simple and requires few structures or premises. The Balam is a well developed traditional boat.

Figure 1. Gillnetter in Cox's Bazaar.

For operational and logistical reasons, the project concentrated in S.E. Bangladesh and time was spent in that area studying building methods.

A number of yards in Chittagong and Cox's Bazaar were visited, both to assess competence and facilities and to select the yards which would be asked to tender. There are perhaps 100 more or less established yards in the area and 50 or so had expressed interest in participating.

Yards vary from the Bangladesh Fisheries Development Corporation (BFDC) (ex Danida) yard in Chittagong with extensive space, plant and equipment and material stores, to informal organisations seemingly without staff or premises which mobilise to build boats if asked. Generally, yards consist of a small area of land with access to water with one or more boats being built or repaired with hand tools. Some have electricity but few have any wood working equipment. Many have nevertheless been building durable boats for generations. Many boats are also built by fishermen employing itinerant carpenters in their own villages.

The design and building of fishing boats has remained virtually unchanged over generations except for the introduction of mechanisation. The Balam type boats vary in length from about 10 metres to 20 metres overall.

Construction methods vary little up and down the coast but represent a regional boatbuilding method not found elsewhere. The hull is started by laying down a 'keel' or 'hog' which is a wide, flat board running from stem to transom. This is propped at the ends and fastened down in the middle to form a pronounced rocker. A substantial stem is attached to the 'hog' with a single knee, and a flat planked transom is supported at the aft end. Rudimentary sawn floors which extend to the turn of the bilge are attached to the 'hog'.

Figure 2. Gillnetter under construction showing floors and transom.

The garboards and bottom are planked with boards about 150 mm to 175 mm wide, the garboards being butted directly to the 'hog' but rabbetted into the stem. Fastening is usually by wire nails and some long bolts.

Thus the bottom structure of the boat is completed with few supports, the shape being defined by the curve of the 'hog' and the floors faired by eye and experience. No structures or building or machines have been necessary, though the planks nowadays are generally machined in a saw mill.

Frames for the topsides are next attached to the floors and nailed or bolted in place rather haphazardly. Planking continues, the boards being wedged together without caulking bevels and nailed to the frames. Planks are kept plane with each other by edge nailing into specially chiselled slots. Thus the canoe body of the hull is complete.

At this stage, there is no deadwood/keel. This is cut from solid or fabricated and attached to the canoe body with long bolts or 'U' bolts. This unusual procedure may have been adequate before mechanisation but is a source of weakness today.

The hulls are decked with a small camber and a simple accommodation deckhouse built over the engine room. Finally, the hull and decks are payed and caulked, using cotton and pitch.

Below the waterline, the hulls are burned and tarred to restrict activity of marine borers. In service, well maintained boats are hauled, burned and tarred, about every three months.

Balam type gillnetters are generally reported to last about 5 or 6 years, during which time there will have been some replacement of planking and steel fastenings.

4. MATERIALS

Timber.

Most of the timber available for boatbuilding in the region comes from the hill country behind Chittagong and Cox's Bazaar. Generally, trees are felled and floated down to saw mills in the towns. In Chittagong, higher usage of timber for various trades has exploited most of the larger trees and boards seldom exceed 10 metres in length. In Cox's Bazaar, large trees can still be felled and due to their size, are pit sawn where they fall. At the yards in this region, boats are built using some planks over 20 metres long.

The most common boatbuilding timber in the region is Gurjun (Dipterocarpus), also known as Keruing and Yang. This teak-like wood is moderately coarse grained, has some resistance to rot and marine borers, and has satisfactory cutting and working properties. It is an acceptable, though not ideal, boatbuilding timber. Other timbers such as Jarul are available in smaller quantities.

Toredo worm is a major hazard in these waters and can reduce a hull to scrap if not protected.

Figure 3. Toredo damage to a section of stem.

Gurjun is moderately responsive to CCA (Copper Chrome Arsentate) treatment to reduce attack by parasites. Ideally, this process needs to be carried out under pressure to obtain optimum penetration and protection. Only one plant, at the BFDC (Danida) yard, has pressure equipment but it is of insufficient capacity to treat timber for outside yards.

Other yards have various immersion tanks but the efficacy of the treatment is lower. Nevertheless, all immersed timber (keel, sternpost, underwater planking, etc.) should be treated as efficiently as possible, after shaping.

Fastenings.

All the yards with the exception of the BFDC yards, construct boats with steel wire nails and a few bolts. Only one yard was using cut boat nails and these were secondhand. The BFDC yard galvanises its own fastenings. There appears to be insufficient galvanising capacity in Bangladesh to allow an adequate supply of fastenings and the Project will have to rely on traditional methods for treating planking fastenings. It is however, considered essential that the main structural bolts are galvanised.

Fabricating and Machining.

There are numerous small workshops with fabricating and machining capabilities. Raw material is in short supply but most items can be made with ingenuity and the re-use of materials.

Paints and Caulking Materials.

Boats are caulked with a satisfactory pitch compound in adequate supply. All boats are tarred below the waterline and many above as well. A few boats have painted topsides and general purpose alkyd paint is available.

Fitting out Materials

There is no organised supply of chandlery or engineering supplies in Chittagong or Cox's Bazaar. Items can be obtained on an occasional basis but the suppliers of the main machinery and equipment must provide all necessary parts for complete installation.

5. CONCEPT OF THE NEW DESIGNS

This then was the background for the design and building of 200 gillnetters and 100 small wooden multi purpose trawlers; a region which since time began has built numerous boats on a casual basis and which has developed an efficient inshore 'Balam' type craft.

Since mechanisation, however, there have been few developments and the hull is no longer ideal. Mechanised boats go further offshore and encounter worse weather. The round sections and lack of keel on the Balams make the vessel tender and unstable. Consequently, much fishing time is lost in even moderately bad weather. The stresses and vibrations from the engine cause problems with the structure of the boat which is weak due to the keel and non-rabbetted garboard configuration. Even with the low horse power mechanised gillnetters, bailing and pumping is often a full time occupation.

There is little tradition of inshore trawling in Bangladesh and few small trawlers exist along the South East Coast.

These vessels will be bottom trawling in areas with strong currents and predominantly silt and mud bottom. This can be difficult and requires a versatile, powerful boat. Great strains can be put on the hull and gear if the trawl doors drop into the mud. The sea is calm much of the time, but dangerously rough during the monsoons, so safety and good sea keeping are essential.

The fishing grounds are about a days steaming from most of the bases and there were long discussions with potential owners about preferred vessel dimensions. There is a natural tendency to associate safety with length and there were many requests for craft out of all proportion to the task and potential catch, though due to the long rake of the stem and narrow waterlines, the volume visualised for a given length is much less than might be expected.

Eventually the following basic dimensions were selected as suiting both practical and personal requirements (clients requested imperial units):

Gillnetter:

L.O.A.	42'7"
Length on deck	37'6"
Length at datum W.L.	34'9"
Beam	11'0"
Beam at datum W.L.	9'9"
Draft to datum W.L.	3'5"
Moulded depth	5'1"
Fish hold capacity	160 cu. feet
Fuel oil tank capacity	60 gallons
Fresh water tank capacity	60 gallons
Displacement at half load	8 tons approx.
Displacement at 4'0" draft (level trim)	12 tons approx.
Main Engine	approx. 35 hp.

Trawler:

L.O.A.	51'02
Length on deck	48'9"
Length at datum W.L.	45'6"
Beam	14'0"
Beam at datum W.L.	12'6"
Draft to datum W.L.	6'0"
Moulded depth	6'9"
Fish hold	565 cu. feet
Fuel oil tank capacity	720 gallons
Fresh water tank capacity	90 gallons
Displacement at half load	23 tons approx.
Displacement at 6'6" draft	27 tons approx.
Main Engine	approx. 160 hp.

Gillnetter

The new gillnetters are designed to be built in the traditional yards, using available materials and to be operated by local crews. The design, therefore, shows a vessel of traditional appearance but designed to be a good, robust sea boat. The shallow forefoot has been retained for beaching, but a substantial keel forms the basis for the structure. This will, incidentally, improve the sailing ability of the hull and outlines of suitable rigs are included in the design.

The lines have been designed to minimise the curvature of the planking to the deadwood as this will be unfamiliar to the builders. It is nevertheless considered essential that the keel/deadwood is integrated into the structure.

Layout of holds, low windage accommodation, cooking and toilet facilities is traditional, although the hold insulation has been upgraded. The construction of the cantilevered heads has been left strictly to the builders. We are not sure what goes where.

The gillnetter has been designed for a crew of up to 8 people with fuel, water and ice for up to 7-day trips. The vessel will cruise at about 7 knots.

Trawler

Trawlers are marine tractors and although the installed horsepower in the project vessels is relatively modest, a substantial and adequately deep hull is required to carry the machinery and stern gear.

The design shows a fine vessel with similarities in appearance to fishing boats in the region and Bay of Bengal. The hull is deep enough for an efficient propeller, and will be dry and stable. The layout is as a stern trawler with a forward wheelhouse and mechanically driven winch. The 16 ton fish hold is sufficient for the maximum likely catch of fish and ice. Trawling in these waters can impose severe strains on trawl gallows so a strong gantry system has been included.

For some of the year, trawling is not economic and layout and topside heights have been maintained for gill netting. During this period, the gantry can be removed and a steadying sail may help the boat to lie to the nets.

Accommodation and tankage are sufficient for 9 people for up to 7-day trips. The trawler will cruise at about 8.5 knots.

6. CONSTRUCTION.

It would be simple to specify construction methods and materials of the highest standard. In practice, it is illogical to specify anything which cannot or will not be used. The designer must accept the fact that the hulls and much of the structure will be made of keruing, that the timber treatment will be rudimentary, the fastenings mostly wire nails and the cotton caulking driven into unbevelled seams. However much lofting, mould making and supervision the Master Boatbuilder does, the boats will tend to look the way the builder thinks they should.

The designs, drawn in our design office by Robert Brasted, take these realities into account. A proper keel structure is essential to provide adequate strength for a mechanised boat but a simple parallel keel is acceptable. Although they may not provide an ideal depth of rebate, they can be sawn, if necessary, in a pit without too much skill and with close frames and floors the garboard strake can be more than adequately fastened. Galvanised boat nails are not indigenous and the facilities for making them are very variable. It has been assumed that ordinary round wire nails will be used much of the time and in order to provide well fastened structures, clenched fastenings have been used in some areas, and threaded through fastenings only in major items of structure. The wire nails will be dipped in hot tar and driven wild in piloted holes.

Mechanical installations are basic with minimum electrics and mechanical drive to the trawler winch. Engine starting is manual on the gilnetters, and would be on the trawlers too if more manufacturers offered the option. With plenty of room and several crew at each end, it is amazing what size of engine can be started by hand.

These two designs define, we hope, strong safe vessels which will perform the functions well, can be built in the existing yards and with machinery which can be maintained in the somewhat rudimentary facilities available.

7. CONCLUSION.

The aim of this project has been to provide finance and technical assistance to enable the Bangladesh fishing and boatbuilding industries to expand their small mechanised fleet using local boatbuilders and local crews. Within this context, the technical assistance is endeavouring to ensure the best possible boats. Progress is not fast but procurement of machinery is under way and construction of the first batch of gillnetters should commence soon.

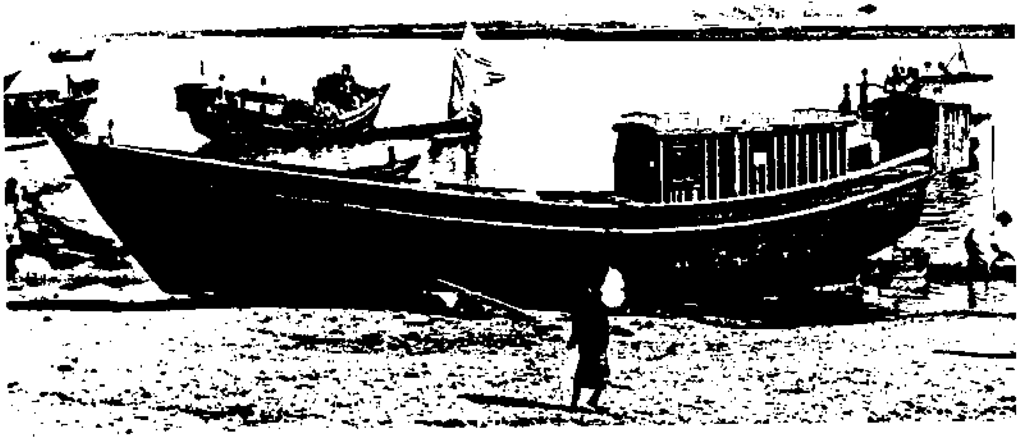

Figure 1. Gillnetter in Cox's Bazar

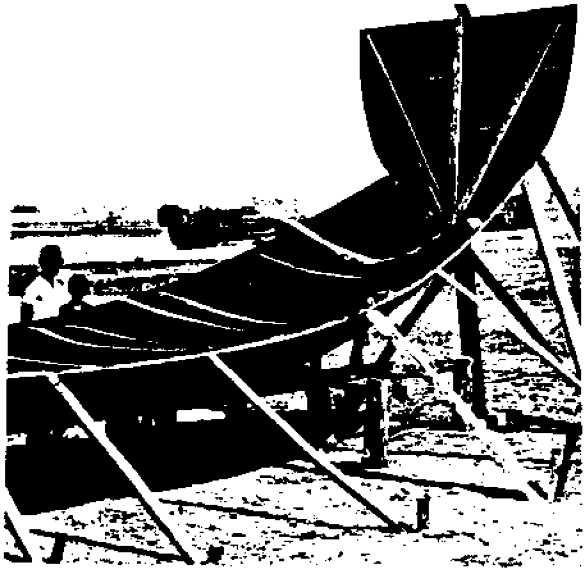

Figure 2. Gillnetter under construction showing floors and transom

Figure 3. Toredo damage to a section of stem.

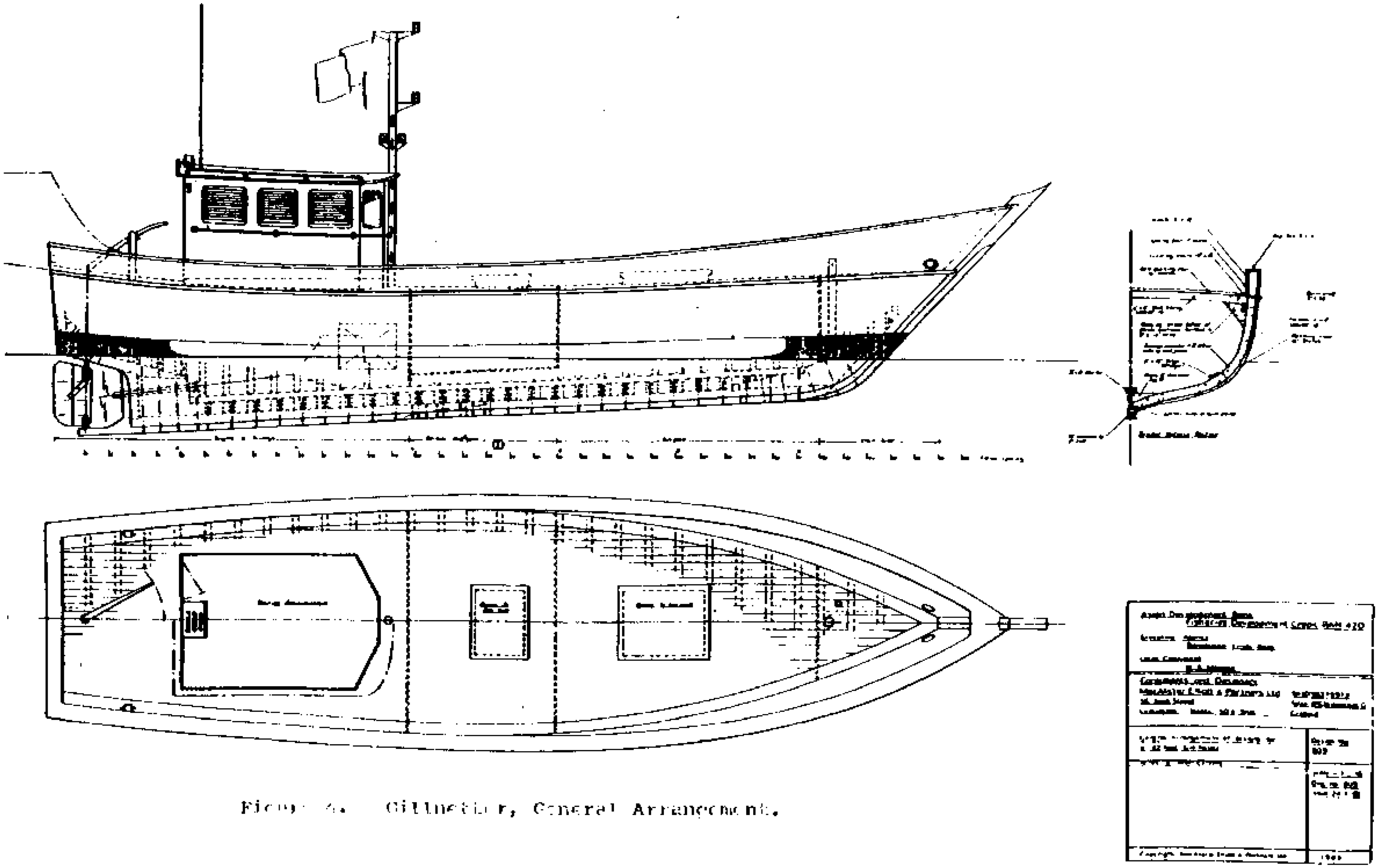

Figure 4. Gillnetter, General Arrangement.

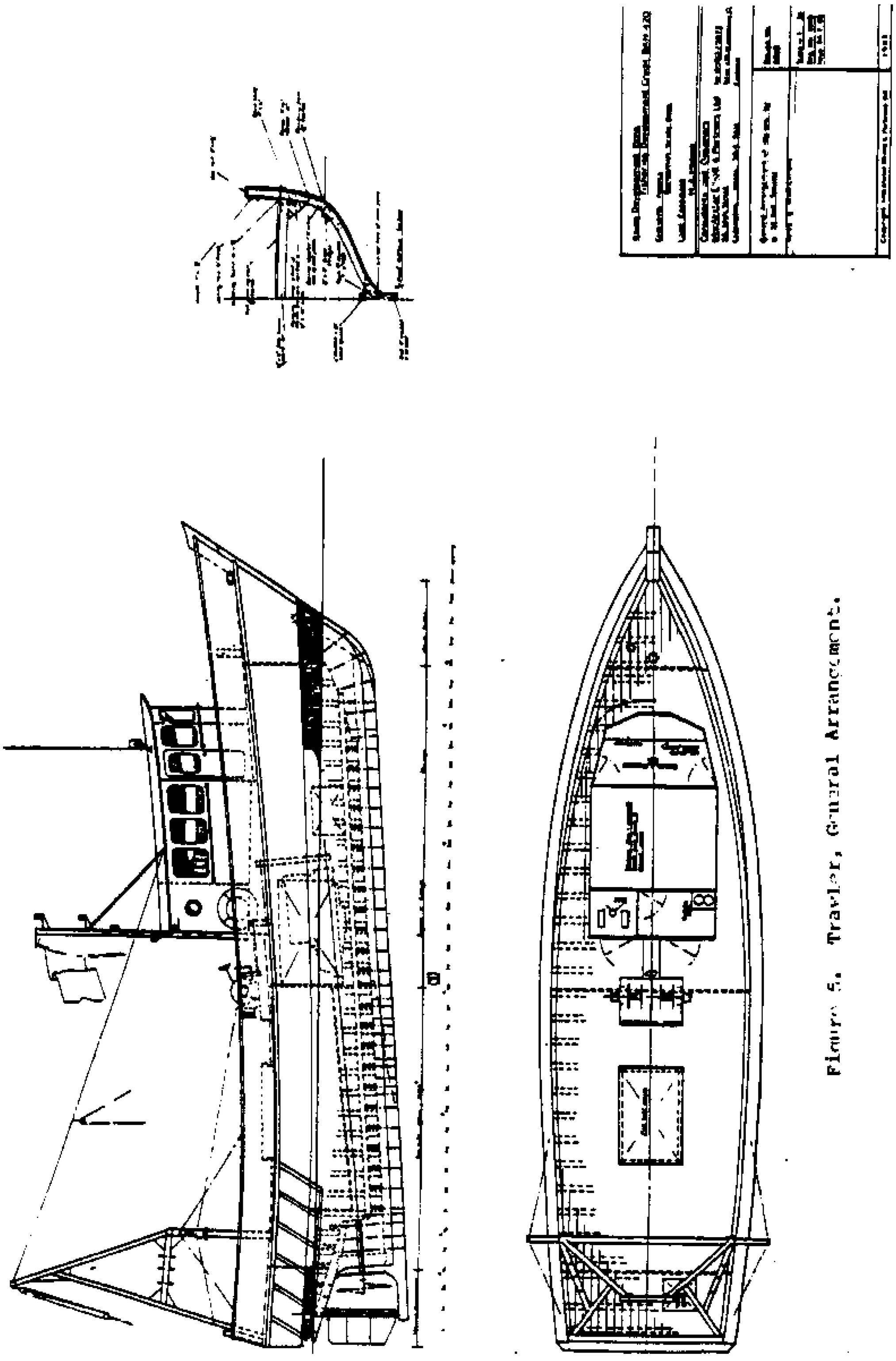

Figure 5. Trawler, General Arrangement.

PARTICULAR NEEDS OF FISHING VESSELS FOR USE IN THE DEVELOPING WORLD

Alejandro Acosta
Theophilus Brainerd
Bambang Priyono
Norbert Simmons
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Abstract

The fisheries of developing nations sometimes require vessels whose characteristics are quite different from those used in the industrialized countries. The panel assembled here is composed of representatives from East Asia, Latin America, West Africa, Southeast Asia, and the Caribbean. Each of these regions encompasses a tremendous variety of fisheries and physical conditions. For example, to discuss the requirements for fishing vessels for Southeast Asia may be like discussing the needs of fishing vessels for North America. Constraints in space and time permit only general overviews, and detailed description of any one fishery is impossible here. In some cases the focus will be on a particular country within the region, and in other cases the discussion may center on considerations which are generally common to an entire region.

Also present in many of these areas are foreign flag, distant water fishing vessels, often with on-board freezing or processing capacity. The following discussion will not include such operations, but will focus on the fishing operations of the nations within each region.

Many developing nations have recently established 200 mile Exclusive Economic Zones, and they are anxious to increase their benefits from the fisheries resources of these zones. Where industrial fleets have increased, small scale fisheries have sometimes been neglected. Nevertheless, small scale fisheries are often extremely important local sources of protein, employment, and income. The particular problems and considerations involved in vessel operations in developing areas often have a greater impact on small craft than on large vessels. For the above reasons, this discussion will focus primarily on the small scale fishing sector. There is endless debate on the definition of small scale fishing, and any definition will find many exceptions. For discussion purposes here, the terms small scale fishery and artisanal fishery are used interchangeably, and defined as those fisheries involving vessels of less than 15 meters in length, generally operating within 20 miles off the coast.

The versatility offered by multi-purpose, combination fishing vessels is a great advantage in many developing areas, if the cost of conversion to a different fishery is not prohibitive. Many developing nations have high unemployment, and in these areas, innovations which tend to decrease crew size might do more harm than good.

Many developing countries have a shortage of foreign exchange with which to buy imports. Where this causes high fuel prices or shortages, fuel efficiency and alternative energy sources such as sail should be important considerations. In the fisheries of developing areas, inboard motors are sometimes far more economical to operate and maintain than outboards.

Motors, spare parts, and repair facilities are sometimes lacking, and new technology should be as easy as possible to maintain and repair. Provision for long-term supply of spare parts is a must. Vessel construction maximizing the use of local materials and labor can sometimes ease problems with foreign exchange.

In many areas, adequate harbor facilities are lacking, and small craft must be launched directly from the beach. In most areas, there is much room for improvement of fish handling and preservation on board.

The traditions and desires of local fishermen are extremely important considerations in the planning of new technology. Whenever unfamiliar equipment or methods are introduced, it is most important that they be accompanied by thorough training programs in their efficient operation and maintenance.

Southeast Asia

Bambang Priyono, Resource Economist, Indonesia

This region covers an area much greater than that of the United States, with many island and coastal nations which harbor a tremendous range of maritime conditions and fisheries. Commercially valuable stocks include pelagic schooling species such as tuna, skipjack, sardines, and mackerel; as well as demersal fish species and crustaceans such as prawns, crabs, and lobsters.

The region's traditional fishing vessels are made of wood. Some are dugout canoe-style, but most are planked. Most small craft have one or two outriggers. This style of construction is common in vessels from six to eleven meters in length overall, and larger outrigger canoes are also seen in some areas. Most of these vessels are equipped with sails, and many also use outboard motors or inboards, a typical installation being a 5-20HP direct drive inboard gasoline engine. The most common fishing methods employed by these craft are handlines, pole and line, gillnets, bagnets, and various seines. Many use light to attract schooling fish at night. A very substantial part of the region's production comes from small craft in this category.

In the early 1960's bottom trawling was introduced to many countries of the area. It proved highly effective in many areas, and led to substantial increases in production. Many countries in the region now have a number of trawlers, longliners, and purse seiners of 50-200 Gross Tons.

Several countries are also involved in large scale offshore fishing operations, with vessels ranging up to 300 GT, either owned nationally or operated in joint ventures with countries such as Japan.

While some areas of the region are overfished, in many places there is room for expansion or improvement of the fleets. Many nations are very anxious to exploit the offshore waters of their recently established 200 mile Exclusive Economic Zones. It is felt that vessels in the range of 30-60 GT might be most appropriate for this purpose. Fishing vessels are quite expensive in the area at this time. For example, in 1980, the price of a new wooden tuna pole and line vessel of 30GT in Indonesia was approximately US$72,000, including motor and fishing gear. High quality wood for boatbuilding is very expensive, and in these tropical waters worms and borers contribute to short vessel life and high maintenance costs. Alternative building materials such as fiberglass, ferrocement, or steel would be superior in many ways. In addition, the following criteria should be considered in vessel design.

Many fisheries are aimed at fast-swimming pelagic species. For chasing these fish and manoeuvering with fishing gear, high speed and moderate speed vessels are most effective. Distances traveled to the fishing grounds are often greater than those seen in the fisheries of other regions. For example, some tuna boats of 20-30 GT make trips of one to four days, traveling to grounds 10 to 80 miles from their ports. Travel time to and from the grounds is often a high proportion of total vessel time, and faster boats facilitate landing fresher catch for higher prices.

Several nations of the region are oil producers, and although fuel is sometimes unavailable on outlying islands, fuel prices in many areas are lower than in the U.S.A. Sailing fishing boats have been used in this region for thousands of years, and their use is still common in many areas. Although the transitions between the monsoon seasons frequently bring strong winds which interrupt fishing, during most of the year steady, moderate trade winds provide favorable conditions for sailing. However, local fishermen are well aware of the advantages of motors, and many consider that the combination of sail and motor is most appropriate. Japanese motor manufacturers have very aggressive and effective practices for marketing, distribution, and service in the region, and most fishing vessels use Yanmar, Kubota, or Yamaha motors. These are also considered the most reliable and easiest to maintain and service.

Although a few fisheries of the region concentrate on low value species such as sardines, most fisheries produce moderate to high priced species, so most vessels do not need the capacity to hold very large quantities of low priced fish.

Some areas are characterized by strong currents and choppy seas. However, in most areas, seas are generally moderate, and permit operation in the medium speed range most of the time.

Most craft which make trips of more than one day's duration use ice on board to preserve the catch, so insulated holds or fish boxes are necessary. The pole and line fishery for tuna is widespread, and vessels for this fishery must have live bait wells.

The more developed areas have good harbor facilities with support services, although drydocking facilities are few, and generally very expensive. Outlying islands may be less well equipped, supporting only very small boats which fish strictly one day trips.

With regard to the above criteria, any new small craft must offer characteristics at least as favorable as the traditional banca fishing craft, or they are not likely to find acceptance by the fishermen.

In introducing prototypes, care must be taken to provide instruction in fishing methods which can take advantage of improved fishing capability of the prototype boats.

Due to the importance of small scale fisheries in the region, the FAO has done a great deal of work in the design of vessels appropriate for use in this sector, concentrating on the Philippines. A few of their sketches of tradional and alternative craft are included in the Appendix of this paper, as well as a design by the World Bank of a 30 GT tuna pole and line vessel. Since many other fisheries of the region are similar, this work may be considered a good base for working in many other nations.

Southern Asia - Focus on Sri Lanka

Rajapaksa Don Warnadasa, Marine Engineering Assistant, Sri Lanka

Sri Lanka is a large island in the Indian Ocean off the southeast coast of India. Its large fishing sector exploits several pelagic species such as tuna, skipjack, mackerel, swordfish, and shark, as well as a variety of semi-pelagic and demersal species. Shrimp, lobsters, and crabs are also fished commercially.

Wind and sea conditions in the area are generally moderate, allowing a relatively high number of fishing days per year. Occaisionally, during the monsoon seasons, winds blow up to 30 or 40 knots, causing several consecutive days of down time for fishermen.

The island has a few trawlers of about 100 GT, but the majority of the fishing activity is done with an estimated 25,000 fishing craft of less than 15 meters LOA. Almost half these small craft are wooden, non-motorized sailing outrigger canoes less than 12 meters in length. Another 5,000 similar boats are powered by outboard motors which run on kerosine (in 1983 kerosine and diesel sold for around US$2 per gallon, while gasoline cost about US$3 per gallon due to excise tax). In the 10 to 12 meter range there are also many fishing vessels with inboard diesel engines which fish offshore waters.

Dominant fishing methods are hook and line, trolling, gillnetting, beach seining, longlining, and shrimp trawling. Boats of less than 10 meters generally make 1-day trips, while the 10-12 meter vessels stay at sea for periods up to 1 week. On these trips ice is used for preserving the catch on board. The larger boats fish out to 80 miles offshore.

Most of the harbors in the country are overcrowded, and almost all the small craft are forced to land and offload catch directly on the beach, sometimes through moderate surf. Support facilities for hauling out vessels, maintenance and repair are also insufficient. Rising costs of materials, equipment, and operations have seriously hurt the fishing sector.

The Sri Lankan government is currently engaged in an ambitious campaign to improve its fishing fleet and increase production. Since the native woods which were used for boatbuilding are becoming very rare and expensive, alternative materials are being used, primarily fiberglass with some construction in steel as well. High fuel costs have prompted a great deal of experimentation with alternative energy sources, concentrating on sails for wind power as well as on solar power. Fuel efficiency is a top priority in boat design for the area.

The experience of recent fisheries development programs in Sri Lanka should provide valuable lessons for similar programs in other regions. The following guidelines for the introduction of new vessels and gear to the artisanal sector have been developed.

Combination vessels, which can switch to different methods and concentrate on different species according to the seasons and fishing grounds, would be most valuable.

Any new technology must be well suited to operation under local conditions, as outlined above. Vessel stability and safety for open water fishing is the top priority. Cost effectiveness is, of course a most important factor.

Since few vessels concentrate on very high volumes of low priced species, carrying capacity relative to vessel size need not be excessive. However, vessels over 10 meters need a capacity sufficient for trips of several days' duration.

Artisanal fishermen are most likely to accept new technology which is not too dissimilar from that which they are familiar with. New types of vessels, motors, and sailing rigs should be as simple and easy to use and maintain as possible. It is essential to introduce new technology carefully, with demonstrations and training programs in which the fishermen are taught proper operation and care of new equipment.

The Caribbean Region

Norbert Simmons, Fisheries Technologist, Bermuda

The Caribbean region includes an area from 10 to 32 degrees North Latitude, stretching from the Grenadines to Bermuda. It contains dozens of self-governing, inter-trading islands. The quality of the different fishing grounds of the region are strongly affected by the two categories of islands:

1) Islands that rise immediately from the seabed with very deep water almost immediately offshore.

2) Islands surrounded by an extensive shallow (coral covered) reef with gradual drop-offs.

The resources of the latter include pelagic as well as reef dwelling species, grouper, snapper, and lobsters. Waters surrounding the former type of island, with a sharper drop-off, generally have fewer reef-dwelling stocks, and fishermen there must concentrate more on deep-water or pelagic species. The major commercial stocks include the reef species mentioned earlier as well as tunas, kingfish, mackerel, shark, and wahoo.

Fisheries play a very prominent role in the regional economy, in most places ranking third or fourth in terms of GNP behind such activities as tourism, agriculture, home industry, and offshore company (tax exempt) business.

Fishing vessels range from the traditional long and narrow West Indian dugout canoe to large modern diesel work vessels and shrimp trawlers of a few areas. The four to six meter open craft usually use 1.5-25HP outboard motors, and operate with one or two man crews. Wood is the most common construction material, but in some areas fiberglass boats are increasing in number. Making one-day trips, these boats use hook and line, fish traps, trolling gear, beach seines, and in some areas, gillnets.

In the southern part of the region, the most common larger vessels, of 9-12 meters in length, are the traditional decked wooden fishing sloops, most of which have small inboard diesels. These may have a crew of up to five members, and fish over 100 traps in deeper water. Some also use deep water reels for snapper and grouper. In the northern islands, moving closer to the USA, it is more common to see fiberglass vessels of 9-14 meters in length, often with relatively large inboard diesels, equipped to fish traps or deep water snapper reels.

Hydraulic and electric line haulers and reels have contributed significantly to offshore vessel development. The larger vessels of this category make trips of up to one week's duration. Most of these preserve the catch with ice. However, during some seasons shortages of fresh water lead to unavability of ice. For many years live wells have been used to keep reef fish and lobsters on board. As the reef dwelling stocks have become less abundant, many fishermen are concentrating more on the faster swimming pelagics, which do not survive in live wells; consequently, the use of ice is on the increase.

The use of sail is not uncommon in most fisheries of the region. It was very common 20 years ago, and is still used in inter-island freight delivery. The steady northeast trade winds provide very good conditions for sailing throughout the region. Fuel is very expensive in most areas (fuel in Bermuda cost US$1.50 per gallon in 1982), and the re-introduction of sail to local fisheries could lead to substantial savings in operating costs. Many fishermen are becoming increasingly interested in this fuel-saving alternative.

The main cities of most islands provide limited landing facilities for commercial vessels, transport and fishing craft together. There is a need for more dock areas and facilities for commercial fishing vessels. In some areas, smaller fishing boats use beaches for landing and offloading catch.

Most fisheries concentrate on species of moderate to high individual value. While these vessels are not required to carry very large loads of low priced fish, the range and duration of fishing trips is generally increasing, and longer trips require increased hold capacity for the catch.

Although there is some evidence of overfishing in some inshore waters, there may be room for expansion of the fisheries for pelagic species, and there is definitely room for modernization of currently used vessels and equipment. The planning of new vessels must take into account the local conditions outlined above. One major design criterion would be a vessel's ability to participate in a variety of fisheries. A combination, multi-purpose vessel would be able to explore new fisheries, and take advantage of the region's diverse resources as well as stocks which migrate seasonally.

The 200 mile Exclusive Economic Zones of many Caribbean nations are presently fished by foreign nations under permits, with vessels generally of the 30 meter, 200GT classes. Most of these vessels are longlining tuna. This is a possible area for future fisheries expansion for many island nations.

Latin America - Focus on Venezuela

Alejandro Acosta, Fisheries Technologist, Venezuela

Venezuela has a coastline approximately 1700 miles long, on the north coast of South America. In some ways it is atypical of Latin American nations - for example, its oil production eases foreign exchange problems, and provides fuel for the fishing fleet at relatively low prices. However, the country provides a good illustration of the contrasts between large and small scale fishing operations working in the same area.

Species most commonly fished commercially include sardines, tuna, skipjack, snapper, grouper, shrimp, squid, and octopus. The continental shelf along this coast is relatively narrow, and it is intensively fished by over 200 Venezuelan shrimp trawlers of the double net type typical in the Southern USA and other areas. Since the resources and grounds of waters off the edge of the shelf are not well known by local fishermen, both small scale and industrial scale vessels compete for the resources of the narrow shelf. The country also runs a small fleet of offshore tuna vessels, pole and liners, longliners and purse seiners, which have the capability of freezing on board, and making longer trips to more distant grounds.

Steady northeasterly trade winds dominate the area, and provide conditions generally favorable for operation of a wide range of vessels.

The small scale fishery in Venezuela supplies much of the fresh fish and the domestically consumed protein. It also provides employment and income for a large percentage of the population in some areas of the country. However, it is generally considered a low priority, and many obstacles hinder its development. The small scale fleet consists of around 6,000 vessels, ranging from 3 to 12 meters in length. Wood is the traditional building material, and the most popular. There are some small fiberglass fishing craft, including boats recently manufactured in the country under a joint venture with Yamaha of Japan. However, the fishermen have little experience operating, maintaining, and repairing fiberglass craft, and despite its advantages, they are sometimes reluctant to accept this new material. More energetic promotion of this fiberglass for fishing craft might overcome this problem.

Most of the small vessels use outboard engines running on gasoline, and Yamaha is the most common brand seen. Vessels over 8 meters in length often have inboard diesel motors. Fishing gear most commonly used by the artisanal fleet includes handlines, longlines, traps, beach seines, trolling, and gillnets. Most craft have neither electronics for navigation and fishfinding, nor mechanical equipment for hauling gear, and fishing operations require a great deal of manual labor. Most of these vessels make one-day trips, but some stay at sea for up to one week, using ice for preservation of the catch. Crews on small vessels generally have one to six members.

While harbor facilities for the industrial fishing fleet are adequate, dock and offloading facilities for small craft are lacking, and many small craft land their boats and catch directly on the beach.

In Venezuela, fishery products have a high demand, and seafood prices are relatively high. It is felt that improved management and modernization of the fleet can lead to increased production in the long run. The mollusks such as squid and octopus are underexploited. More research is needed to estimate resources farther offshore, outside the range of the traditional fisheries. Vessels of 11 to 15 meters in length might be within the reach of some artisanal fishermen, and such craft could work on the resources near the edge of the continental shelf.

Since local stocks are very diverse and knowledge of potential new fisheries is incomplete, any new vessels must be designed for multi-purpose fishing. Conversion from one fishery to another should be as fast and simple as possible, allowing the fisherman to experiment with new stocks and grounds, as well as to enter seasonal fisheries.

In order to increase the efficiency of vessels in the 8 - 12 meter range, some electronic and navigational equipment, such as compasses and depth recorders would be very helpful. Mechanical or hydraulic equipment for hauling fishing gear would also be a significant improvement. However, since the coastal region depends heavily on fishing for employment, innovations which lead to reduced crew size would probably bring more problems than benefits. Planners should be careful not to displace fishermen from their current occupations.

Unfortunately, in contrast to cheap fuel, imported equipment and machinery is quite expensive, and sometimes unavailable. For successful operation in the long run, any new vessels or equipment must be accompanied by a commitment to provide spare parts and service in the country.

Sea and wind conditions would be favorable for sail-assisted fishing craft, but in Venezuela where fuel is inexpensive, they might not be able to compete with faster motorized craft.

Artisanal fishermen have very little exposure to vessels and gear not found in their area of operation. The introduction of new technology must be accompanied by education and training of fishermen. They should be made aware of the advantages and disadvantages of unfamiliar systems, and when new vessels are actually delivered, it is essential to provide thorough training in the operation, maintenance, and servicing of the new equipment.

West African Region (Eastern Central Atlantic)

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Along the coast under consideration, from Morocco to Zaire, fishing is an important activity for the production of food and income. The area contains large concentrations of pelagic stocks, such as sardines and anchovies, as well as moderate to sparse distributions of demersal species. Some countries have reasonably high stocks of shrimps and other shelfish. The fishing grounds in the northern part of the region are generally the richest.

For discussion purposes, the region can be said to contain four fishing sectors - a foreign flag distant water fleet, an African-owned industrial fleet in which some nations enjoy the right to fish in the waters of other African nations, local industrial scale fleets fishing in the waters of their flag states, and small scale, artisanal fleets. The distant water fleet is composed principally of foreign-owned freezer or processor fishing vessels, sometimes operating under agreements or joint ventures with local governments.

In the industrial sector of the African owned vessels, boats over 15 meters LOA include shrimp trawlers, purse seiners, side trawlers, longliners, and vessels which empoloy fish and lobster traps. The most common construction material is steel, but there are also wooden and fiberglass vessels. These vessels fish the grounds from 5 miles offshore to out past the edge of the continental shelf. Duration of voyages ranges from 3 days to one month, depending on the country and the fishery. Most of these boats use ice for preserving the catch, but some have freezing capability.

In this region there are approximately 40,000 artisanal fishing vessels, fishing one-day trips within 10 miles off the coast, using hook and line, cast nets, set nets, drift nets, surrounding gillnets, purse seines, and longlines. These craft fall into 3 main categories. The first category consists of vessels ranging in length from 5 to 6 meters with beams less than one meter. They are usually dugout from tree trunks, operated by one man with paddle as the means of propulsion. In recent years, the number of these vessels has declined considerably as fishermen move to the larger vessels.

The second category consists of vessels with lengths ranging from 6 to 9 meters, with beams greater than one meter. These vessels are usually operated with outboard engines in the 6HP to 15HP range. In some countries the use of sails is becoming popular due to high fuel costs and economic conditions. These vessels are built with wooden planks, and carry up to 10 fishermen.

In the third category, which has received a great deal of attention because of its suitability for use with larger fishing gear, vessels range in length from 9 to 15 meters with beams less than 5 meters. They are operated with outboard engines in the 25HP to 40HP range. Construction is of wooden planks, and the boats carry up to 15 fishermen. The fishing gear used by these boats consists of surrounding nets, purse seines, gillnets, and some longlines. Since winches and hauling machinery are not generally used in the area, a great deal of manual labor is needed to haul and set the relatively large nets.

In the past, storage facilities such as insulated ice boxes have been lacking on the vessels, and this limits the duration of fishing trips. Recently, insulated fish boxes are becoming popular in some countries, but their use is also dependent on the sporadic local availability of ice. Cabins or other crew shelter are lacking on virtually all small craft.

In small fishing craft, wood has been the traditional building material because of its availability, and also because local boatbuilders have experience only in wood construction. Fiberglass vessels have been introduced in some countries, but have not found much success.

In many areas, there is room for expansion of both the industrial and the artisanal fleets. New vessels could fish the same stocks currently exploited, as well as some underexploited species such as squid and sharks.

In planning effective vessels, certain deficiencies in the supporting coastal facilities must be kept in mind. Adequate harbor and port facilities are sometimes few and far between, and in many areas the small craft land and unload product directly on the beach. Problems in obtaining foreign exchange often cause difficulties in importing motors and spare parts, and it is best to find out in advance which manufacturers have effective local networks for parts and service. It is all too common to see fishermen unable to fish because their motors need repairs or parts which are simply not available in the country.

Fuel is generally quite expensive in the region; for example, in 1983, in Sierra Leone, gasoline and diesel cost approximately US$2.00 per gallon. Any new vessels should be as fuel efficient as possible. The northeast trade winds and southwest monsoons are prevalent, often providing conditions favorable for sailing, and the use of sails as auxiliary power is increasing. Vessels which concentrate on pelagic species should be capable of relatively high speeds.

Since many large and small vessels fish low priced, high volume species, carrying capacity for any vessel should be as large as possible relative to thecost of the boat. In these tropical temperatures, there is much room for improvement in on-board equipment for fish handling and preservation.

The versatility offered by multi-purpose vessels could be a very valuable quality for new fishing craft here, especially small craft. However, even with a multi-purpose vessel, conversion from one fishery to another is sometimes very expensive, especially with larger vessels. A careful feasibility study should precede any added construction expense involved in making a vessel suited for various fisheries.

Labor is relatively cheap in the region, and unemployment is sometimes a problem. Consequently, crew numbers tend to be quite high relative to vessel size. While modernization of equipment could bring some benefits, changes which would reduce crew size might cause serious social and economic problems.

Worms and borers are a serious problem for wooden vessels in the area, and fiberglass, steel, or WEST (wood-epoxy saturation technique with an outer coat of fiberglass), would offer significant advantages in increased durability and low maintenance. However, the infrastructure and expertise for service and repair of these materials is presently incomplete, and support facilities must accompany such new developments.

Artisanal fishermen may be reluctant to try technologies which differ significantly from traditional methods. Planning of new vessels and gear should include input directly from the fishermen. Actual introduction of new technology must be done carefully and tactfully, and thorough training programs to familiarize fishermen with new equipment are a must.

DEVELOPMENT OF A 48 FOOT MULTI-PURPOSE FISHING VESSEL : A 12 YEAR RETROSPECT BY THE BUILDER

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Abstract:

This paper explores the concept of semi-custom boat building for a variety of commercial purposes, using a standard production hull form. The viewpoint is the builder's, from the inception of the idea, through the production and sale of the boats, to a point some 12 years later, when certain conclusions are drawn as to the success of the product based on usage. The author's company for semi-custom construction, for seven years of operation, was Marine Management Company, of Miami, Florida.

Design Development and Basic Construction

Prior to 1969, there were almost no fiberglass boats in the 40'-50' size range designed for commercial fishermen, using state of the art technology, being built on the U. S. east coast. Most of the boats in this size range were re-interpretations of hulls designed for pleasure or military use. Based on my past experience in boat operation and repair and on many conversations wtih commercial fishermen, I decided to commission a design for a commercial hull to be built in fiberglass. The underlying philosophy was that the needs of fishermen and other commercial users in diverse areas could best be met by taking a well designed, versatile, production hull and customizing it to fit an individual's needs. Construction would be rugged, easy to maintain and building techniques would be flexible enough to permit semi-custom construction at a price that the user could afford while the builder could make a profit. To this end the basic hull design had to be suitable for multipurpose uses.

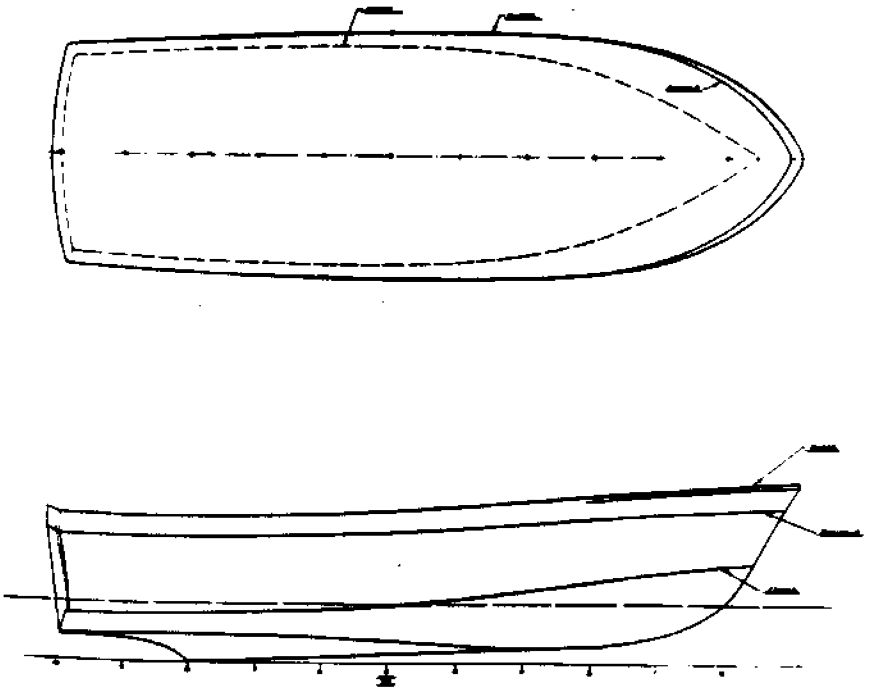
To meet these requirements, the boat had to be seaworthy and fast in the lightly loaded condition, given adequate horse power, and at the same time, be seaworthy and efficient at pure displacement speeds in a heavily loaded condition. A draft of four feet or less was considered practical to provide access to shoal harbours. Adequate protection was required for the propeller in case of accidental grounding. Lastly, the vessel had to provide a relatively stable and comfortable work platform.

A hull of about 50′ in length seemed to offer the most versatility as a boat small enough to be operated by two men on a day trip basis and yet large enough to accommodate 4 to 5 men, plus ice and gear for longer trips offshore.

Because of the varied load and speed capacities which were anticipated, trim was a very important consideration in the design. Regardless of load situations, traveling for any distance either down by the bow or stern was not acceptable. Ballasting and/or trim tabs were not options for practical use by working owners.

Given these basic criteria, Wynne Marine Inc. designed a hull to be built in fiberglass with the following specifications:

loa 46′7" or 48′
beam 14′7"
draft 3′6"
hold capacity 600 cu. ft.
fuel 1200 gallons for single engine
500 gallons for twin engines
water 300 gallons



Basic Hull Drawing

The design is hard chine with a sharp vee forward to reduce pounding and ample flare for dryness. Aft, the hull flattens out for load carrying ability. The molded-in keel helps provide good propeller protection and, in combination with a specially designed tunnel, allows shoal draft and a low shaft angle of 6 1/2 degrees in the single engine configuration. Twin engine installations also make use of shallow tunnels, so that shaft angles are low and the propeller tips are above the bottom of the keel.

In the single engine installation, the engine with the fuel tanks outboard on either side is located near the center of buoyancy. Trim does not change adversely with a change in fuel load. This also permits the use of a short propeller shaft of approximately 80 inches. Space is available for a below deck fish or storage hold forward of the engine, thus eliminating the often noted problems of long shafts and inaccessible intermediate bearings.

The topsides are almost vertical, not slab sided but with some curve, to make it easy and safe to work over the side. The working deck is 30′ long in the 46′7" model and 32′ long in the 48′ boat. The length options and single and twin tunnel options are made possible by the use of mold inserts. There is sufficient room under the forward deck to provide 4 bunks , head and galley.

One of the most obvious variations from boat to boat is in the configuration and layout of the superstructure/pilot house. Especially in commercial fishing, the location of steering and controls are significant to the individual user. While one style of molded deck house and fore deck, of core construction, was offered, most customers required a special design and fabrication in this area.

Construction is on the heavy side. The hull is solid glass laminate (i.e.,no core material) and is laid up in a one piece female mold, as described in the Appendix.

After stringers, bulkheads, deck clamp and frames were installed, the hull was removed from the mold, by crane, and set on a wheeled, steel shop cradle. The fuel tanks, of fiberglass or metal, were then set in place. If a standard molded fore deck and/or house were to be used the crane would position the pieces. The cradle was then moved to a construction station, the hull leveled, the shear cut to owner′s specifications, and finish work commenced.

Construction differed from industry standards in several ways. As previously mentioned wood frames were installed from station 3 to the transom on 34" centers. These frames extended from just above the chine to the shear line and served several purposes - to stiffen the hull sides and to eliminate stress concentrations at the hull to deck joint, when the rub rail lands against a dock or loading platform. These frames also provide an easy point for securing the cockpit ceiling, with space between the frames for air vents to engine spaces. All interior cabinet work was considered structural and glassed to the hull accordingly.

As indication of the rigidity of this construction, all engine alignment was completed in the shop to .003" at the shaft coupling faces. Each boat was then trucked about 6 miles to the water and launched. After launching alignment was checked with a dial indicator. In the three years after adopting this procedure, it was not necessary to realign an engine after launching.

A method was developed whereby deck houses were generally built of fiberglass and balsa core panels laminated in conveniently sized sheets, then cut and joined as necessary. While this technique was not new in custom construction, it offered much flexibility for the semi-custom builder faced with a variety of cabin arrangements.

Since our basic construction techniques were the same on all boats and as the work force was experienced and stable, it was not necessary to have formal drawings for each commercial boat even though each was different from all the others in some ways.

Pleasure boats were usually custom designed by Wynne Marine Inc., with a complete and detailed set of plans. With few exceptions all of our commercial customers were experienced and knew what they wanted, how it should work and what it should look like. The only problems were communications. Experience proved that the overall appearance of a boat, in plan and profile, could be worked out with the owner quickly by using preprinted basic hull drawings, a pencil and a Big eraser.

We would discuss with the buyer in detail how he planned to work the boat. These conversations made clear the problems of weight distribution, gear location, work stations, etc. that had to be solved. Compromises were usually worked out by going aboard boats under construction. With four or five boats in the shop at varying levels of completion, the eventual solution to a problem could be illustrated in actual construction detail. We had very few cases of misunderstanding and subsequent rework using this system.

With the details worked out, a fixed price contract was prepared. Pricing was somewhat laborious even though many elements and costs were the same from boat to boat.

A detailed contract form, describing construction techniques and materials, was utilized, and a very specific payment schedule was part of the contract. A 10% deposit upon signing of the contract was followed by an additional 20% upon the date that the hull was begun. Other payments were tied to specific events in the progress of construction - such as the date of engine delivery or the date that the cabin exterior was in place. A balance of about 15% to 20% was reserved for the date of delivery and acceptance by the buyer. With this very clear cut approach to finances, we experienced little difficulty with cash flow and had a good relationship with the customers in terms of our mutual responsibility. For the protection of both parties to the contract, it was understood that the cost of builder's risk insurance was tied to boat value and built into the contract price.

Review of Boats In Use

Evaluation of the semi-custom approach to work boat construction is obviously tied to financial results. Assuming that the builder can make a profit, the winner or loser is then the boat owner. The building-in of specific utility for the individual and the building-out of maintenance and breakdown worries can go a long way towards enhancement of the bottom line. After 7 years of production and approximately 65 boats built, we determined that most of the boats were still being operated by their original owners. Their uses were extremely varied and included such applications as a fast research support boat for the University of Miami, mackeral and king fish gill netters with roller rigs for south Florida, tub trawlers and offshore lobster boats for New England, tour boats for Bermuda and a tuna boat for Hawaii. One boat was in service as a treasure/salvage vessel and several had been U.S. Coast Guard certified to carry diving parties of up to 35 persons. With suitable variations in superstructure, interior layout and power, the hull was being used for charter sports fishing and pleasure cruising.

With this variety of employment, some hulls were being operated with light loads at relatively high speeds, some were viewed as strictly displacement hulls and others were used in both modes. To demonstrate the success of the design concept as implemented by semi-custom construction methods, a description of several boats is offered here.

The attached plan and profile drawings demonstrate the flexibility that can be developed with this single good hull design.

One South Florida fishery requiring speed and load carrying capability is the gill netting of kingfish and mackerel. In this fishery, spotter planes are frequently used. Working with several boats, the plane's pilot searches for fish, and when a school is sighted, he radios the location to his boats. The race starts for the fish with the first there controlling the school and making the first set. If there are large numbers of fish, another boat may have an opportunity to set its net and share the catch. It is not unusual to catch 20,000 to 35,000 pounds in a set. Consequently, when the net is hauled back and cleared, the boat is back to the slow speed range. Power for most of these boats is usually a 12V71TI, which provides speeds of approximately 26 to 28 knots, running light.

A particular New England offshore lobster boat is a good example of the hull used at semi-displacement speeds. In this case the cruising speed is 11 to 12 knots. The average trip is 3 days with 12 hours spent traveling each way and 48 hours fishing. Cruising speed is about the same in each direction as the vessel loading does not change much. The weight of lobsters in an aereated, sea water tank on the return trip is offset by the use of 4500 lbs of bait, fuel and water consumed prior to returning. Fuel consumption is 500 to 600 gallons of diesel, with a Caterpillar 343 for power. The live tanks will hold up to 6700 pounds of lobster.

This boat tended 800 traps set in trawls of 25 to a line in 600' of water. Sea keeping was important. Even though the boat did not begin a trip in adverse weather conditions, she was caught offshore in a blow many times. Her original owner operated her so successfully for 7 1/2 years that he has just taken delivery of a replacement for her - a 65' aluminum vessel whose cost is approximately a half million dollars.

The day fishery for stone crabs in southwest Florida requires a fast, maneuverable boat when pulling traps and a seakindly load carrier when moving traps. In this usage, the boats leave the dock before daylight at a time that will permit them to be at their trap lines at first light. The daily run out may be as short as fifteen miles or as long as sixty miles. Power for boats in this fishery is usually provided by either a GM12V71N or GM12V71TI, giving cruising speeds of about 20 knots light and 10 to 12 knots loaded.

Most boats of the 40 to 50 foot size are equipped with two hydraulic haulers mounted at the stern. These are operated by the two crew members on an alternate basis, so that one man is hauling a trap while the other is engaged in clearing and rebaiting a trap just hauled. Traps are individually buoyed and set in water depths ranging from 10 to 60 feet with 500 to 650 traps per line. Traps are spaced approximately 200' to 300' apart and are hauled, cleared, baited and reset, at the rate of one per minute per hauler in 25 to 30 feet of water and one every 1.5 minutes in deeper water. Speed of the boat and spacing of the traps are such that there is just enough time for each puller to complete his job before his next buoy is along side. This pace requires team work and a very maneuverable boat, especially when working in cross winds and tides. It is also a strain on engines and steering systems. In a typical hauling cycle, engine RPM can vary from idle to 1800 RPM in forward, then quickly change to idle and up to 1800 RPM in reverse when a buoy is reached. This procedure represents a minimum of 1200 gear changes a day and often times some powerful backing down.

In this daily routine, boats usually return to the dock between 3 and 7 P.M.. The product is light in weight and is carried to the dock unrefrigerated, so speed is important. By contrast, the seasonal setting out or bringing in of traps is slow work requiring unusual load carrying ability. Moreover, lines of traps are often moved in the course of the season. During these operations a full line of 600 traps, weighing 42,000 pounds dry or 48,000 pounds wet, is stacked on deck.

Builder's Conclusions

The preceding examples prove that a well designed production hull can be an effective multi-purpose boat when finished on a semi-custom basis.

The top fishermen of today using boats in the 40' to 50' range have significant capital investments. A stone crabber with a new 48' boat might have $150,000 in boat and electronics costs and $85,000 in traps. Additional funds will be committed to off season fishing gear, such as long lines and fish traps. This crabber will employ two full time crew members to help him work the boat, and two additional persons may be needed during the off season to build and repair traps. This man is not ready for a smaller, fuel conservative boat that might put him back into the "mom and pop" category of fishing, although he is definitely receptive to means of reducing costs and improving profit. For example, a savings in fuel costs is not beneficial if such factors as travel time, productivity or crew wages are adversely affected.

With specific regard to construction methods and materials, my experience supports the concept of over building for strength and longevity. Most of the fishing boats that I have known well in the past 15 years have no regular preventative maintenance programs. Repairs are made when a failure stops the fishing. The gradual deterioration of structural components is apt to be ignored as long as the boat can be fished successfully. This means that hull, underwater gear, steering systems and engine foundations should be over built to compensate for rough usage. Use of light weight, modern materials and sophisticated building techniques must be balanced by regard for the abuse which the end product will receive.

The semi-custom concept has particular value for the builder who prefers a relatively small operation while appealing to a broad market. With the aid of the small personal computers now available, pricing and contract preparation could be accomplished in minutes instead of several hours. The computer could also be used to generate shop drawings and regulate inventory and billing. This would give the innovative builder in a small shop more time to solve customers' problems.

In each of the fishing situations described previously, the boat is the central tool in an economic unit which provides employment for several persons while producing a significant food product for distribution to a large number of people. My observation of motivated fishing boat owners suggests that they have in common a traditional approach to their work and that they are interested in innovations in their tools only if those innovations improve their financial basis. In this regard, the boat builder can emerge as a force for conservation by employing good structural techniques and reliable, easy to maintain materials. The semi-custom building situation offers a unique framework in which to keep costs under control and quality high.

APPENDIX

General Specifications for construction of hulls by Marine Management Company

DIMENSIONS: 46′ or 48′ long x 14′6" beam x 3′6" draft

HULL: hand lay up, lamination of solid fiberglass, with approximate thickness in sides of 1/2" to 3/4" at chines to 1 1/8" or 1 1/4" at keel. Hull laminate schedule: gel coat/one layer of 3/4 oz. mat / 5 layers of 1 1/2 oz. mat and 24 oz. woven roving in pairs to develop the hull thickness of nominal 1/2". Additional hull thickness is developed when stringers are glassed in place as follows.

STRINGERS: Four full length of 1 1/2" x 8" mahogany, totally encapsulated with one layer of 1 1/2 oz. mat and two layers of 24 oz. roving. In way of engine(s), stringers are doubled to 3" thickness and covered with two layers of 1 1/2 oz. mat and four layers of 24 oz. roving. Voids between wood and hull are filled with fiberglass putty and a radius is formed in the putty between stringer and hull.

BULKHEADS: Four standard bulkheads of 3/4" plywood glassed in place on each side with bonding angles of one layer of 1 1/2 oz. mat and two layers of 24 oz. roving. All four bulkheads are watertight. Voids between bulkheads and hull are filled with fiberglass putty and radiuses formed.

FRAMES: Frames from 1" x 6" mahogny glassed in place with one layer of mat and two of roving. Frames installed on 34" centers starting at station #3 and going to transom, and extending from shear to chine. Forward of station #3, framing and support are provided by cabin floors and interior accommodations, such as bunks, which are glassed in place with one mat and two rovings and which are considered structural.

DECKS: Main deck of 3/4" plywood nailed with threaded bronze nails and glued to 2" x 4" deck beams on 12" centers. Deck beams bolted to clamp with 5/16 stainless steel machine screws. Main deck is fiberglass covered with two layers of 1 1/2 oz. mat and gel coated.

Forward deck is of same construction as main deck, or is of molded glass, hand laminated with 11/16" balsa core, depending on buyer′s requirements.

RUB RAILS at shear of mahogany, 1 1/2" x 3" x 2", with metal guard at buyer′s option.

GUNWALES and COAMINGS of 3/4" plywood secured with stainless steel screws and glassed with one layer of mat.

ACCOMMODATIONS to buyer′s specifications.

FINISHES and MATERIALS: All molded surfaces to have gloss gel coat finish. Other fiberglassed surfaces which require finish to be painted with mat gel coat. Painted wooden surfaces to have one coat of primer and one coat of gloss enamel of epoxy type. Interior mahogany trim to be varnished. All woven roving to be 24 oz., and all mat, except surface next to hull gel coat, to be 1 1/2 oz. Bottom painted with anti-fouling.

FUEL TANKS: With single engine installation, either molded fiberglass as standard (two tanks with capacity of 550 each), or custom tanks of aluminum or epoxy coated black iron.

With twin engines, custom tanks of aluminum or epoxy coated black iron.

WATER TANKS: Fiberglass or aluminum to meet buyer's requirements.

UNDERWATER GEAR: Extra heavy duty construction. Sturts, rudders and skegs of stainless steel; shafts of Armco Aquamet stainless steel, 2" or 2 1/2", depending on engine; rudder ports, shaft logs, and propellers of bronze, with propellers to be four blade sized to engines.

Single engine boats to have "Y" strut and skeg; twin engine boats to have "V" struts and no skeg.

ENGINE(S) of buyer's choice.

ENGINE BEDS: 1/2" x 6" x 6" steel, epoxy coated, full length of engine, and through bolted with ten 1/2" carriage bolts on each engine stringer.

RUDDER PORT BRACKET: 6" steel channel, epoxy coated, with bronze bearing.

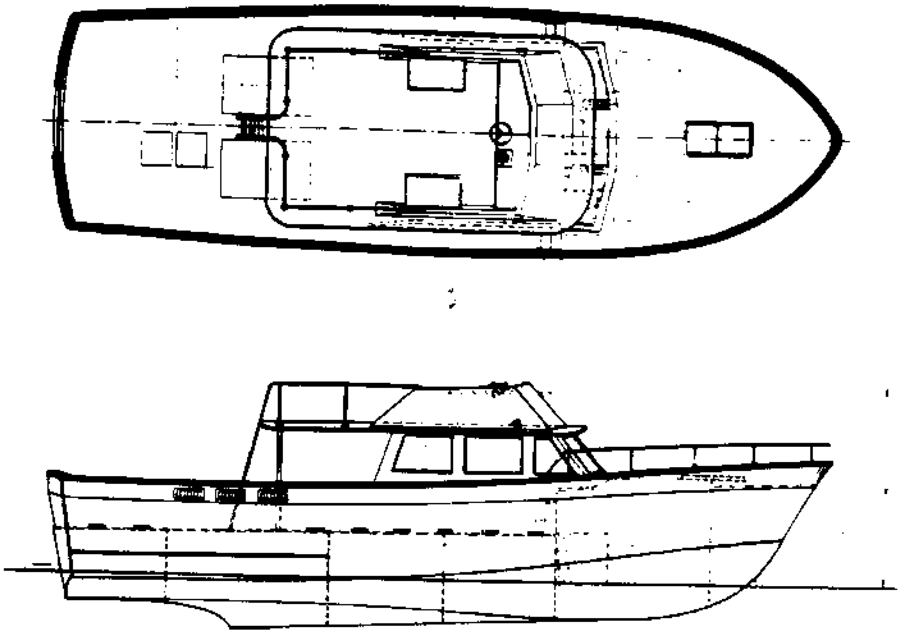
STEERING: Mechanical or hydraulic to buyer's requirements.

EXHAUSTS: Dry or wet as required.

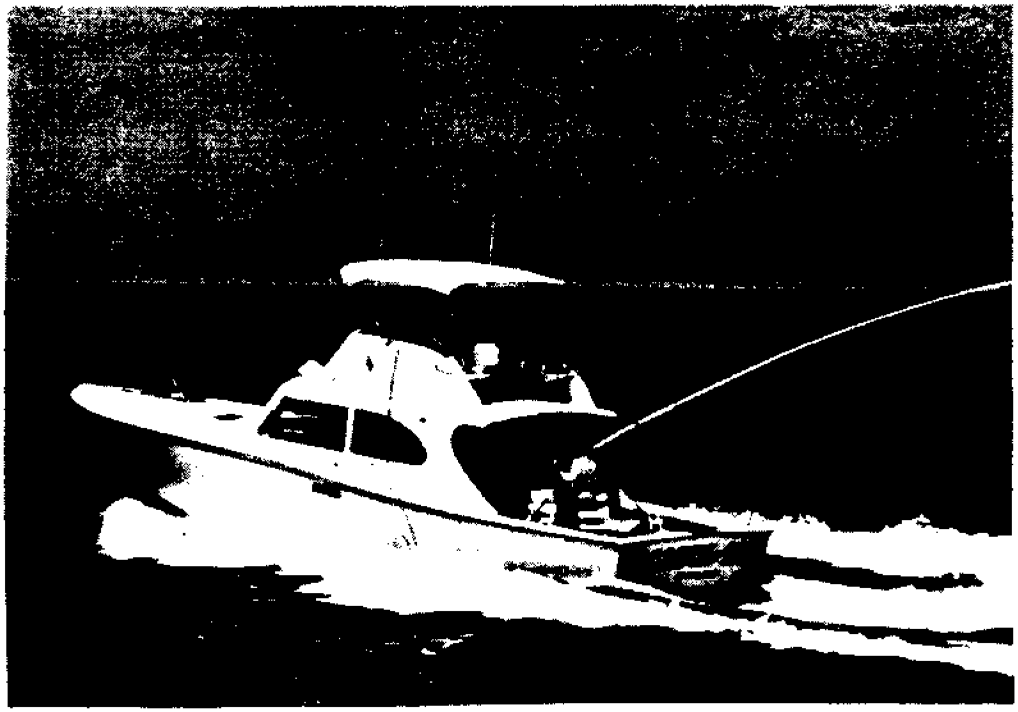
ELECTRICAL SYSTEM: 12-24-32 volt systems available. All circuits properly fused and wire correctly sized for load and length of run. All underwater metal parts bonded.

THRU HULL FITTINGS: Bronze with bronze gate valves if below water line and acetal resin U.L. approved if above water line.

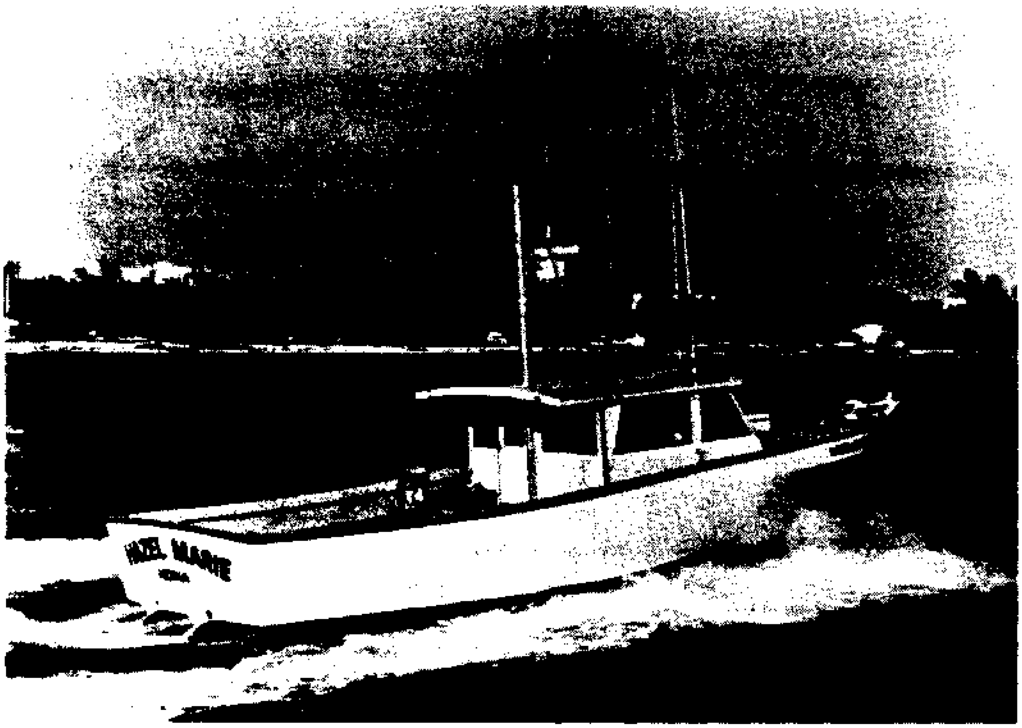
LIGHTS: Navigation lights in accordance with International Rules; cabin and deck lights to buyer's requirements.



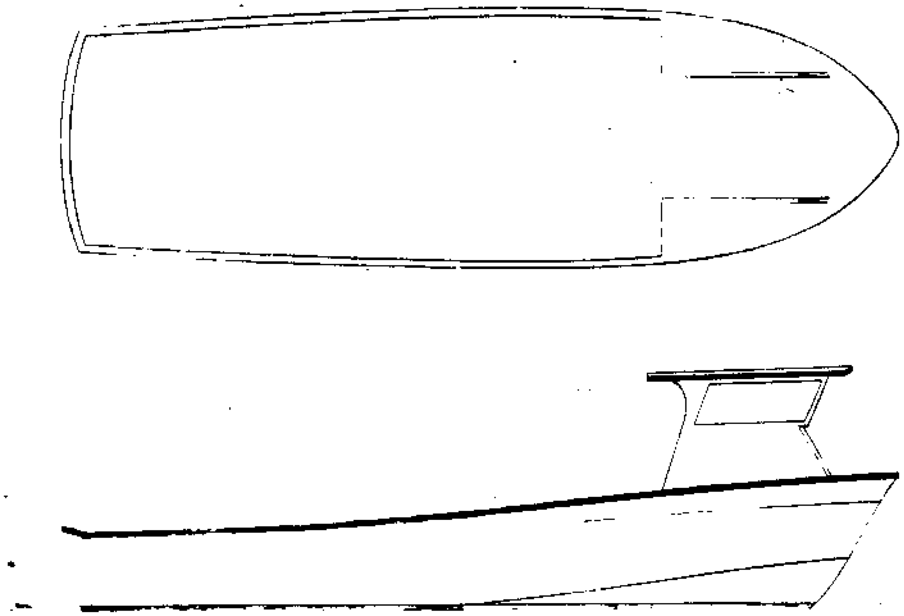
48' DIVE BOAT - FLORIDA KEYS



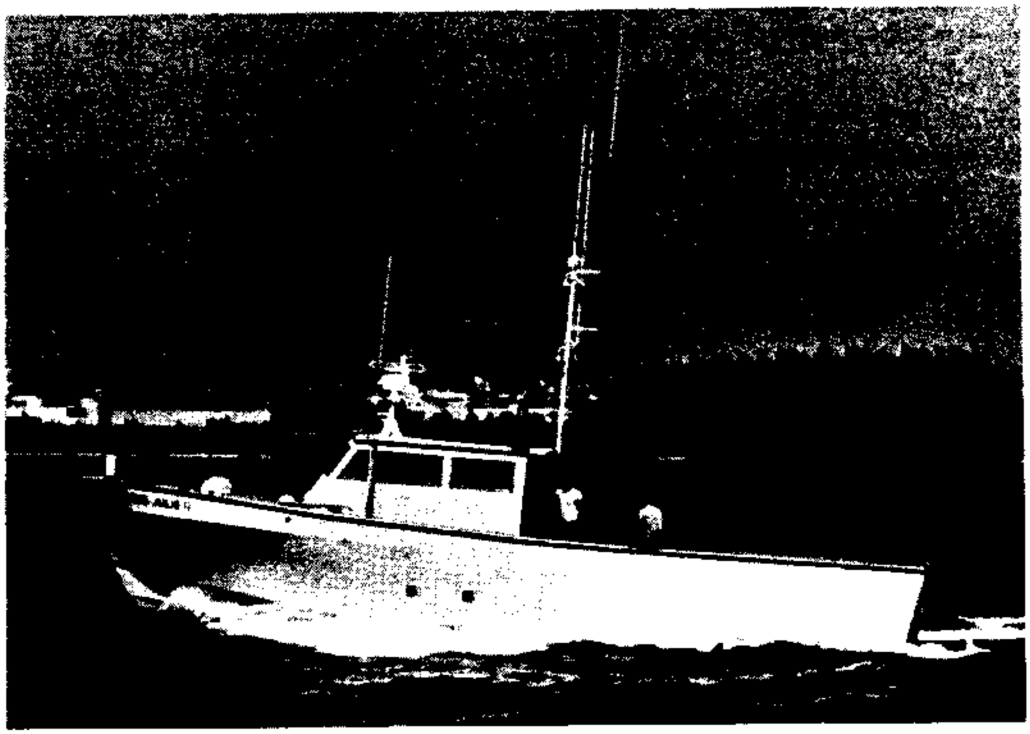
48' CHARTER SPORTS FISHERMAN - FLORIDA



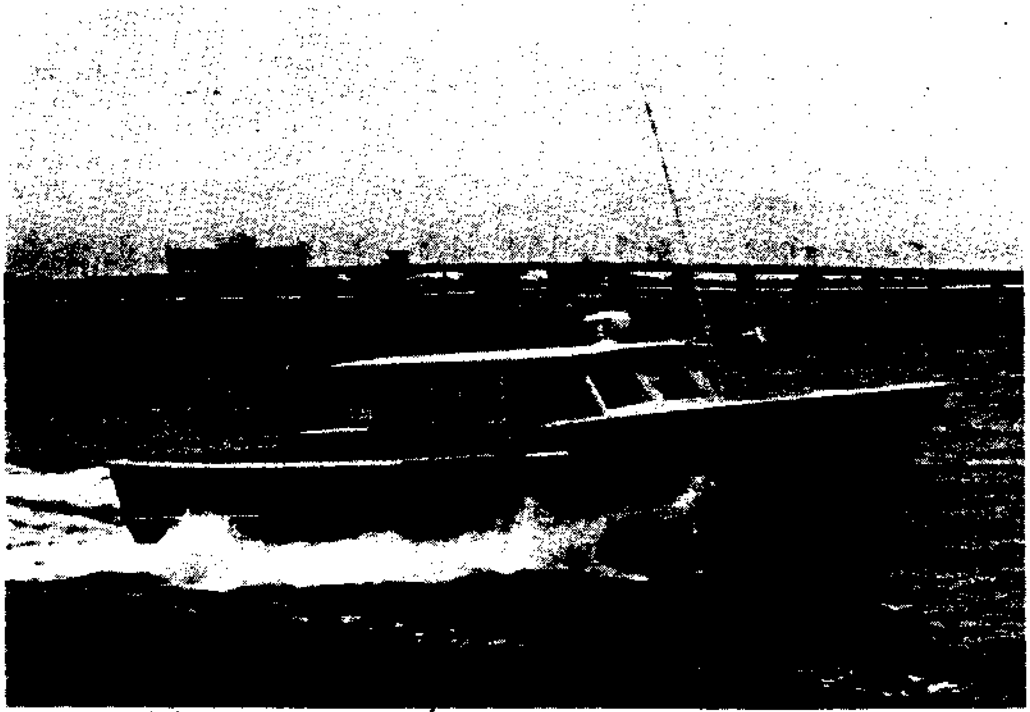
48' TUNA BOAT - HAWAII



48' STONE CRAB/GROUPER BOAT -S.W. FLORIDA



48' OFFSHORE LOBSTER BOAT - MASSACHUSETTS



46' [illegible] TRAWLER/SCALLOPER - MASSACHUSETTS

ENERGY SAVING AND RIG DEVELOPMENT FOR ARTISANAL FISHING BOATS

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1. INTRODUCTION

Over the last 20 years considerable progress has been made in mechanising artisanal fishing boats by fitting diesel engines or outboard motors. This has increased their fishing capacity, reduced labour and improved safety.

Unfortunately the value of many of these improvements has now been overtaken by the increased price of oil, which may sometimes represent half of the total cost of the fishing operation. Already in many areas fishing boats cannot afford to go to sea unless they can be assured of an unusually good catch. This problem will increase in severity, particularly in those countries without their own oil resources.

Artisanal fishermen contribute approximately one half of the world's catch, with larger proportions being common in tropical areas, and they are potentially highly economic. It is therefore vitally important that the energy efficiency of artisanal boats should be increased.

Nearly all artisanal boats were at one time sail propelled and many still are. Generally, however, the rigs used gave a performance greatly inferior to that achieved with mechanisation, so until fuel oil prices increased, they declined in use. At best, however, most traditional rigs are usually laborious and relatively inefficient though they are, in the main, capable of improvement.

In response to these changes Giffords have been involved for many years in the development of low cost, fuel efficient boats for beach fisheries. During this time many different rig and propulsion engine configurations have been evaluated, culminating in a year long study undertaken for the Commission for the European Communities (CEC) entitled "Feasibility Study for Energy Saving in Artisanal Fisheries". This work involved an experimental investigation of the performance of the hull appendages needed for good sailing performance, careful measurements of the thrust and fuel consumption of a range of propulsion systems and a direct comparison of the performance of four different sailing rigs. The first stage was completed by August 1983.

In parallel with this CEC work two practical projects were undertaken on artisanal fishing boats in South India and Sri Lanka.

2. EARLY RIG EXPERIENCE

In a series of different artisanal boat development projects Giffords have fitted or specified a wide variety of different rigs on both single and double hulled craft. The particular choice of rig was generally made taking into account the existing local rigs and availability of materials. In no case was it possible to obtain much of a measure of the absolute performance of the rigs, nor even their performance relative to other rigs.

Since performance is very far from being the only parameter by which the suitability of a rig should be judged, this limitation has not been too important though there was a growing feeling that some direct measure of rig performance would be a valuable aid to the selection process.

The various rig configurations tried included the Lug (Fig 1), Oru (Fig 2), Sprit (Fig 3), Lateen (Fig 4) and a form of Lateen/Gunter hybrid based on the traditional South Indian Kattumaran rig (Fig 5). Each of these rigs had its own particular practical advantages and disadvantages and with experience it has proved increasingly possible to specify the rig most appropriate for a particular boat. However the lack of knowledge of the actual performance of each rig remained a problem, so when the opportunity arose to make true comparative trials for the CEC study, it was a major step in rig development.

3. EXPERIMENTAL RESULTS FROM THE CEC STUDY

The overall aim of the CEC funded work was to work towards a standard methodology by which the fuel use of a particular operation could be predicted and the potential for savings assessed. The results discussed in this paper are the experimental studies of propulsion systems and sailing rigs, which were carried out to provide basic input data for the method.

These results are the foundations of a data base which can be used to perform the following tasks related to the analysis of fuel efficiency.

* Quantify the effects of changes in hull form and weight

* Quantify the effects of changes in operating speed

* Quantify the effects of using different types of propulsion system

* Selection of the most appropriate rig for sail power or sail assistance

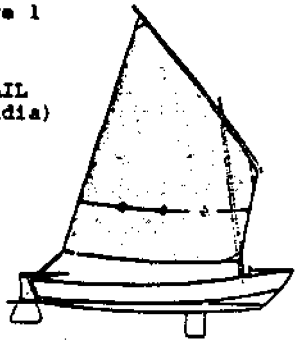
* Selection of the most appropriate appendage to give the required sailing performance

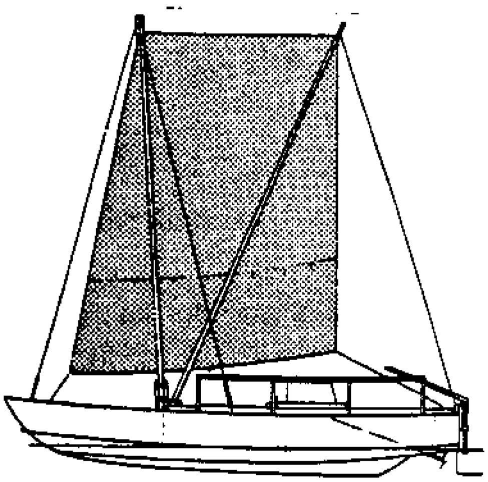
The combination of these results allows predictions to be made about the potential for fuel saving in existing craft and the potential for improvements through design changes to hulls and propulsion systems.

As they stand the methods are suitable for use by experienced people trained in boat design or naval architecture. However, with further refinement they will be presented in a form accessible and useful to non- specialists.

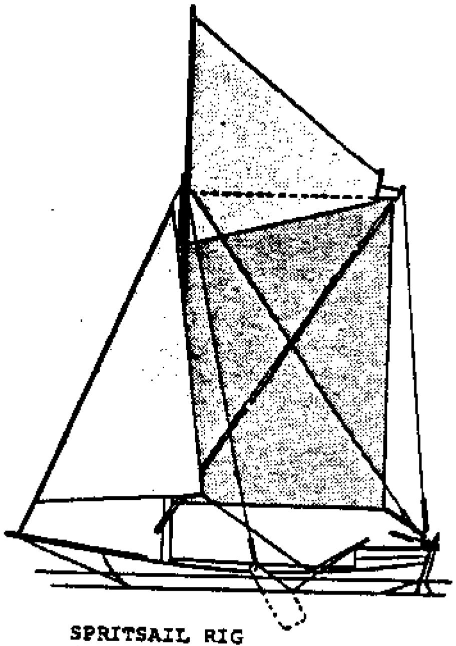
Figure 1

LUGSAIL
(S India)

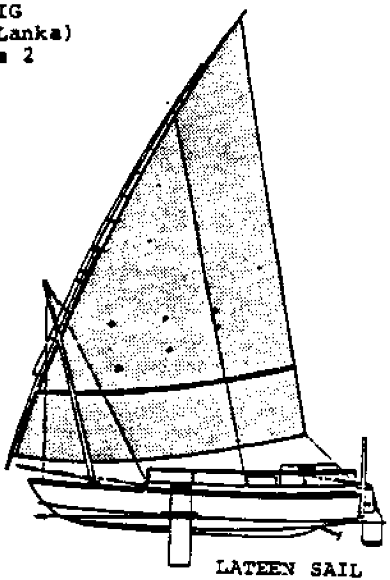




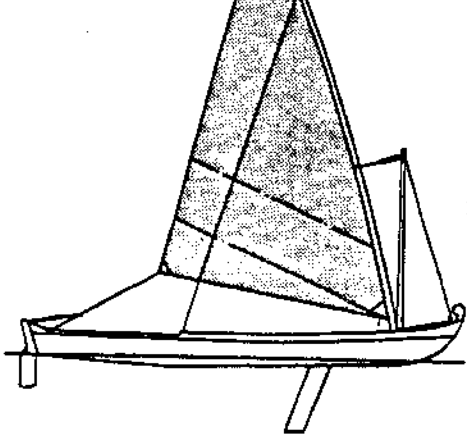
ORU RIG
(Sri Lanka)
Figure 2



SPRITSAIL RIG
(Bangladesh)
Figure 3



LATEEN SAIL
(Sri Lanka)
Figure 4



LATEEN/GUNTER
(S India)
Figure 5

SUMMARY OF ENGINE PROPELLER COMBINATIONS

Engine Type	Manu-factures Power (HP)	Fuel	Reduction Ratio	Propeller Size (cm) Diameter	Pitch
Outboard	8	Petrol	2.1:1	22	13
			2.1:1	22	18
Outboard	5	Petrol	3:1	Manufacturer's Standard Unit	
Outboard	25	Petrol	2:1	24	1
			2:1	24	2
Air-Cooled Diesel	12.5	Diesel	1:1	22	1
			2:1	38	2
			3:1	47	2

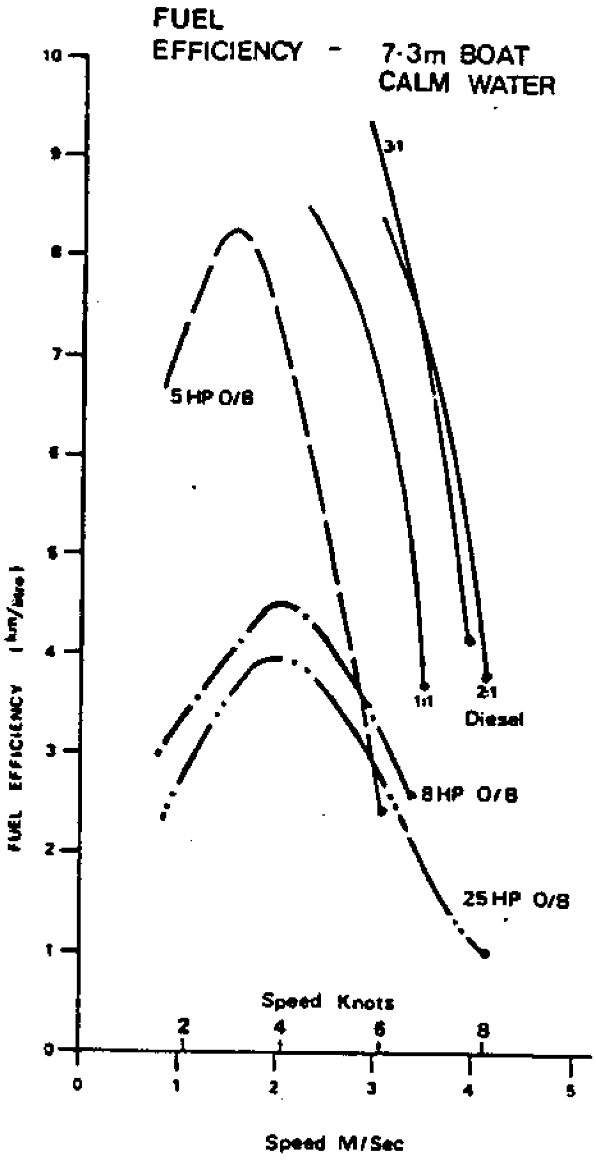
ANALYSIS OF ENGINE EFFICIENCY

Engine	Rated Power	Max Speed m/sec	Overall Propulsive Efficiency η_o	Estimated Propeller Efficiency (by Calculation) η_p	+η_T η_o/η_p	Fuel Efficiency km/litre
Outboard	5	3.1	0.44	0.56	0.79	2.4
Outboard	8	3.3	0.37	0.48	0.77	2.6
Outboard	25	4.1	0.28	0.43	0.65	1.0
Diesel						
1:1	12.5	3.6	0.42	0.48	0.88	3.7
2:1	12.5	4.1	0.56	0.57	0.95	4.2
3:1	12.5	4.0	0.52	0.62	0.84*	3.8

* This propeller did not absorb full power at maximum engine speed unde the condition tested. With more load (from greater hull resistance) i would have maintained the same speed, resulting in better transpor efficiency and a higher transmission efficiency.

+ This figure is calculated as the ratio of measured thrust horse powe to the manufacturers quoted engine power.

Figure 7



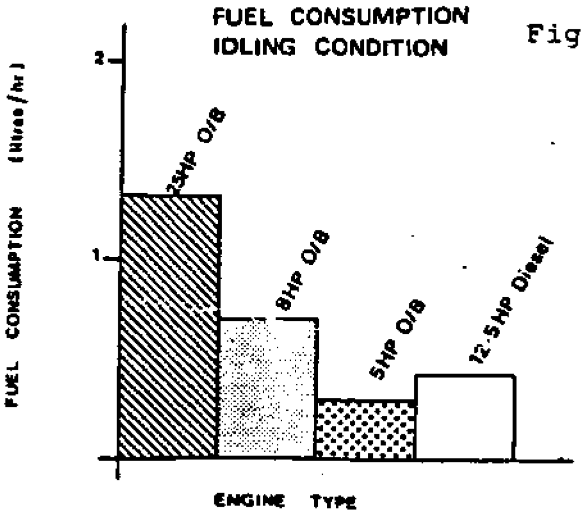


Figure 8

At maximum speed the fuel efficiency varies considerably, the best diesel configuration being over four times more fuel efficient than the 25 hp outboard. The small outboards fall between the two.

Note also how the overall efficiency of the outboards is significantly less than the diesel, and how the ratio of overall to propeller efficiency is much better for the diesel. This ratio is an indication of the reliability of the manufacturer's rated power figures and also of the transmission efficiency. For the outboards the mean figure is approximately 75% and for the diesel in excess of 90%. This difference most likely reflects the greater reliability of the power rating of the diesel engine, rather than great differences in transmission losses.

The variation of speed between the different engines somewhat confuses the comparison of fuel efficiency. At constant speed the relative ranking becomes more apparent:

Engine	FUEL EFFICIENCY (Km/l)		
	2.0m/sec	3.0m/sec	4.0m/sec
5 hp outboard	7.2	2.9	-
8 hp outboard	4.5	3.4	-
25 hp outboard	4.0	2.9	1.2
12.5 hp diesel	12.0	8.5	4.8

The marked superiority of the diesel is obvious from these figures, as is the very poor performance of the 25 hp outboard.

The 5 hp outboard gives good fuel efficiency at low power settings, but deteriorates rapidly as full power is approached.

Figure 8 shows how the idling (tick over) fuel flow varies between the engines. In this case the diesel is not the best, but ranks second after the small outboard. On a specific basis, i.e. fuel flow per horse power, the diesel comes out best by a considerable margin.

Figure 9 shows the thrust and fuel consumption for full power settings in the bollard pull conditions. The diesel and the 25 hp outboard produce comparable levels of thrust, with the 3:1 reduction ratio giving the diesel overall superiority. The fuel consumption picture is rather different. The 25 hp outboard uses almost three times as much fuel as the diesel. Thus for a given amount of fuel the outboard will only produce one third of the thrust produced by the diesel. This is illustrated in Figure 10, where the vertical axis is thrust/fuel flow, i.e. the thrust produced per unit of fuel consumed. This shows how the large outboard is out performed by both the smaller outboards and the diesel.

A way of approaching this result is to combine it with the top chart in Figure 9. This shows that the total thrust produced by the 25 hp outboard is three times greater than the smaller outboard.

However, the thrust fuel efficiency as shown in Figure 10 is much poorer for the 25 hp motor, hence for the same thrust it would be more fuel efficient to use three 5 hp motors than one 25 hp motor.

BOLLARD PULL CONDITION

Figure 9

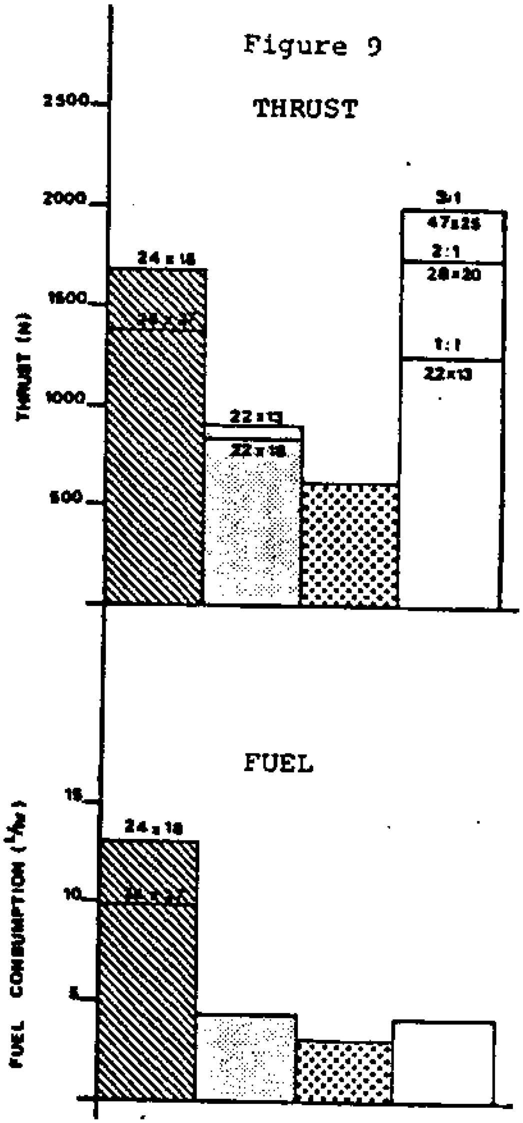
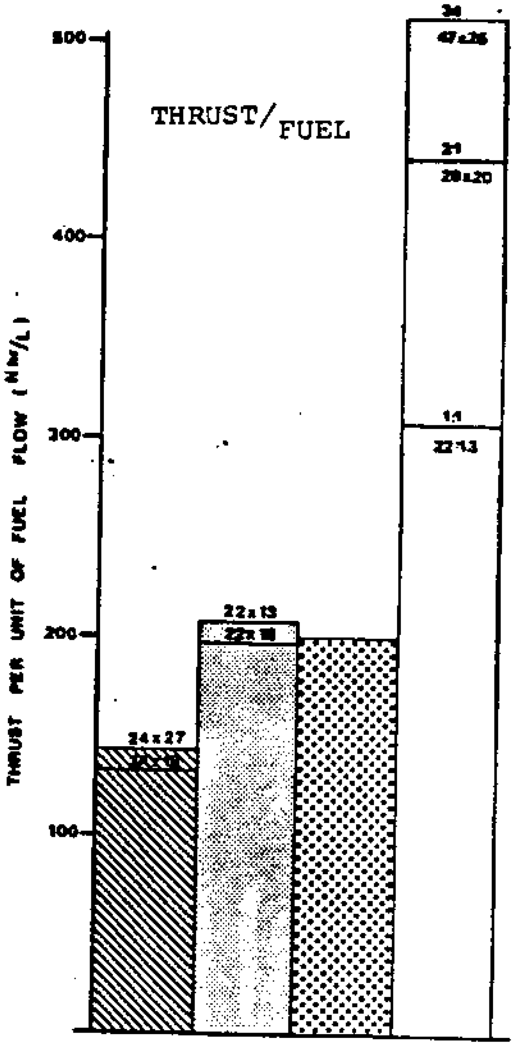


Figure 10

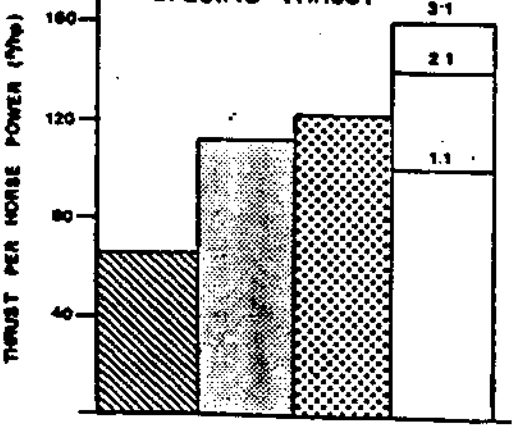


KEY

- ▨ 25HP O/B
- ░ 8HP O/B
- ▒ 5HP O/B
- □ 12.5HP DIESEL

Figure 11

BOLLARD PULL SPECIFIC THRUST



Also of relevance is the specific thrust, or thrust per unit of rated power. This is equivalent to propulsive efficiency in the free running case.

Figure 11 shows the result for the four engines. The diesel (at least with 2:1 and 3:1 reduction) comes out best, followed by the 5hp outboard. The 25 hp outboard again turns in the poorest performance.

3.2 Trials Under Sail

Two distinctly different types of sailing trials were conducted, one to establish the absolute sailing performance of a particular boat/rig combination and the other the relative performance of two different rigs.

For the trials two identical boats were provided. Both were 6m Sandhopper double hull boats, ballasted to the same displacement of 0.76 tonnes. The boats were kept ashore between trials to minimise variation due to water uptake and marine growth. The double-hulled form was chosen partly because of the convenience of deck space and stability but also to eliminate the heeling component for simplicity of comparison of results.

One boat was permanently fitted with a Bermudian rig. The geometry chosen was representative of the type that might be most suited to working sailing boats. Its aspect ratio was moderate and a minimum of control lines were provided.

The other boat was fitted with three different rigs, all of the same total area as the Bermudian (18.6m^2). The four rigs are illustrated in Figure 12. They were representative of types that are used on artisanal boats in different parts of the world, namely:-

Lateen Rig
Spritsail Rig
Gaff Rig

The absolute sailing performance of the boat with the Bermudian rig was determined first. This proved to be a very time consuming exercise, which even so only produced rather scattered results. This difficulty of obtaining good absolute performance results accords well with the reports of other workers. Figure 13 shows the polar performance curve plotted as a ratio of boat speed to wind speed. The curve drawn is based on a knowledge of the points thought to be most reliable, but has a considerable subjective element in it.

In practice the absolute performance is not necessarily of great importance since it is only valid for one particular combination of hull and rig.

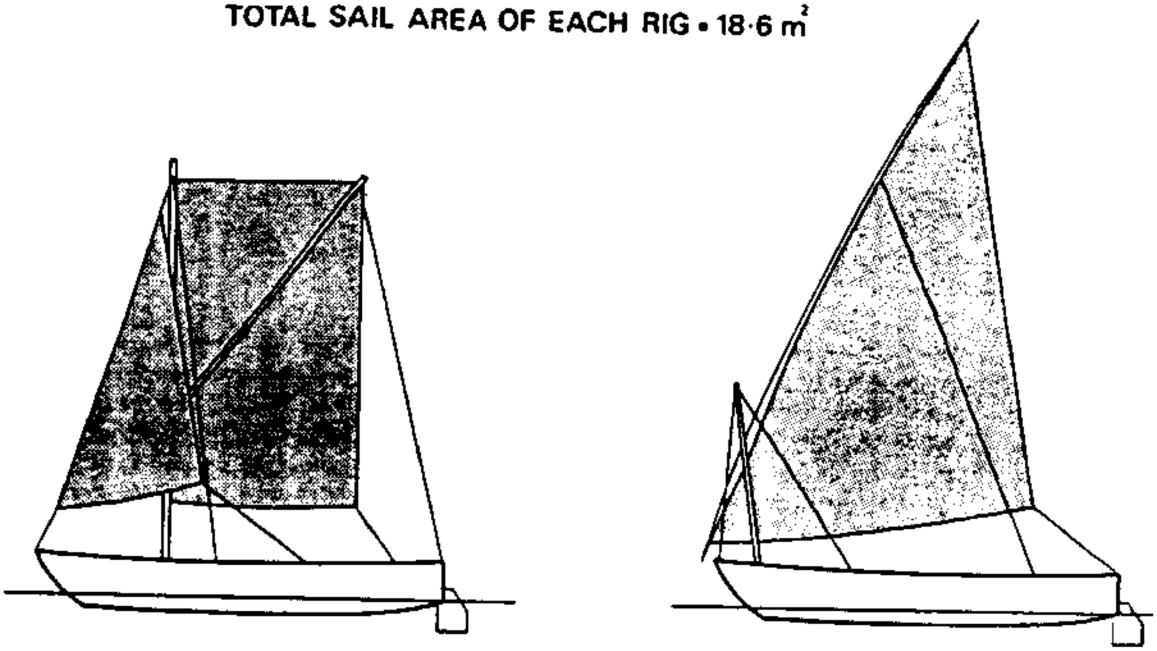
What is really of interest is the relative performance of different rigs. This information is sufficient to assist in decision making about suitable rigs, and is much more easily and reliably obtained.

For comparative sailing trials it is not necessary to make such precise measurements of the environmental conditions as for absolute trials. However it is necessary to have two identical boats, whereas absolute trials can be conducted with just one boat.

Figure 12

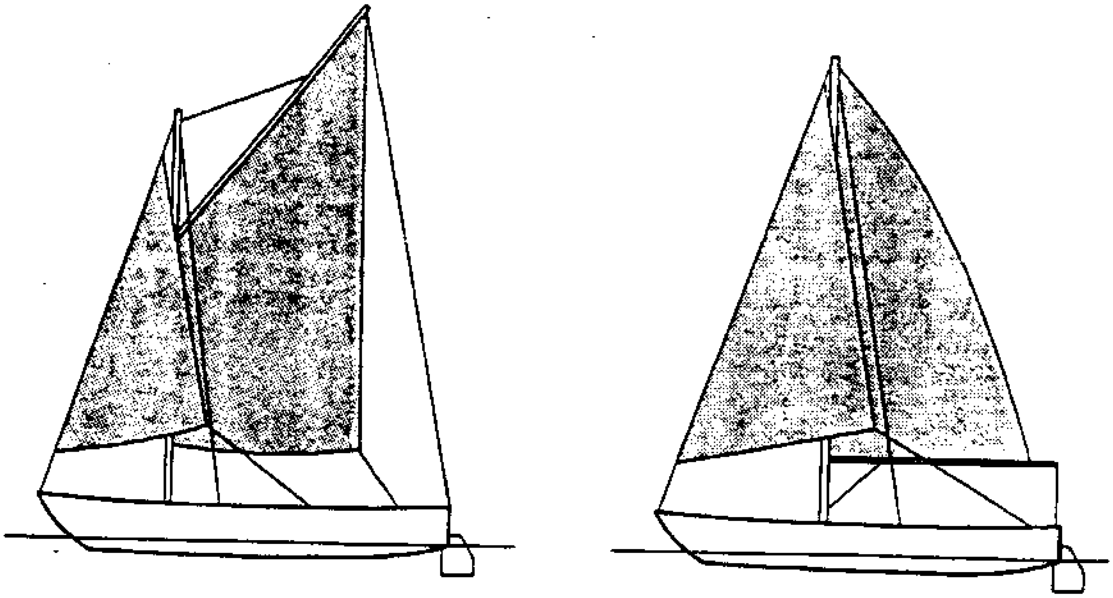
RIGS USED IN COMPARATIVE SAILING TRIALS

TOTAL SAIL AREA OF EACH RIG • 18·6 m²



SPRIT

LATEEN

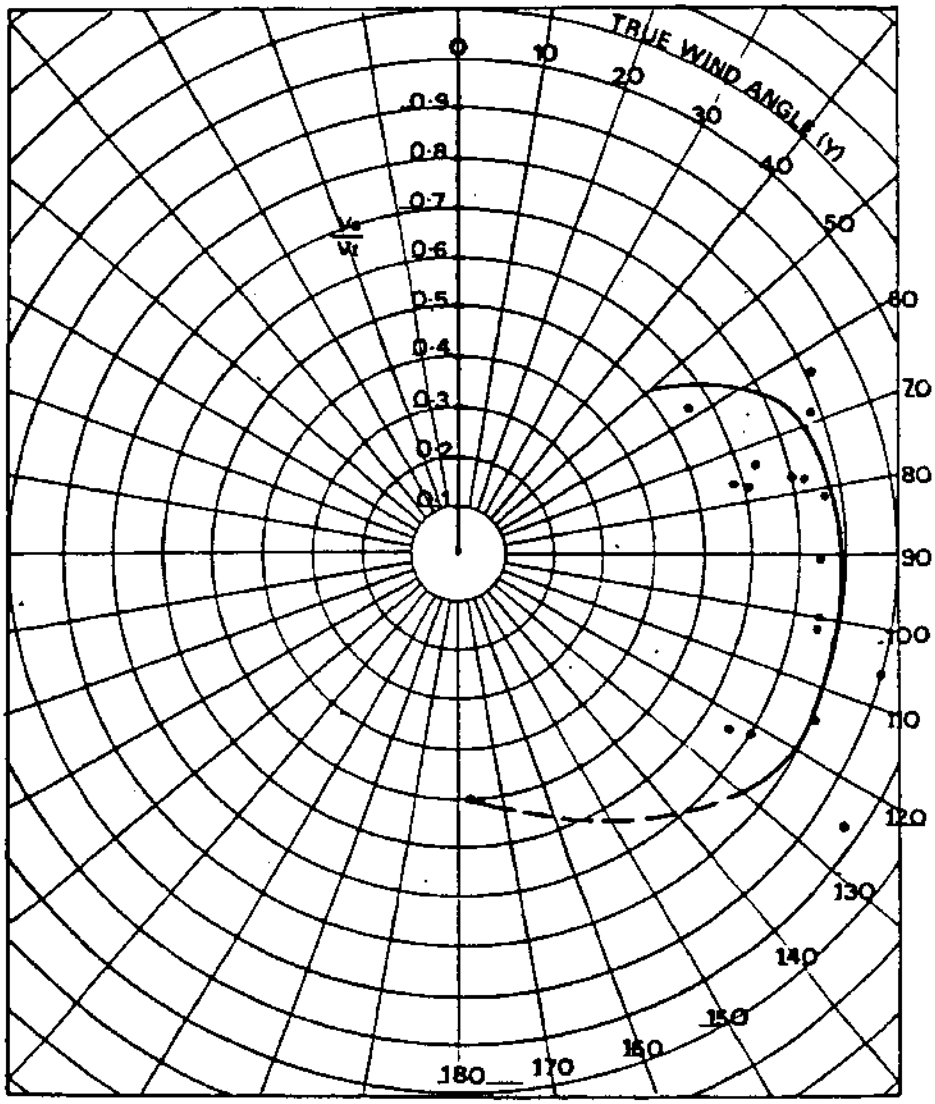


GAFF

BERMUDIAN

Figure 13

POLAR PERFORMANCE
BERMUDAN RIG



Provided two identical boats are available comparative trials can be carried out quickly and economically. They are less sensitive to variations in wind speed and direction and can be carried out over long distances to give good time averaged results. The effects of tidal streams are also much less significant than for absolute trials.

The main requirement is that a variety of courses of known distance can be sailed at different angles to the wind. A series of courses set around fixed marks will quickly build up a representative range of headings. For running the trials the following procedure was adopted.

> A sequence of courses was predetermined once the boats were launched.
>
> Both boats grouped close to the first mark.
>
> The one expected to be faster set off along the course, followed by the other boat.
>
> The time was noted as each boat passed the first mark, then the boats were sailed as well as possible along the course noting the time as each passed the end marker.
>
> Care was taken to ensure that the boats did not interfere with each other, and when tacking was required, the boats made long tacks and agreed a common moment to go about, so minimising the effects of wind shifts.

Using this technique the performance of the three artisanal rigs was established relative to the Bermudian rig.

The results are shown in Figure 14, and were at first sight very unexpected. Both the sprit rig and the gaff rig are superior on all points of sailing to the Bermudian, with the superiority being particularly marked when sailing to windward. In very light winds the lateen rig is comparable to the Bermudian, but as wind speed increased it became inferior due to excessive flexing of the yard.

The normalised presentation of Figure 14 may be difficult to interpret. To illustrate its implications the absolute polar performance curve for the Bermudian rig has been used as a basis for the true comparative presentation of Figure 15.

The shapes of the curves are considered reliable for true wind angles from 70° around to 180°. From 70° to 0° the absolute data is sparse, so the curve is drawn with a considerable element of judgement. However it still serves the present purpose, since the relative performances of the rigs were reliably established between 0° and 70° (as well as 70° to 180°).

One of the overriding impressions from the trials, subsequently substantiated by analysis, was the marked superiority of the spritsail rig when sailing to windward. The sprit boat pointed higher to the wind than the Bermudian and was faster through the water.

This result is totally at odds with conventional wisdom, so might be considered suspect. However some reflection shows that it is not as unlikely as it appears since:-

RELATIVE SAILING PERFORMANCE

COMPARISON OF ARTISANAL RIGS WITH BERMUDAN

(AT EACH HEADING BERMUDAN PERFORMANCE NORMALISED TO 100%)

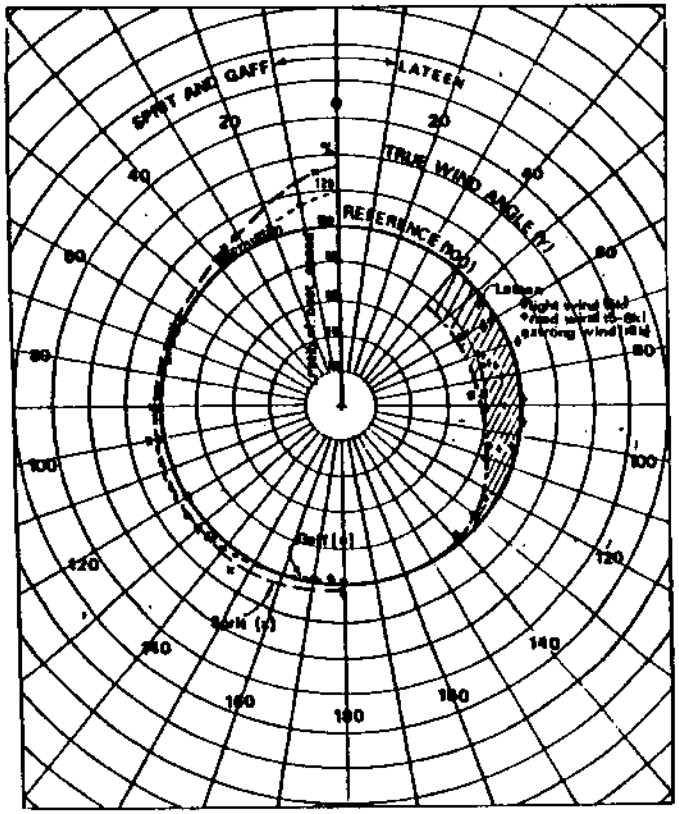


Figure 14

COMPARISON OF SAILING PERFORMANCE
BERMUDAN -LATEEN-SPRIT -GAFF
SANDHOPPER HULL.WIND SPEED RANGE 5-10kts
ALL AT EQUAL AREA OF 18·6m²

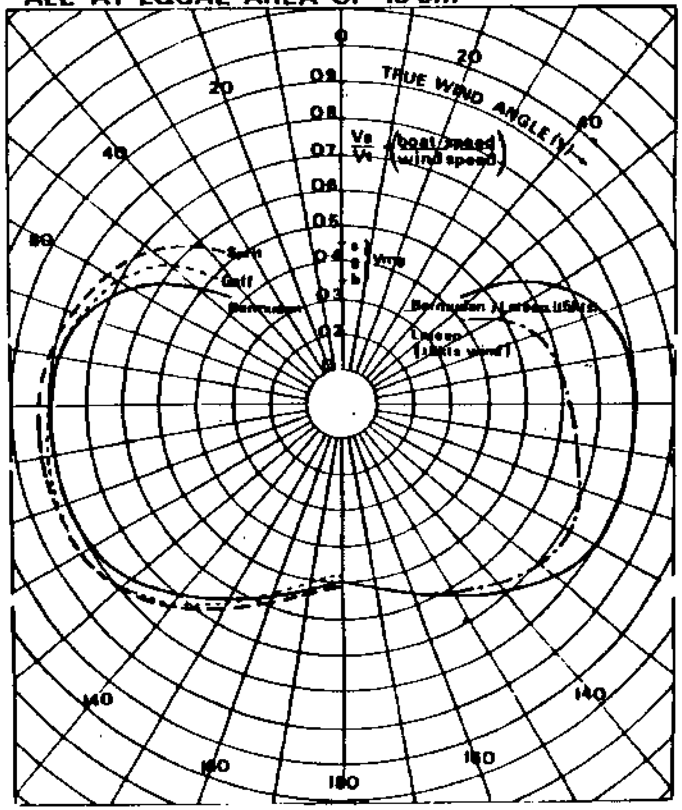


Figure 15

The sprit rig was fitted with vangs, so could be sailed free of twist.

The sprit rig mainsail was more full than the Bermudian (a beneficial feature in the light to moderate winds of the trials).

The rectangular plan form of the sprit rig is aerodynamically superior to the triangular shape of the Bermudian.

The performance of the lateen rig deserves some comment. It was the most difficult rig to control and in wind strengths much above 5 knots it tended to distort and become inefficient. In light winds it provided a performance comparable to the Bermudian, but as wind speed increased towards 8 to 10 knots it became considerably worse.

This deterioration is only considered to be partially an inherent feature of that rig. With suitable development, and perhaps improved sailing skills it is anticipated that it could turn in a much better performance. Already since the trials, improvements have been noted following the fitting of a stiffer yard.

However being a single sail rig it may well never be quite as good as jib/mainsail rigs which can exploit the extra drive which arises from the slot effect of two close coupled sails.

It should however compare favourably with Genoa and cat type Bermudian rigs which are commonly proposed for working sailing boats. In such a comparison its greater versatility and familiarity with tropical fishermen may give it a practical advantage.

4. PRACTICAL EXAMPLES

Sandskipper in Sri Lanka

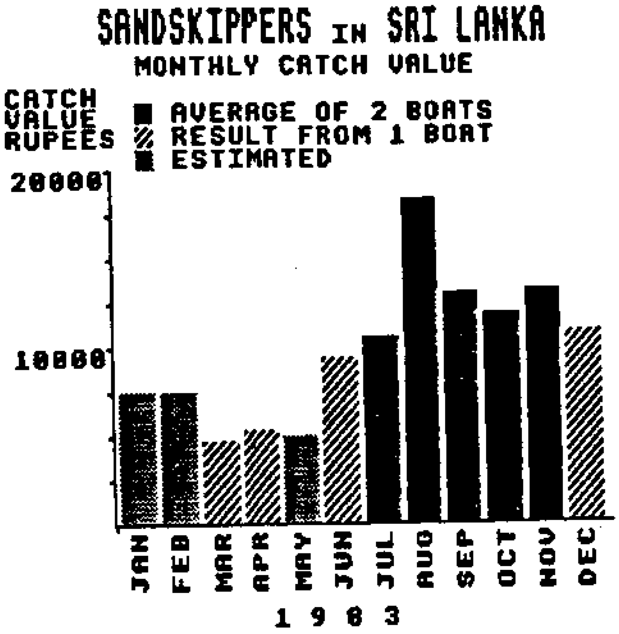
For the past year, two Sandskipper 24 double hulled boats have been fishing from Wadduwa, a small village South of Colombo (on the West coast of Sri Lanka). This is an ITIS funded project, introducing the boats to a fishery currently exploited by small, rowed orus and 28' harbour based mechanised boats. The orus, which operate from the beach are restricted in the surf size they can manage and the 28' boats can only be operated from harbours. They are heavy monohull boats designed before fuel efficiency became a major consideration.

The Sandskippers operate from the beach and are capable of negotiating larger surf than the orus, though there are still a few days per year when they have to stay ashore. So far they have been very succussful, providing an excellent economic return which is enabling the local skippers to buy the boats.

The boats are fitted with 34m^2 lateen sails, which are used voluntarily by the fishermen as a fuel saving measure. The double hull configuration gives a good sail carrying power, so on reaching courses the sails give as much, or more, speed than the small Petter Diesels fitted to the boats.

One year's operational experience has shown that the 24' boats are a little too small to carry as many nets as the fishermen would like and to negotiate the worst of the monsoon surf. Accordingly a 28' version has been designed and is currently under construction. It will be built in GRP (a favoured material in Sri Lanka) and fitted with a 12.5hp Deutz diesel with 3:1 reduction ratio for good propulsive efficiency.

Figure 16



SANDSKIPPER 28 in SRI LANKA

PREDICTED ECONOMIC PERFORMANCE

ENGINE:12.5HP [illegible]

R/G 37sq.m LATEEN

NETS:30 UNITS ,120 [illegible]

REVENUE DERIVED FROM [illegible] DATA

MEDIAN VALUE FROM 4 BOATS

TABLE 1

DISCOUNTED CASH FLOW ANALYSIS [illegible] and PARTNERS 1984

	YEAR 1	2	3	4	5	6	7	8	9
CAPITAL COSTS									
Boat+sail	[illegible]	.00	.00	.00	.00	.00	.00	.00	.00
Engine	25000.00	.00	.00	.00	25000.00	.00	.00	.00	.00
Nets	64000.00	.00	.00	.00	64000.00	.00	.00	.00	.00
Equipment	3000.00	.00	.00	.00	.00	.00	.00	.00	.00
Beach winch	[illegible]	1500.00	1500.00	1500.00	[illegible]	1500.00	1500.00	1500.00	.00
Sail replacements	.00	.00	2000.00	.00	[illegible]	.00	2000.00	.00	.00
TOTAL	206500.00	1500.00	3500.00	1500.00	[illegible]	1500.00	3500.00	1500.00	.00
OPERATING COSTS									
Fuel Oil	1875.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	625.00
Fuel rate	8.30	8.30	8.30	8.30	8.30	8.30	8.30	8.30	8.30
Fuel cost	15562.50	20750.00	20750.00	20750.00	20750.00	20750.00	20750.00	20750.00	5187.50
Consumables	10500.00	14000.00	14000.00	14000.00	14000.00	14000.00	14000.00	14000.00	3500.00
Labour	36000.00	48000.00	48000.00	48000.00	48000.00	48000.00	48000.00	48000.00	12000.00
Maintenance	7500.00	10000.00	10000.00	10000.00	10000.00	10000.00	10000.00	10000.00	2500.00
Beach fuel	994.00	1325.00	1325.00	1325.00	1325.00	1325.00	1325.00	1325.00	331.25
TOTAL	70556.50	94075.00	94075.00	94075.00	94075.00	94075.00	94075.00	94075.00	23518.75
TOTAL COSTS	277056.50	95575.00	97575.00	95575.00	[illegible]	95575.00	97575.00	95575.00	23518.75
REVENUE	161250.00	215000.00	215000.00	215000.00	215000.00	215000.00	215000.00	215000.00	53750.00
NETT CASH FLOW	-115806.50	119425.00	117425.00	119425.00	[illegible]	119425.00	117425.00	119425.00	30231.25
DISCOUNTED CASH FLOW	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
DISCOUNTED BALANCE									-218.79
INTERNAL RATE OF RETURN	[illegible]								

A sailing rig will be fitted, initially a 38m2 lateen, but this may well be eventually changed to the twin tunny rig or some other twin rig to improve ease of handling and increase the area which can be carried.

Based on the catch data recorded for the Sandskipper 24 operation, predictions have been made of the economic performance of the Sandskipper 28. Catch figures are available for the two Sandskippers for most of the past year and are shown in Figure 16. These results were obtained with an average of 17.5 sets of nets, whereas the SK28 will carry 30. Experience has shown that drift netting is essentially a statistical based technique, and on average more nets equals more fish, on a pro rata basis. Hence these figures are factored in the ratio of 30 : 17.5 to predict the revenue that will be earned by the SK28.

Using these figures in association with estimates of local building costs, crew costs, fuel etc. a full discounted cash flow analysis was carried out. It indicates an internal rate of return of 96%. Table 1 shows this result. To put this isolated figure into perspective various sensitivities of this result were investigated.

The main areas of interest were

Capital Costs
Revenue
Fuel Cost/Fuel Saving

The results are shown in the following figures:

Figure 17 Effect of Capital Cost and Revenue on Internal Rate of Return

The lower (solid) line shows how increasing the capital cost of the project reduces the rate of return in a non-linear fashion. Initially the effect is very marked, but as the rate of return reduces, so it becomes less sensitive to capital costs (i.e. the first 25% increase in capital cost reduces the IRR by more than $1/3$ whereas the increase from 75% to 100% reduces it less than $1/4$). The upper dotted line shows the effect of increased revenue - the line being moved bodily upwards but retaining the same characteristics as for the standard revenue.

Related to the influence of capital cost on internal rate of return is the effect of revenue changes.

Figure 18 Variation of IRR with Revenue and Fuel Use

The datum value for this analysis was not the standard condition, but a situation which gave a 56% IRR. It shows that the IRR is very sensitive to revenue changes - in the datum case a drop in revenue of 25% takes the operation from being very profitable to quite marginal (IRR from 56% down to 3%). Also shown on this figure is the effect of fuel consumption, as a percentage increase over the datum case.

The -25% case represents a reasonable target for the effect of sail assistance, whilst the +300% case approximates to the position of the existing heavy displacement 28' boats. To generate the same rate of return as the SK28 they would need to catch around 50% more fish with the same crew and nets and capital costs.

Figure 17

Effect of Capital Cost and Revenue on Internal Rate of Return

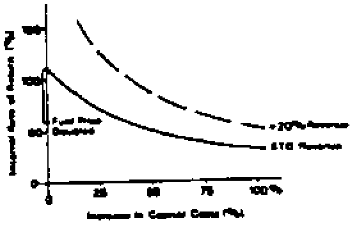


Figure 18

Variation of IRR with Revenue and Fuel Use

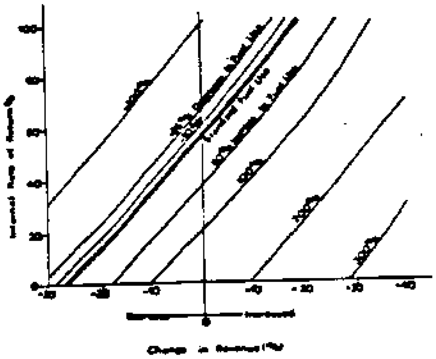


Figure 19

Variation of IRR with Fuel Cost

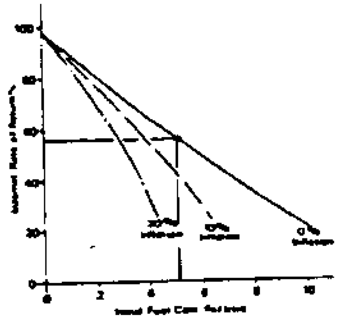


Figure 20

Capital Value of Fuel Saving

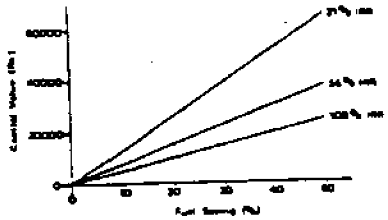
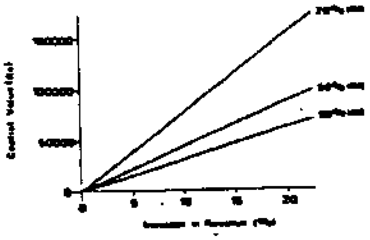


Figure 21

Capital Value of Increases in Revenue



Conversely the effect of 25% fuel saving on the SK28 operation is not large, since with the small, efficient diesel engine fuel costs are only a small fraction of total costs. Such a fuel saving is equivalent to a 5% change in catch (or revenue) or 11 percentage point change in IRR. As the price of fuel rises the effect of fuel saving becomes more marked.

Consider the line for the heavy displacement boat (+300% fuel). This is equivalent to a fourfold price rise at constant fuel consumption. The +200% line thus represents a 25% fuel saving at this high fuel price. In this case the effect of the 25% fuel saving is equivalent to an 18% change in revenue.

The other way of looking at this effect is that it corresponds to 22 percentage points increase in IRR - which is a big improvement by any standards.

The influence of fuel price on rate of return is shown more directly in the next figure.

Figure 19 Variation of IRR with Fuel Cost.

This again uses the 56% IRR case as the datum, with a fuel price of Rs 5.0/litre. The variation in fuel cost is interchangeable with variation in fuel consumption, i.e. a fuel price of Rs 10/litre has the same economic effect as twice the consumption of fuel at the datum rate of Rs 5/litre.

In addition to the basic relationship between fuel price and return, the other two lines shown illustrate the effect of fuel prices subject to inflation. The cases shown assume fuel price inflation rates of 10% and 20% per annum. It is interesting to note that starting with a fuel price of Rs 5/litre, if the fuel price inflation rate goes up to 20% per annum the internal rate of return approaches zero, (compared to 56% with no inflation).

Moving to the left of the picture (towards lower fuel price or lower consumption) it is seen that the results are far less sensitive to inflation. Since sail assistance is an effective method of fuel reduction its adoption provides a worthwhile defence against the effects of fuel price inflation. On todays prices the value of sail assistance may not seem to be great, but against the prospect of long term fuel price inflation it makes sound economic sense. If it is adopted now it will be second nature by the time that large gains can be made.
Economic projects of this nature contain a balance between initial capital investment and recurring costs. An oft quoted example is the choice between outboard motors and diesel engines. The outboard is cheap to buy, but expensive to run; the diesel expensive to buy but cheaper to run. Depending upon the price of money one or other can be shown to be the better option.

Similarly for a particular project it is possible to determine the relationship between a recurring cost feature and capital investment - establishing what is in effect the capital value of the change in the recurring cost.

Figure 20 Capital Value of Fuel Saving

This picture shows the capital value of varying degrees of fuel saving at three different rates of return. The lower the rate of return (or the longer the period required to recoup the initial investment) the greater the capital value of fuel saving.

Practically, one use of this picture is to look at the economic worth of different sail assist schemes. For example on a scheme with 56% IRR an investment of Rs 30,000 in a rig is justified if it leads to a fuel saving of 35%. On a longer term project up to Rs 50,000 can be justified against a 35% fuel saving.

Considering that the 38m^2 lateen rig is expected to cost Rs 2,000, it can be seen that it only has to produce a fuel saving of about 4% to earn its keep even on a project where an IRR of around 100% is expected.

Using this picture it is possible to build up a good feel for the economic potential of fuel saving measures, which is invaluable when making judgements in the design stage. Practically it is very difficult to make any hard and fast predicitions about the actual fuel saving that will result from sail assistance. In the end it relies a great deal on judgement and the experience of others, so any additional information such as this provides very useful extra input for evaluating different schemes.

In the same way that investment in fuel saving measures can be evaluated it is possible to look at the effect of investing in improving the revenue earning potential.

Figure 21 Capital Value of Increases in Revenue

The results show a similar pattern to those for fuel saving: increasing investment is justified as economic rate of return decreases.

Even for the highest rate of return shown, quite large investments would appear to be justified for relatively small increases in catch, e.g. at 98% IRR it is worth investing Rs 50,000 for a revenue increase of 15%. To put this into perspective the price of a complete set of 30 nets is Rs 4,000 and a new boat Rs 95,000.

One note of caution in interpreting these capital value results. The lines drawn apply to a once only investment and take no account of the life cycle costs of the equipment bought with the capital (eg maintenance costs and additional manning requirements). For the high rates of return the effect of this will be small, but for longer term investments it could become significant. The general result will be a reduction in the gradient of lines which becomes more significant as IRR reduces - so the spread of the lines will be compressed.

5. ECONOMIC CONCLUSIONS

With these results it is possible to get a good overall picture of the economics of the operation which can point the way to future developments.

It would be tempting to conclude that the primary need is additional capital investment in larger boats and more gear to ensure larger catches. Theoretically, this is undoubtedly the correct view, but practically it is unlikely to live up to expectations.

The fishermen for whom this project is run naturally form into certain sized groups, of three to five men. This immediately sets an upper limit on the net size that can be handled and influences the capital that can be raised for boat purchase. The boats operate from the beach, which sets limits on boat size and weight.

Consequently there are practical restraints on the capital investment, at least insofar as it might be thought best to put it into bigger boats or longer nets.

Instead, a better approach is to optimise the selected fishing units. Examine ways of making a fixed boat/crew/net combination more effective. Within this context the use of net haulers, fish finders and different patterns of operation might be considered.

With present fuel prices and the fuel efficient design of the boat considered, the value of fuel saving is not perhaps as attractive as measures to improve catch. For example Figure 18 indicates that a 50% increase in fuel use is justified by a 10% increase in revenue. Thus it may make sense to travel further to more lucrative fishing grounds. With these curves it is at least possible to get an idea of the trade-off that is appropriate.

One final observation on the capital value. The boat studied is a beach boat and hence limited in size. Larger catches may well be made by bigger more powerful harbour based boats - but do they justify the overhead cost of the harbour*

Assuming the relationships remain linear outside the range presented, a 50% increase in revenue is worth some Rs 250,000. A harbour for say 200 boats could well cost upwards of Rs 300 million; an investment of Rs 1.5 million per boat - i.e. stretching the results still further it is implied that a 300% increase in catch will be needed to justify the harbour± (How that will be physically achieved and whether the fish stocks are there to be caught for the lifetime of the harbour will be left for the planner of large scale schemes to contemplate).

6. THE FUTURE FOR SAIL ASSISTANCE

The preceding sections have provided some insights into the performance of different sailing rigs and the economic value of sail power and fuel efficient fishing boat design. There is no doubt that sail assistance can be economically justified, particularly in a world where fuel prices show every indication of being subject to price inflation.

What then are the factors which inhibit the adoption of sail assistance on artisanal fishing boats?

The following are perhaps the most significant

Fashion
Lack of Sailing Skills
Desire for Speed
Additional Labour Requirements.

Fashion - engines are modern; a sign of progress whilst sail is a sign of the past. Fishermen can be suspicious of suggestions that they return to the use of sail, seeing it as a retrograde step. Consequently they tend to be unsympathetic to the use of sail and thus unlikely to make the best of sail assist ideas.

Lack of Sailing Skills - In many places sail may never have been used or has long lapsed into disuse. The skills needed to handle sail are then not available.

Desire for Speed - Fishermen the world over like to go fast. In most cases sail power will not give as much speed as even a quite small motor, which greatly detracts from its appeal.

Additional Labour Requirements - Sails require additional time and effort to set and trim compared to the ease of using an engine (when it is working).

These are four important areas which must be tackled in sail power projects. The relative importance of each will vary from place to place, but always one of the major hurdles will be the great appeal of a shiny new engine compared to even the most modern looking sailing rig.

The appeal of the engine is not restricted to the fishermen who use it, but extends to those who can benefit from its supply. In a developing country engines tend to represent a great concentration of value and thus offer potential for manufacturers, importers and suppliers to make substantial profits. This increases the pressures for adopting mechanisation as it is much more difficult to make equivalent profits from the supply of sailing rigs. It means that the motives for the provision of engines are often quite unrelated to the real needs of the fishermen who will pay for them and try to make a living from them.

However the economic realities of fuel prices, engine maintenance and frequently limited unreliable engine life, can temper the immediate appeal of mechanisation, leaving the fishermen to view sail power with a more sympathetic eye. Then is the time to initiate sail assistance projects, with the aim of either total engine replacement or, more commonly, a combined use of sail and power to give the optimum economic mix of the flexibility of mechanised operation and the low cost of operation under sail.

6.1 Sail Power on Small Boats

A very large proportion of artisanal boats are small in size, of length up to no more than 10m. As boats become smaller they represent an increasingly difficult problem for effective sail assistance.

The sea is no respecter of size, so big and small boats experience the same wind and waves. Thus, for similar performance the smaller boat requires proportionately more power to over come the effects of wind and wave.

Big or small, boats are still handled by the same human beings. Speed is subject to inherent psychological influences which tend to result in small boats being driven much harder than larger ones.

Psychologically a speed of 10 knots is quite 'fast' for a fishing boat and 6 knots is 'slow'. For a 30m boat, 10 knots is hydrodynamically a moderate speed whilst 6 knots is a proportionately much higher speed for an 8m boat. An equivalent speed would be just over 5 knots, definitely 'slow'. Consequently there is a tendency for small boats to be overdriven; to carry proportionately more power than their larger sisters.

Thus there are two influences pushing up power levels in smaller boats, which means that proportionately more sail is required for effective sail assistance. Speed under sail depends primarily upon stability, or power to carry sail. This has the perverse characteristic of increasing with increasing vessel size. Small boats thus lose out both ways - they require proportionately more propulsive power yet have proportionately less power to carry sail.

Thus for effective sail assistance the smaller the boat the more that stability becomes the critical consideration. On light slender boats, dugout types and their derivatives, crew weight can have a powerful influence. To augment this, balance boards have, for example, long been a feature of the artisanal boats of India, allowing the crew weight to be exerted further from the centre line and hence increase the righting moment. In many places various forms of outrigger have evolved, allowing extra stability to be gained by both buoyant forces and the distribution of crew weight.

For high speeds under sail, outriggers or multi hulled configurations have long been established as the only practical solution. The superiority of this form is confirmed by the modern trends in the design of yachts for long distance 'open' races. In these circules the debate is about the relative merits of two hulls or three; the possibility of using a monohull configuration is not even thought to be worth considering.

Thus for smaller vessels in particular the multi-hulled or outrigger configuration has a great deal in its favour if sail assistance is required. Indeed in the long term it is difficult to see small monohull boats making successful use of sail power where multi-bulled craft could be used instead.

CHANGE ON THE CHESAPEAKE

The Influence of Economic, Political, Legal and Management Factors on Small Scale Fishing Operations in Maryland's Chesapeake Bay

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ABSTRACT:

Since colonial times Maryland watermen have evolved distinctive craft capable of carrying out their tasks in particular geographical locations. For more than 150 years legislation has frequently been utilized, at the urging of local watermen, to preserve traditional fishing communities by an apportionment of available resources and restrictions on fishing methods and gear. Court decisions have profoundly affected management of the resources and these, as well as worldwide shifts in economic and energy matters, have caused the small scale waterman to adapt to these changes. While existing in a highly regulated environment, watermen have made many changes in the design, size, and powering of their vessels. An overview of the evolution of Chesapeake Bay area vessels is described. Changes in fishing craft, equipment, and operations are related to economic, legal, political, and management factors.

The Bay

The Chesapeake Bay is the largest estuary in the United States and one of the largest in the world. It includes about 4,000 miles of shoreline in the states of Maryland and Virginia and has a length of approximately 200 miles and a width that varies from 4 to 30 miles. It is, to be precise, the drowned river valley of the Susquehanna, which provides the Bay with about half of its freshwater inflow. It is an excellent place for species such as oysters, crabs, and fish to spawn and live and is one of the earliest settled areas in the country. It was only natural, then, for the colonists to begin to utilize the seafood found there and to evolve craft suitable to havest them in the conditions prevalent.

Early Inhabitants

Many of the early settlers in the Chesapeake Bay region came from England. Today, although most do not realize it, many of the items found in this area in terms of harvesting equipment and other features, can be traced back to those found on the southern coast of England. The waterman's crab shanty is similar to the "croglofft". The nippers and hand tongs are similar to those found there. The cylindrical eel pot used today is a variation on early pots woven of strips of split oak. Many of the dialects found in the Bay area are of Cornish origin.

Vessel Evolution

The earliest vessels found in the Bay were the log canoes built by the Indians. These were made by burning and scraping logs to produce a serviceable boat enabling the Indians to reach oysters, crabs, and fish with which to feed their tribes. The early colonists used vessels of this type and gradually improved on the design by increasing the size with the addition of logs to form larger craft capable of carrying increased loads. Canoes of two, three, and five logs became commonplace. These vessels were not only used for harvesting but for transportation since there existed no road network connecting most of the Bay communities. Although most of the remaining log canoes today are owned by sportsmen who use them to race during the summer months, there are some which have remained, fitted with engines and small cabins, in the work fleet.

Fishery Expansion

The most rapid development of the Bay fisheries began after the War of 1812 when New England schooners, having decimated many of their local oyster beds, began to come to the region to harvest from the abundant Chesapeake beds. They brought with them the oyster dredge and carried large numbers of oysters with them as they sailed back to their northern states. First they harvested large, mature oysters and later the small seed stock with which to replant the areas which they had depleted. Chesapeake watermen began to agitate for exclusion of the outsiders, even as many enlarged their boats and added dredges to compete with them in a race to see who would harvest the last oyster. With the competition came new designs in the Chesapeake craft, many of which were adapted from the New Englanders' schooners. The schooner, sloop, and pungy (named, in all likelikhood after the boatbuilding town of Pungoteague in Virginia) all had disadvantages in being able to dredge for oysters however, either the bulwarks were too high to efficiently pull the dredges or the draft too deep for the areas in which oysters were most prevalent. The log canoe became the double masted brogan which in turn evolved into the bugeye. Log boats were less expensive to build than the framed vessels of that period.

With the decline of many of the larger trees came the development of the deadrise bateau or skipjack which is still in use today. Built as a cheap workboat, it is amazing that many of these vessels have survived into this century. Most of the remaining fleet was built around the turn of the century.

By 1880 over 2,000 larger dredging vessels and 6,856 sailing canoes and other small craft were plying their trade on the Bay. Many of the local ports were thriving and had built up by the migration of many of the New England dealers moving to the region because of changes in the Maryland laws which forbade non-residents from harvesting local products. The conflicts between the tongers working from their smaller vessels and the dredgers working from their larger craft escalated, many times into pitched battles with death and destruction occurring. It was during this period that the Marylagd Oyster Navy was formed which survives to this day as the Marine Division of the Natural Resources Police. There also exists the stories of the "outlaw drudgers" and "arster pirates" which have been passed down throughout the years.

Many Bay craft were designed to meet specific differences in fisheries or geographic location. Since the bay is shallow, shoal draft boats were necessary.

The sailing vessels which evolved to become most efficient were built with centerboards and low sides. They were used during the summer months to haul farm products to market and to transport lumber to the cities.

Power craft used for tonging and crabbing had no fish hold and a raised engine box. A small cabin was usually located forward with just sufficient space to afford minimal protection from the cold winter weather while oystering.

Current Fishery

Today, about 6,000 individuals make their living as watermen in Maryland. The fisheries have remained much the same over the years due to pressure from watermen's groups to restrict efficiency in the harvesting of seafood. The Maryland fisheries are some of the most highly regulated in the United States in terms of harvesting equipment, quotas, and seasons. While relying upon these traditional harvesting methods, watermen have become influenced by outside factors. An extensive road network now connects all ports and towns. The construction of a bridge across the Bay at Annapolis has made easier access between shores and, consequently, markets. New construction materials such as fiberglass are replacing the traditional wood vessels. Electronic and hydraulic equipment have made some fisheries easier to participate in.

Interestingly, in spite of restrictions placed upon how one might harvest, little was done to actually regulate how much was harvested. The result was that overharvesting did occur despite legislated inefficiency, and, at the present time, Maryland is in the rather incongruous position of marking the 350th Anniversary of the founding of the state with the worst oyster harvest in recorded history.

Other influences such as legal and economic ones have likewise influenced Bay fishing operations.

Legal Factors

Until 1971, watermen were legally allowed only to harvest in the waters in which they were residents. In 1971, however, in a case brought by a waterman from Smith Island, the State Court ruled in Bruce vs. Director, Department of Chesapeake Bay Affairs, that these statutes violated the Maryland and United States Constitutions and were therefore illegal. This allowed watermen to travel from county to county harvesting oysters and crabs wherever they were most plentiful. While the Bruce Decision tended to equalize the prices paid for the products around the state, it caused a great deal of trouble for the management agency. Watermen would congregate in large numbers wherever catch rates were highest causing rapid depletion of the stocks and a great deal of animosity between residents and non-residents of the counties which had the greatest resources. Coupled with this new found mobility was a tendency for watermen to invest in larger vessels since they were in many cases traveling long distances by water and needed vessels which were capable of withstanding heavier weather and which had larger cabins for extended accomodations. Boats which would enter the patent tong fishery for oysters were frequently more than 40 feet in length.

Of recent development have been court decisions which have declared invalid residency requirements between states. At the present time there are some problems occurring with large numbers of Maryland watermen working in the Virginia portion of the Bay harvesting crabs as a direct result of a case brought by the Smith Island watermen which resulted in a decision last year.

Economic Factors

The energy crises of the 1970's changed the way many watermen thought about vessels from the standpoint of motive power. The search for speed under sail had adapted itself very nicely to the use of engines during the 20th Century. This was apparent by the engines which watermen were using in their vessels. Most of the boats were powered with large gasoline engines converted from automobile use to marine by means of adapter kits. These large displacement engines used great amounts of cheap fuel. As long as the fuel was cheap it was economical to use these conversions. The first "fuel panic" which occurred in 1973 caused prices to rise from around 35 to about 70 cents a gallon. There was a slightly noticeable shift in engines. Some watermen, especially those with larger boats or who traveled a great deal, began to invest in diesel engines and to adequately size them to their boats. Previously, many of the large gasoline engines supplied more power to the shaft than the boat was capable of using. There seemed to be as much a status to having a large engine in one's boat as in one's automobile.

The second fuel crisis in 1979 boosted the price once again and brought it in line with the real cost of energy in the world. Prices about doubled again from the 65 cent range to around $1.25 a gallon. The shift in engines at that point was fairly dramatic. Many of those who were still using gasoline conversions changed from large displacement eight cylinder engines to smaller six cylinder models. Larger boats were changing much more readily to diesel and there began a trend to smaller fiberglass boats which were trailerable.

This shift to small trailerable boats marked an interesting change in the Chesapeake Bay away from locally designed and built craft to those from other areas.

In the 1970's a few companies, located primarily in the areas of North Carolina and New England, began to produce small fiberglass boats designed specifically for the commercial fisherman. This was a distinct departure from earlier companies which had concentrated primarily on the recreational boating market. These boats were very attractive to the Bay watermen because they offered low maintenance with a well designed fiberglass hull with no extraneous frills and the added advantage of being able to trailer the vessel to the place nearest his point of work. With an excellent road network built up over the years and many public and private launching facilities around the Bay area, he could move easily from place to place in search of the oysters, crabs, and fish which had formed the livelihood of him and so many before him. These small commercial workboats have become very popular in the Bay area and many watermen have either totally converted to these small boats or have one in addition to their large vessels.

The Maryland waterman has come quite a distance from the use of the log canoe with which to get around in a place which had no interconnection by roads to the use of small fiberglass vessels which make good use of the current excellent road network in order to carry on his work.

DEVELOPMENT OF A SAIL-ASSISTED FRP FISHING CANOE FOR THE COMOROS

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ABSTRACT

The development of a FRP fishing canoe with sail and outrigger is described. The light weight, beachable hull with a small marine diesel has gained more energy-saving performance by fitting with a simple alminum mast - Genoa sail combination. Mental acceptability to the local fishermen was also considered to facilitate the smooth introduction of totally new concept to the artisanal fishery.

BACKGROUND AND HISTORY

Yamaha Motor Co.,Ltd. - well known as one of the leading motorcycle manufacturers in the world - has been also contributing in the world fisheries scene by supplying its FRP fishing boats and outboard motors to the fishermen for more than two decades.

Particularly for these years the growing demand from various countries has led the company up to design and manufacture FRP fishing boats which are really suitable for their specific uses, and the company has answered one by one to those demands. Sri Lanka, Senegal, Nigeria and Mauritania are some of the countries in which its design projects have been conducted.

Incidentally and opportunely, one of the Japanese Government Overseas Aid Schemes had materialized in 1981 as the development of local fisheries in the Federal and Islamic Republic of the Comoros. Yamaha was awarded a tender to supply fifty (50) FRP fishing boats as a part of the aid and thence started a design of the new type of fishing craft for the country.

THE COMORO ISLANDS

The Federal and Islamic Republic of the Comoros chiefly consists of three islands which lie in the north of the Mozambique Channel off the east African coast. Its land area totals about 1,600 KM^2 and on the coastlines of those three islands, most of its 400,000 population settle. Among them some 5,000 are fishermen using very small dugout canoes called "Pirogues" in French - their former ruler's language. The craft falles into a category of so-called "Double Outrigger Canoe" which has a float on each side of the hull. This type of canoe is widely distributed from the Pacific Islands to Madagascar, and is reported to have certain correlation with each other. The Comoro canoe is said to be a good example which indicates the westernmost trace of the outrigger canoe culture.

DESIGN CONCEPT

In General: The objective of the development was to introduce a mechanized FRP replacement of the traditional wooden dugouts and to improve the local fishermen's lives through this process. The word "replacement" did not necessarily mean the same shape and size of the boat. It should supersede the old one. A drastic change should, however, be avoided because fishermen have a common tendency being reluctant to the new ideas - particularly when they are brought from outside. Therefore, the appearance and general arrangement were designed as similar as possible to the existing canoes, yet all basic design features be observed. (Fig.1)

Hull: The size of existing canoes (mostly 3 - 5M) must have long been decided by the fishermen who took various factors into account, such as local availability of wood, building cost, weight restriction in case of beaching (they carry the canoe by hands - not pull on the ground.) etc.. From the safety point of view as well as economical standpoint they are too small. To be able to yield the sufficient amount of catch to step up from the subsistance level, the size should be at least doubled.

On the other hand, the lack of harbor facilities and inevitable beach landing operation requires as small and lightweight as possible craft even though it is a modern mechanized type.

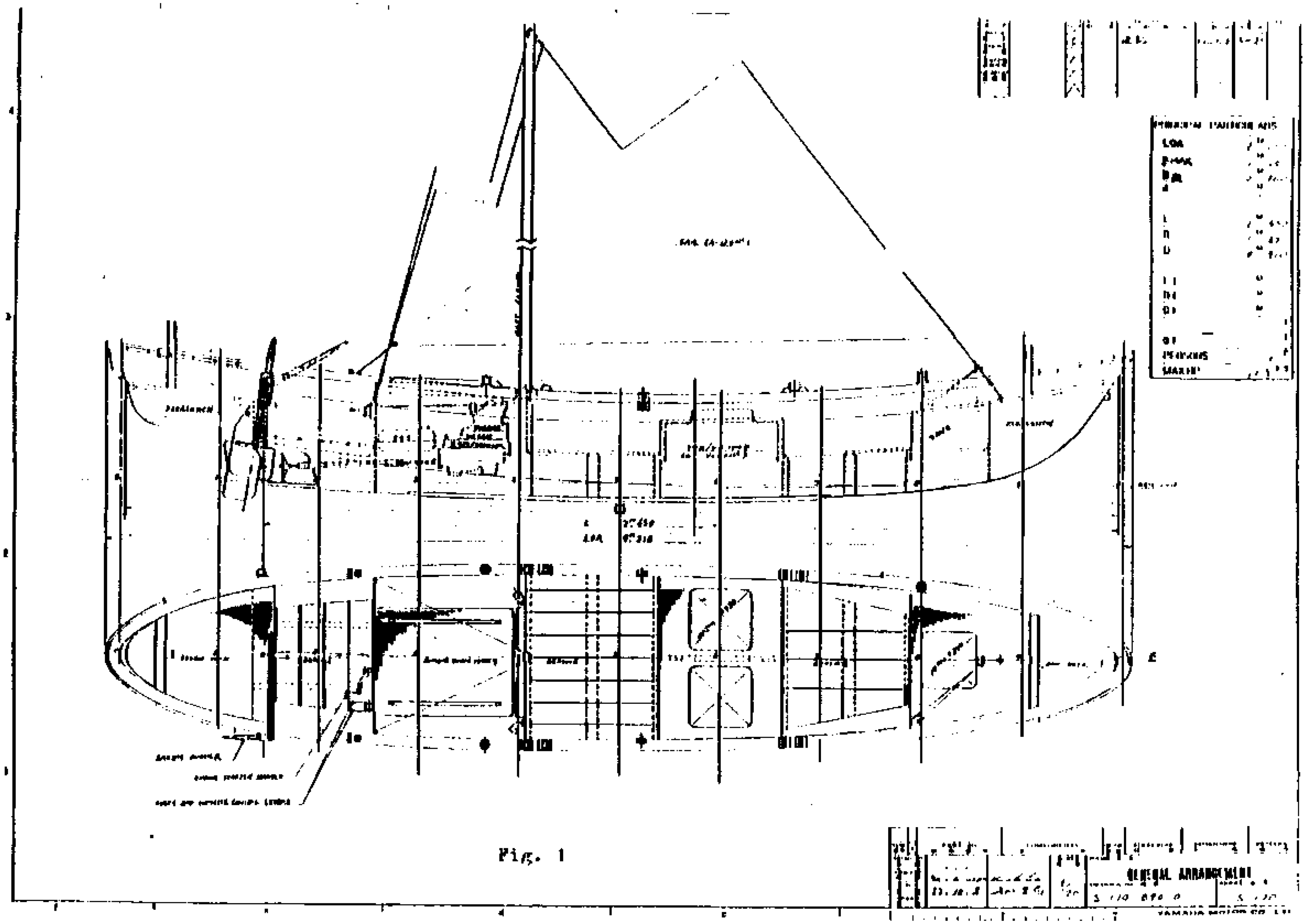

Fig. 1

The company's previous experiences showed that the launching weight of one tonne (1,000 KG) would be maximal for the hand-hauled beach landing craft. Taking this figure into consideration, the principal dimensions were decided as shown in the SPECIFICATIONS. This 9.21M canoe may be depicted as one of the minimum offshore fishing craft with marine diesel.

The hull shape is rather classic; long slender double-ender with moderate chine. Wide shallow keel runs through centerline like a spine, and on its aft end allocated the propeller.

Engine: The mechanization of existing canoes had been partly done by fitting with small (5 - 8PS) outboard motors and has so far proven successful. It was desirable, however, to install a small marine diesel in the mentioned boat. There have been many controversies between diesel inboards and gasoline outboards, and it is not practical to lean always to one side. However, as in this case, once its higher initial cost - the most essential disadvantage of diesel - is met by government grant, there is no hesitation to dieselize. The lower running cost and simpler maintenance would certainly profit the local fishermen.

Sail Rig: The sea around the Comoros is predominated by the tropical monsoon almost all-year round. The local fishermen have been making full use of the wind for their trips to and from the fishing grounds. Mechanization is indeed a very important step towards the modernization of artisanal fisheries, but there has also arisen a new problem inherent in this step; ever-increasing fuel cost makes big holes in the fishermen's pockets and forces their fishing efforts almost profitless.

Reconsideration of Sail-Assistance to the mechanized craft is therefore of prime importance, particularly for such a small scale fishery as of the Comoros. The first option of the sailing rig was the conventional lateen rig which is still used on the existing canoes. That idea was eliminated however, on the way of design consideration because of its long heavy yard which would without doubt spoil the workability onboard. Secondly, the modern and efficient Bermuda sloop rig was studied. But it was also unadapted to the purpose for the reasons of complex sheeting arrangement and existance of the cumbersome boom.

Finally adopted was a low aspect ratio Genoa sail as shown in Fig.2. Its advantages are:

1. No boom, no yard, then no hindrance to the deckwork
2. Easy furling by a roller furling device
3. Easy single-sheet handling of the sail
4. Higher sailing efficiency owing to the sharp leading edge of the sail not spoiled by mast or yard

While there are some disadvantages such as the deeper sail draft in the free-running leg due to its long unsupported foot of the sail. For all those points, it was obviously better than other sail arrangements for the FRP canoe. The sail area was decided according to the maximun allowable heeling angle under certain wind force assumed as most predominant in the area. Hem of the sail was patched with blue-colored strips to protect from the strong ultra-violet rays while furled around the forestay.

Outrigger: Although the hull was designed to have sufficient stability under sail, double outrigger was added as an optional equipment mainly because of the local fishermen's preferences; so-called "Security Blanket" effect was expected as to the stability of the hull. Broader working space between outrigger booms was also considered useful. The outrigger assembly itself would be utilized, anyway the practical usage was left to the fishermen.

The floats were made of FRP, and outrigger beams of anodized alminum pipes Both were jointed with stainless steel bands. Conventional rope-tying method was employed to secure the outrigger to the hull so as that the fishermen could easily deal with it. All jointed and rope-tied portions of the pipes were sheathed with rubber collars to prevent electrolysis and abrasion. To avoid unnecessary frictional resistance, the floats were set at the height 10cm above the designed water line. The float generats its maximum buoyancy of about 140 KG at the heeling angle of 15 degrees to either side.

SPECIFICATIONS

A) Principal Particulars

L.O.A.	9.21 M
Beam	1.76 M
Depth	0.90 M
Gross Tonnage	2.1 Tons
Displacement	950 KG
Crew	6 Psns.
Construction	FRP Single Skin Construction

B) Propulsion

Main Engine	Yamaha ME60H 4 cycle diesel; sea water cooled
Output	11.5 PS/ 2,600 rpm
Reduction Ratio	1.03
Propeller	Alminum Die-cast D 9¼" x P 6½" 3 blades
F.O.T.	48 Liters (diesel)
Autonomy	100 Nautical Miles
Sail	Duclon 4.5 oz 12.0 M^2
Mast	A6063 Mast Section L. 5.4 M
Oars	Wood (one pair)

C) Fishing Gear

Multi-purpose Winches	3 sets
Trolling Rod Holders	2
Insulated Fish Hold	0.6 M^3

D) Other Features

Outrigger Float	FRP 31 KG ea.
Outrigger Beam	A6063 O.D. 100 mm t 2.0 mm

SEA TRIALS

After completion of the first production canoe, the sea trials were conducted under the supervision of the Japanese Government Authority.

As for the speed of the boat, Maximum of 7.95 kt was obtained under light condition without outrigger. It was quite acceptable for the size of the boat and engine. After necessary engine break-in, it is expected to well exceed the design speed of 8 kt.

The sailing speed was also measured on several different legs, and the maximum speed of 4.23 kt was observed in abeam run at the wind speed of 6 M/sec. approx. The heading angle was nearly 50 degrees to the wind direction and tacking was possible when sailed without outrigger. It was observed that considerable amount of wind power was exhausted by turning the propeller underway. It would be desirable to lock the propeller shaft while under sail.

Beach landing test was conducted on the sand beach of a small island near the boatyard. Necessary towing force was measured for the future data accumulation. Using plastic sleepers it was easily pulled up by hands of ten (10) men. It is recommended to use the launching trolleys which were also included in the aid materials wherever their use is possible.

CONCLUSION

Fifty (50) units of the FRP canoes were shipped to the Comoros in February 1983, and to date all major design objectives have been met with regard to the local fishing conditions. The fishermen who used the canoes are reported to be satisfied with the greater loading capacity and better stability as well as the quite similar handling response to the existing pirogues.

Although the practical data proving the improved profit by the introduction of FRP canoes are not yet available, from the sea trials result it is evident that their superb speed performance and fuel economy can result in the significant saving of both cruising time and running cost for the Comoro fishermen.

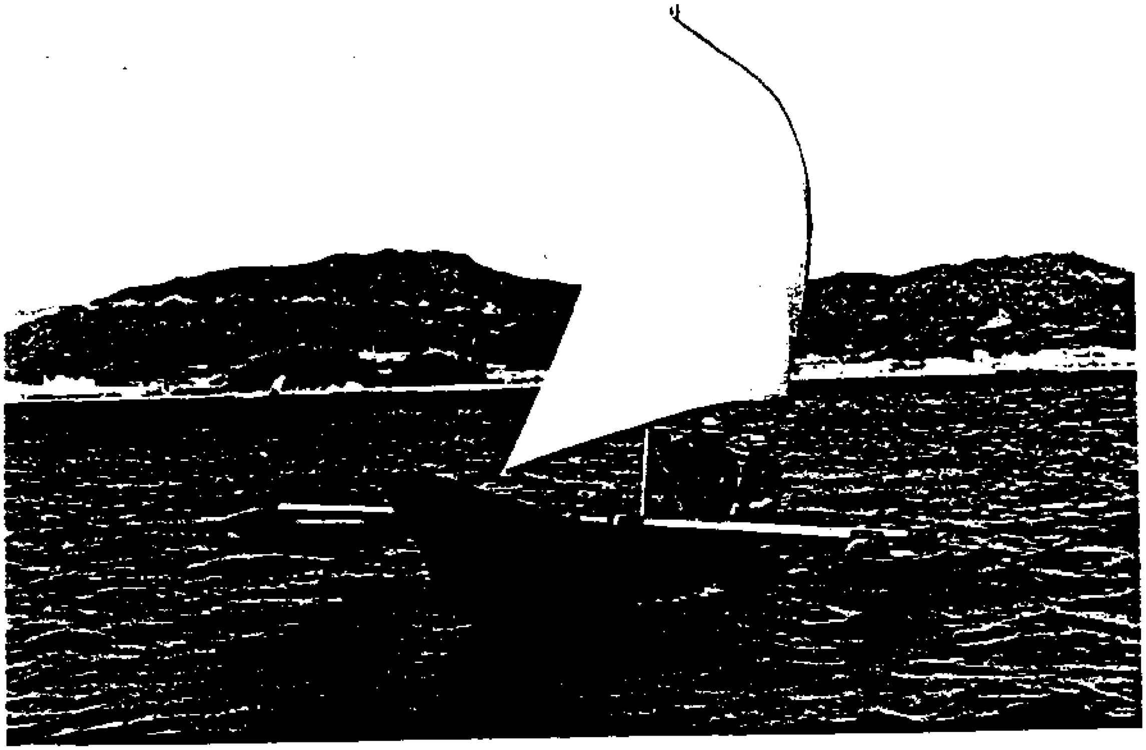

Fig.2

APPENDIX

CONFERENCE PARTICIPANTS

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