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Structural Considerations in the Design of the POLAR Class of Coast Guard Icebreakers

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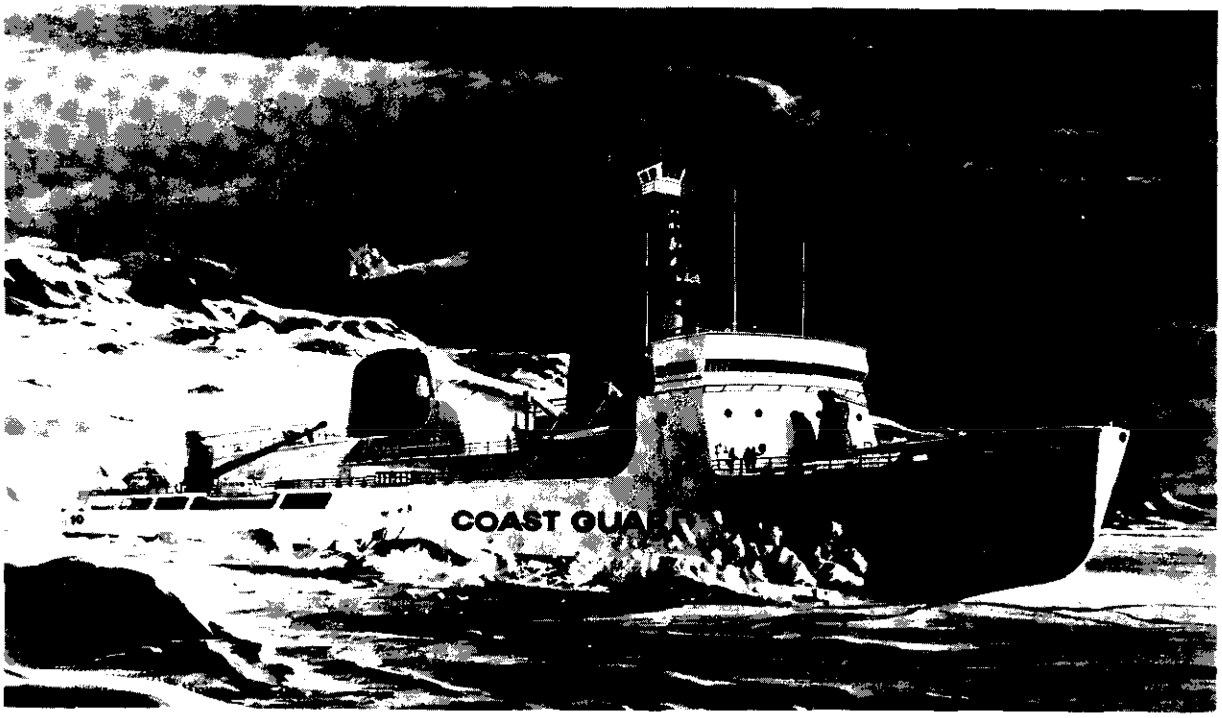


Artist's conception of United States Coast Guard icebreaker POLAR STAR (WAGB-10)

ABSTRACT

The design of the Coast Guard's POLAR Class icebreakers incorporated the results of four years of research, analysis and testing aimed at optimizing the structural design. Operation and load data were gathered by instrumenting existing icebreakers. Extensive tests and studies of available steels as well as a methodical analysis of reported structural failures were undertaken to assist in the selection of hull materials. Various finite element analyses were conducted for portions of existing and proposed structural arrangements with special emphasis on the forward structure. With the cooperation of other agencies, studies of the strength of sea ice led to updated concepts of loading that resulted in significant changes in design philosophy and refinements over previous icebreaker designs.

INTRODUCTION

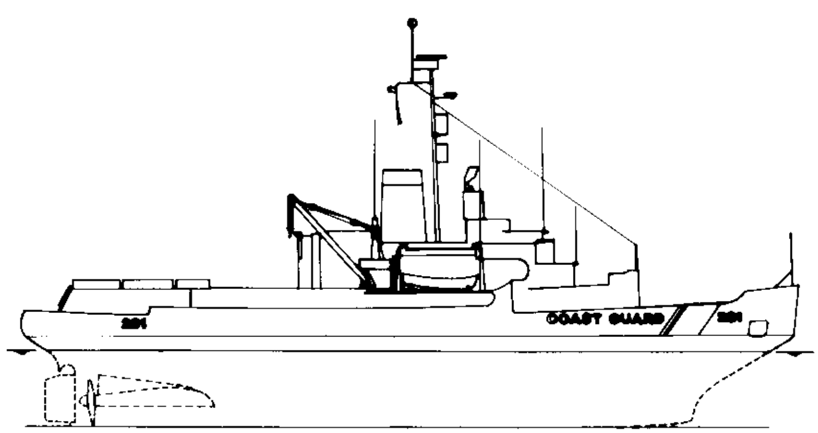
Polar icebreakers operate under the most inhospitable ocean conditions the world has to offer. Designed for optimum performance at low temperatures in ice fields varying from uniform plate ice to deep windrowed ridges, they also transit areas of intense heat, high winds, and extreme sea conditions. Their hull structure sees a variety of loads ranging from those thermally induced to concentrated ice impact loads to those impossible to design for, such as grounding on uncharted rock pinnacles. The icebreaker designer must allow for extreme loads on the one hand, but must pay attention, like all naval architects, to detail design, with a view toward eliminating structural failures from more subtle causes such as elastic instability, brittle fracture, and fatigue.

The United States Coast Guard affords icebreaker support, both domestic and polar, to a variety of ship operations conducted in areas made hazardous or impassable by ice. Typical polar icebreaker missions include escort of vessels supplying outlying military or scientific stations, independent logistics support to similar outposts, and ice and ocean survey operations and support in both polar regions.

Coast Guard icebreakers engaged in these operations have recently consisted of six WIND Class vessels (Figure 1) and the GLACIER (Figure 2). Budget reductions and old age will have soon forced the decommissioning of all WIND vessels except for WESTWIND and NORTHWIND, each of which has undergone extensive structural and machinery renovation to prolong its service life.

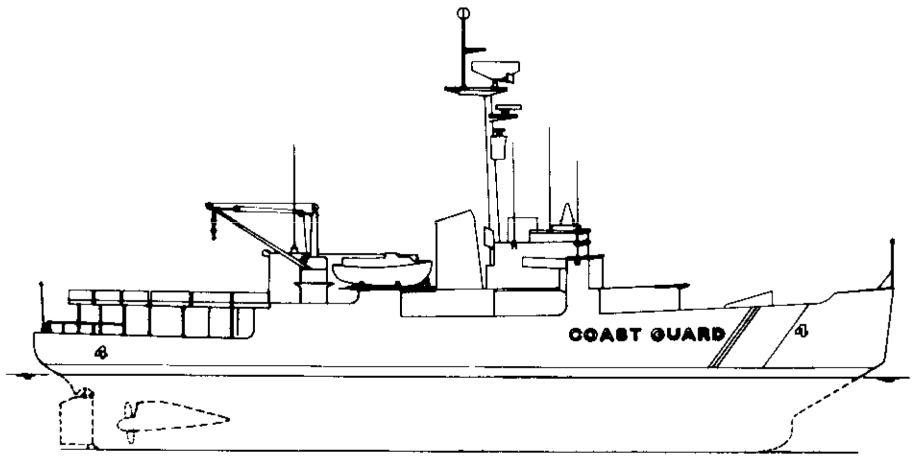
The need has been apparent since the early 1960's for a new class of icebreaking ships to replace aging members of the fleet and to undertake more extensive duties as Coast Guard responsibilities change and expand. The recent extension of the search for oil into the Arctic region, for instance, appears likely to require a strong Coast Guard response capability in this area. In 1966 the Icebreaker Design Project was established in the Naval Engineering Division at Coast Guard Headquarters to initiate preliminary design work on a new polar icebreaker. The initial thrust of the effort was toward a nuclear-powered cutter, but this was later modified to conventional diesel-electric and finally revised to include a gas-turbine mode of operation. During the existence of the Icebreaker Design Project, its personnel delved into the significant aspects of ship design as it applied to icebreakers. Its most significant contributions to icebreaking technology were probably in the areas of hull form, powering predictions, and hull structure, the latter including material selection. In excess of 100 discrete projects were completed, many dependent on Coast Guard-initiated research. Literature searches were conducted to insure that existing technology was not neglected. Project members received significant support from contractors, other government agencies, and a large number of Coast Guard personnel.

By the time of the dissolution of the Icebreaker Design Project in 1969, a basic preliminary design had been developed. Hull form and principal dimensions had been defined, rough arrangements completed, a machinery plant size and type selected, a basic structural



Length overall------269'-0"	Displacement to DWL-----5,300 long tons
Length, DWL---------250'-0"	Displacement, maximum---6,515 long tons
Beam, maximum-------63'-6"	Complement--------------174
Beam, DWL-----------62'-0"	Speed, knots, cruising--16
Depth to main deck--37'-9-1/2"	Propulsion--------------Diesel-electric
Draft to DWL--------25'-9"	SHP---------------------10,000
Draft, maximum------29'-1"	Number of screws--------2

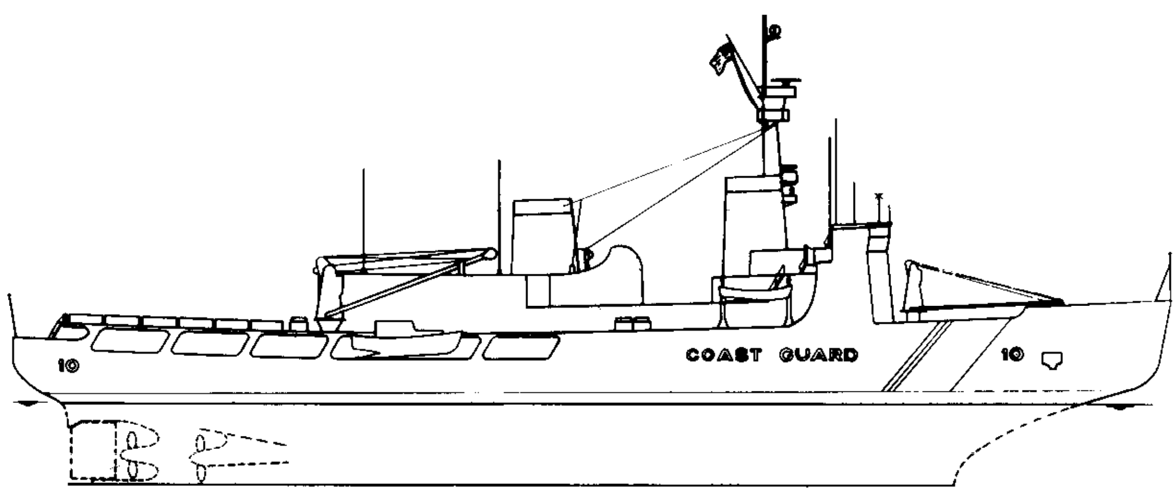
Figure 1. WIND Class Profile and Characteristics



Length overall------310'-7"
Length, DWL---------290'-0"
Beam, maximum-------74'-3-1/2"
Beam, DWL-----------72'-6"
Depth to main deck--38'-4"
Draft to DWL--------25'-9"
Draft, maximum------28'-6"

Displacement to DWL-----7,600 long tons
Displacement, maximum---8,449 long tons
Complement--------------197
Speed, knots, cruising--17.6
Propulsion--------------Diesel-electric
SHP---------------------16,000
Number of screws--------2

Figure 2. GLACIER Profile and Characteristics



Length overall------399'-0"
Length, DWL---------352'-0"
Beam, maximum-------83'-7"
Beam, DWL-----------78'-0"
Depth to main deck--49'-3-1/2"
Draft to DWL--------28'-0"
Draft, maximum------31'-9"
Displacement to DWL-11,000 long tons

Displacement, maximum---13,179 long tons
Complement--------------138
Speed, knots, cruising--17
Propulsion--------------Diesel-electric or Gas turbine
SHP---------------------18,000 D-E
60,000 G-T
Number of screws--------3

Figure 3. POLAR Class Profile and Characteristics

arrangement chosen, and tentative scantlings determined. In March of 1970, these early beginnings were transferred to the existing Design Branch of the Naval Engineering Division for contract design development. Completion of the contract design in 1971 was followed by bid solicitation and award of a contract to Lockheed Shipbuilding and Construction Company, Seattle, to build one icebreaker, the POLAR STAR (Figure 3). Delivery of POLAR STAR took place during 1975. A second ship, POLAR SEA, will be completed in 1976.

STRUCTURAL DESIGN

Structural studies within the Icebreaker Design Project began with an assessment of existing ships. GLACIER and WIND Class experience, totalling some 220 ship-years of operation, was analyzed along with that of other icebreakers, domestic and foreign, to identify structural inadequacies and to organize this information in a format useful to the designer. Accumulated experience "reinvested" in subsequent designs is, of course, the essence of the classic ship design process, and the method appears to retain its usefulness as technology advances.

The areas of structural concern listed below were identified as requiring special investigation:

- Load Definition
- Framing System
- Allowable Stresses
- Analysis Methods
- Hull Material
- Detail Design

Although any ship design requires that these items be addressed, special emphasis was imposed in this case because of (1) the lack of documentation for many decisions reached in designing previous icebreakers, (2) obvious technological advancements in the intervening years, (3) the accumulation of experience with post-war icebreakers and resulting concept realignments, and (4) the realization that this project presented an opportunity to perform benchmark studies that could be referenced with confidence in future design work.

LOAD DEFINITION

Introduction

Determining rational design loads for a ship of this size and type is a considerable undertaking. The need to research twenty-five years of highly theoretical icebreaker and ice technology literature was obvious. Further, if these studies were to be of any value they had not only to be compared to the original design philosophy of the WINDS and GLACIER but also correlated to our operational experience with these ships. It was also necessary that the investigations put in proper perspective the state of the art in countries such as Canada and Finland, which had been busily building icebreakers since the early 1950's.

The design philosophy of the WIND Class took into account only a few major ship parameters such as displacement, shell slope at the ice belt, ship length, and beam, by use of the formula reported in Volume 54, 1946 of the SNAME Transactions [1]

$$P = \frac{D \sqrt{1 + m^2}}{L}$$

where P is the load per foot of waterline perimeter with the ship supported solely by ice at the waterline, D is the displacement, m is the average shell slope, and L is the waterline perimeter. The unit load P readily yields the load per frame which was in turn used to design the ice belt plating and framing. The load per frame was envisioned as quasi-concentrated at a pressure of 3,000 psi and applied halfway between transverse frames in the design of the plating. The framing was then designed to withstand the same load at midspan. The above design philosophy yields a structure in which the framing is incapable of withstanding a uniformly distributed pressure of the order of the crushing pressure of ice and considerably less than the distributed pressure that the plating can support. This "framing pressure capacity" is less than 150 psi for the WIND Class and about 200 psi for the GLACIER, both of which have experienced damage in service.

Ice Strength

There was therefore no question of the need to establish valid design ice pressures based on realistic operating conditions. A thorough ice technology literature search indicated a wide disparity in the measured physical properties of ice and, more significantly, showed that most reports did not indicate important factors such as sample size and orientation, temperature, and salinity (Table I).

This resulted in a decision to undertake an independent ice pressure study. Assistance was provided by the Army's Cold Regions Research and Engi-

DESCRIPTION	THICKNESS	TEMP DEG F	STRENGTH COMPRESSION PSI	STRENGTH TENSION PSI	SOURCE
Clear, cold, solid ice	Not stated	Not stated	3,000	250	H. F. Johnson SNAME-54 (1946)
Not stated	Not stated	Not stated -1°C (Markessa)	213 to 388	190	Fruhling of Koenigsberg (prior to 1900)
Clear fresh water ice pan	Not stated	Not stated	327 to 1,000	Not stated	W. Ludlow (prior to 1900)
River, Kennebec	Not stated	-4°F	399 to 970	Not stated	Prof. Kolster, Helsinki (prior to 1900)
Not stated	Not stated	32°F	363	Not stated	Prof. H. M. Mackay Prof. H. T. Barnes (1914)
St. Lawrence river ice	Not stated	28°F	300	Not stated	
Ice of excellent quality	5-in. X 5-in.	14°F	693	Not stated	Prof. E. Brown
Blocks loaded to 1,000 pounds in 2 secs	test pieces	2°F	811	Not stated	(McGill University)
Hard old arctic ice	Not stated	Not stated	1,000	250	A. Watson, IME (1958)
Coast Guard criteria based on temperature, salinity, & strength profiles	10-25 feet	0°F surface 30°F bottom	600 psi	---	Dayton [2]

Table I. Ice Characteristics as reported in Volume 67, 1957 of the SNAME *Transactions* [3] with Coast Guard data added neering Laboratory (CRREL) and the studies were documented by Dayton [2].

The following parameters were considered:

- Sample Size - Due to the greater incidence of faults and impurities, the average compressive strength decreases as the sample size increases.

- Salinity - Varies through the thickness of sea ice; strength is in turn a function of salinity.

- Temperature - Varies through the thickness. Strength decreases with increase in temperature.

From the above, strength profiles and average values for representative thicknesses of ice were calculated (Figure 4). The following average design pressure values were then arrived at, considering data reliability and contact area factors for each condition and in both cases departing from a maximum ice sample strength of 1,000 psi:

- Uniform static load for beset condition:

  Factors for

Sample size	.4
Strength profile	.75
Contact area	.5
Data reliability	2

  Design pressure =
  1,000 psi X .4 X .75 X .5 X 2
  = 300 psi

- Uniform impact design pressure for bow and stern areas:

  Factors for

Sample size	.4
Strength profile	1
Contact area	1
Data reliability	1.5

  Design pressure =
  1,000 psi X .4 X 1 X 1 X 1.5
  = 600 psi

<u>Load Distribution</u>

It was next necessary to determine the manner in which these loads were to be applied to the hull. Since impacts establish the maximum load to be absorbed by the shell structure, studies were carried out to determine the distribution of impact loads on the hull under varying conditions of speed, floe size, ice thickness, and impact location.

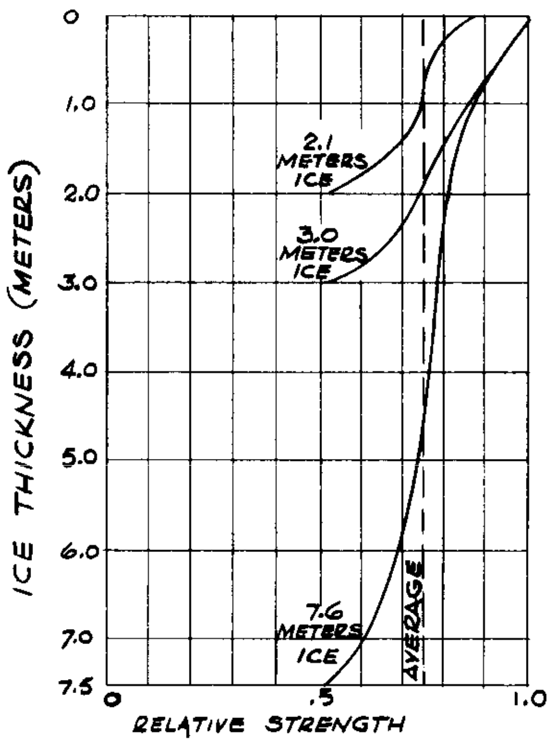
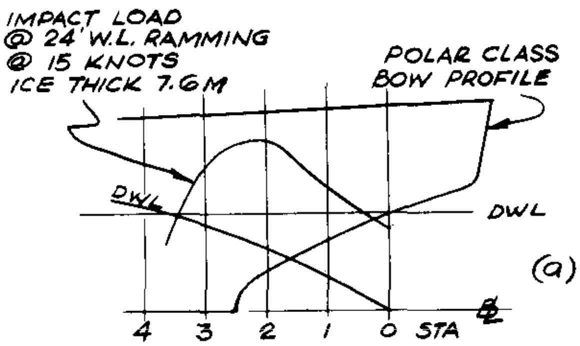


Figure 4. Ice strength profiles, from [2]

Work of this nature included that by Nogid [4], White [5], Kheisin [6], and Tarshis [7]. The Icebreaker Design Project initiated additional studies, resulting in contributions by Estes [8] and Dayton [2].

Dayton applied computer techniques to Tarshis' procedures to determine impact load and area as a function of location. The analysis was dependent on ship speed, hull shape, mass and rigidity of the ship structure relative to the ice, and entrained mass of water. A significant finding of this work was that impact loads could realistically be considered as uniformly distributed over a large area, typically 480 square feet for an 18,400-ton impact load.

Figure 5(a) illustrates the effect of location on total impact load as determined by Dayton for one of several hull designs developed during the preliminary design phase. It is apparent from the superimposed waterline that the greatest loads are predicted for the shoulder area at about station 2. Figure 5(b) provides a correlation with WIND Class experience. The two-peaked curve indicates the relative number of reported damages versus longitudinal location, as compiled by D. E. Woodling and R. A. Yuhas of the Coast Guard. The higher peak occurs at the bow screw bossing (the bow propeller was never installed on most ships of the class and has been removed from the others), which



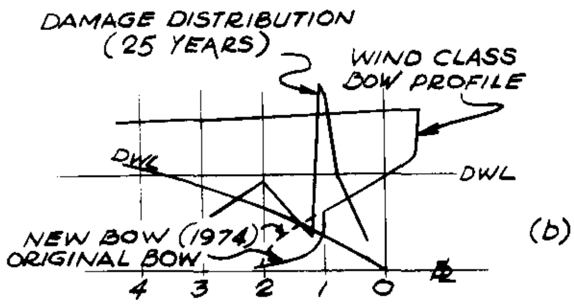


Figure 5. Bow impact locations, POLAR Class (predicted) and WIND Class (actual)

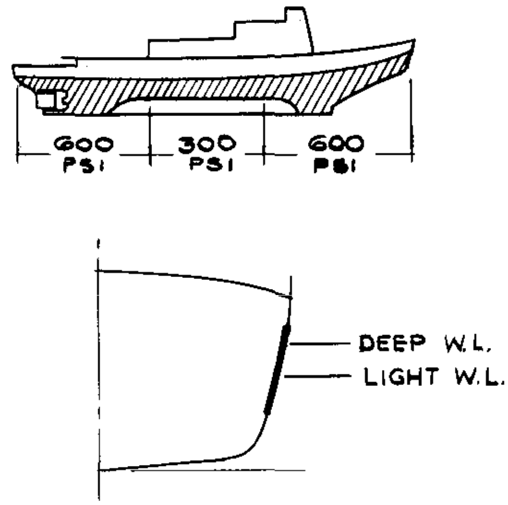


Figure 6. POLAR Class icebelt configuration showing design pressures, from [9]

provides a "stop" in heavy ramming conditions and has precipitated numerous structural failures at station 1. (The faired stem of the POLAR Class and renovated WESTWIND and NORTHWIND is expected to alleviate this problem by reducing shock loads resulting from sudden energy dissipation during compressive loading of the ice by the step.) Dayton's analysis did not include the bossing area and thus related better to the new hull form. As the figure indicates, WIND Class shell failures have also peaked in the shoulder area where the analysis predicts peak loads.

Icebelt Definition

The icebelt is that portion of the shell plating that is strengthened to accept the design ice loads. Figure 6 indicates the extent of the icebelt on POLAR Class ships.

Longitudinal Bending

Studies were also made to confirm the magnitude of wave-induced bending stresses as well as those experienced when the bow is "beached" on an ice shelf. These studies confirmed previous investigations conducted with the MACKINAW [10] in which it was concluded that the stresses induced by these loading conditions are of secondary importance in the determination of the shell and deck scantlings but nevertheless influence their design. Still-water bending stresses were included along with ice loading in the design of the ice belt. The decks and bottom structure were sized to avoid plate buckling failure under wave-induced bending and hydrostatic loads. The bottom plating thickness was further increased to account for corrosion and to improve the resistance to grounding damage.

FRAMING SYSTEM

Various schemes of longitudinal framing systems were considered and discarded because of weight considerations. Included among these was a system that made use of extrusions incorporating up to two frames and their corresponding areas of shell plating. The study proved this to be heavier as well as costlier because of the unique procurement and fabrication problems involved.

The transverse framing system historically used for icebreaking ships was therefore adopted. Supports for transverse frames were readily available in the form of decks; the considerable side tankage permitted the use of decks or partial decks throughout the length and depth of the hull. The framing was canted at the bow and stern so as to place the framing members more perpendicular to the shell for more effective load-bearing ability. The 16-inch frame spacing again proved to be the least-cost alternative when compared to smaller grillages requiring more frames and lighter plating.

Both WIND Class and GLACIER have, within approximately the midships six-tenths length, transverse frames supported by truss structure, Figure 7,

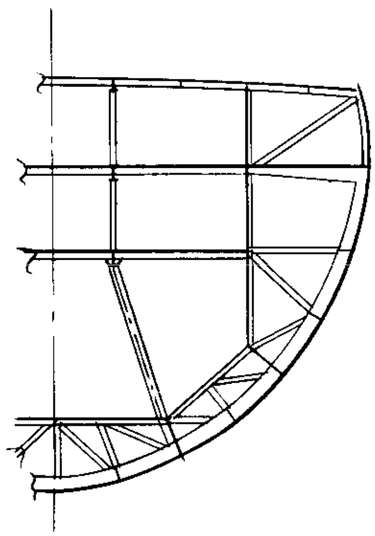


Figure 7. Truss framing system of WIND Class and GLACIER

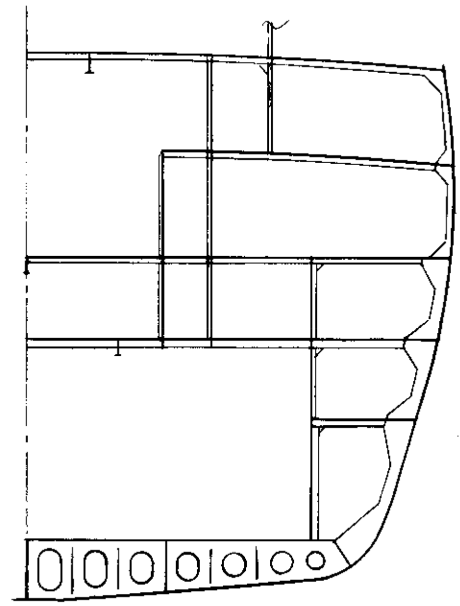


Figure 8. Grillage framing system of POLAR Class

whereas other U. S. Coast Guard icebreakers (CACTUS, STORIS, etc.) and most European icebreakers employ deep webs and girders or grillage structure for shell support. In all cases, however, bow and stern framing consists of grillage structure. Detailed studies of each system, performed by the Coast Guard and by Lloyd's Register of Shipping [11], compared the strength and weight of optimized truss and grillage structures. Particular attention was paid to the behaviour of the structure under overload conditions. As expected, the truss was shown to be the lesser-weight system when compared on an equal design-strength basis. It was the response of each to overloads that led to the selection of the grillage, Figure 8. The truss system, in response to an overload condition, develops significant bending moments at the connections which, combined with the axial loads on the compression members, could lead to a progressive failure of adjacent frames through elastic instability, and subsequent collapse of the shell framing. This mode of failure has been observed on present Coast Guard icebreakers. A stable grillage structure, on the other hand, may experience local plastic deformation under overloads but will not necessarily lose its ability to sustain additional loads at the design load level.

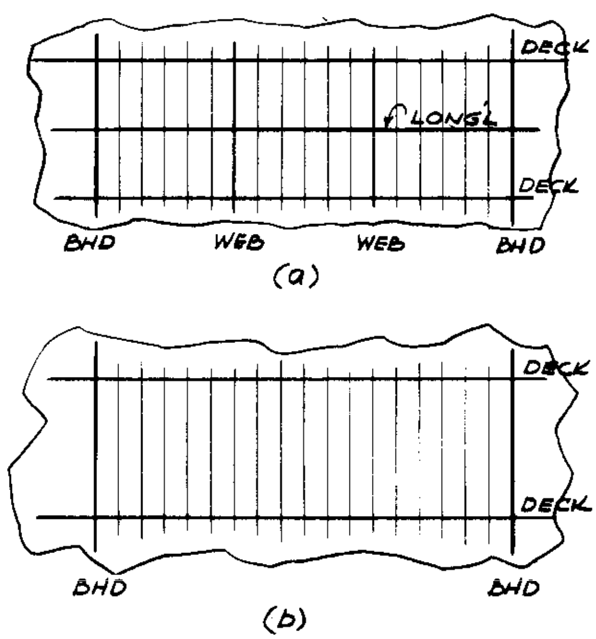


Figure 9. Alternate POLAR Class shell framing details, (a) proposed and (b) final

Figure 9(a) illustrates a framing system that was under consideration into the preliminary design phase of the POLAR Class. This system was consistent with an early assumption concerning impact load distribution, namely that the load would be applied to one transverse frame over its length between two decks. The purpose of the web frame-supported longitudinal member was to distribute the load onto adjacent frames. As the load criteria were refined, however, it became apparent that this was not a realistic load assumption and that the

600 psi impact load could be considered distributed uniformly over a larger area supported by many transverse frames. This reduced the basis for selection to a simple one of minimum weight and cost, thus the system of Figure 9(b) was chosen.

ALLOWABLE STRESSES

Since the design pressure values discussed earlier contained reliability factors, it was decided to design the shell structure to the yield strength when loaded amidships to 300 psi, the uniform beset load, or at the ends to 600 psi, the uniform impact load.

Plastic deformation will be expected when imposed loads exceed the design loads. Dayton's analysis predicts the formation of three plastic hinges at a 1,200 psi distributed load. Greenspon's methods [12] indicate a permanent shell plate set of 0.09 inches at a uniform pressure of 1,150 psi.

ANALYSIS METHODS

The finite-element method of analysis proved to be an effective tool for evaluating icebreaker hull structure. Genalis [13] investigated the applicability of the program STRESS[1] to two- and three-dimensional frame analysis and compared WIND Class transverse frames by the two methods. He concluded that the two-dimensional computer analysis was effective but that the three-dimensional approach, which included the effect of longitudinal members, was preferable, despite its greater cost, because of the improved accuracy. He noted that the two methods predicted frame bending moments that were different by as much as 33 percent. Lloyd's Register of Shipping also used STRESS for the GLACIER analysis in their GLACIER-MOSCOW comparison in 1967 [11].

The program FRAN[2] was used by Lloyd's for the MOSCOW analysis and by Kiesling [16], who studied the application of available computer methods to the structural analysis of an icebreaker. He compared STRESS and FRAN and concluded that the latter was more suitable since it could (1) handle more complex problems, (2) more easily account for temperature effects, and (3) was more efficient in the use of machine time. Kiesling cautioned that further development was necessary in structural modeling techniques and in determining the loads acting on an icebreaker, and noted that reliance on these programs, with their two-dimensional representation of plates and shells, is unrealistic if quantitative data for such members is required.

Fey et al [8] studied the bow structure of an icebreaker using SAMIS[3], a matrix interpretive code which modeled the structure as a combination of beams, bars, and shell elements. In order to save time and decrease the possibility of error, an input data generator was developed to aid in providing descriptive data for nodal coordinates and boundary conditions. The WIND Class bow was modeled and it was determined that relatively low pressures were sufficient to produce yielding; this was in agreement with previous observations. This study also reviewed the various impact loading criteria and verified that the rise times of these loads were sufficiently great that dynamic effects could be disregarded.

The detailed structural design of the POLAR Class utilized both manual and computer calculations. The program STRESS was used to analyze typical transverse frames and the Neilsen Grillage Computer Code [18] saw particular application in the design of side shell grillages in way of the machinery spaces.

HULL MATERIAL SELECTION

<u>Experience</u>

Shell structural failures experienced by present icebreakers, principally WIND Class ships, have always been cause for concern. Figure 10 is typical of major damages incurred by these ships. On several occasions of significant shell damage, the opportunity was taken to perform metallurgical examination on cropped-out hull plate [19, 20, 21, 22]. These analyses were invariably characterized by comments such as, "extensive brittle fracture", "indicative of brittle cleavage type failures", or "highly notch-sensitive". In at least one instance it was noted that catastrophic failure would have occurred had it not been for the local nature of the load and the lack of a significant tensile field beyond the immediate damaged area.

Table II describes the physical characteristics of samples of plate from the WIND Class icebreaker USS ATKA (now USCGC SOUTHWIND) in 1965 [22]. The material is unnormalized Navy High-Tensile Steel (HTS) conforming to [23], the HTS specification in effect in 1943. Reference [19] reported on an analysis of plate samples removed from USCGC EASTWIND in 1956, and concludes, "The combination of poor notch-toughness, low

1 Structural Engineering Systems Solver [14]

2 Framed Structure Analysis Program [15]

3 Structural Analysis and Matrix Interpretive System [17]

Figure 10. WIND Class shell structure damage

operating temperatures and impact is considered responsible for brittle behaviour of the plating during failure. . . . Normalizing of the plate material [in the laboratory] resulted in a marked improvement in notch-toughness properties. It may be assumed that had the plating been normalized (as now required by specifications), the extent of the casualty would have been of a less serious nature." The HTS specification has included normalized plate since [24] was introduced in 1953.

By far the greater part of the Coast Guard's structural failure data has been generated by WIND Class ships. Although GLACIER has incurred damage, it is but one ship versus the WIND Class' six and it has a somewhat greater design pressure due to its ice frames being of HTS rather than mild steel as on WIND Class hulls. The present authors could locate no reports of analyses on GLACIER hull steel which is, like the WIND's, HTS. Whether this steel was normalized is not mentioned in available documents, although the hull construction plans antedate by about one year the HTS specification that first required normalizing.

It was apparent from WIND Class experience that an improved material should be utilized in any new icebreaker. The environmental conditions of low temperature and high impact loads, combined with a notch-sensitive material, promoted brittle fracture in extreme icebreaking conditions.

Hull Steel Requirements

Studies by Yuhas and Schumacher [25] of the Icebreaker Design Project identified the following factors as significant in the selection of icebreaker hull steel:

- Low-temperature toughness. The air temperatures experienced by polar icebreakers have been measured as low as -50° F. Figure 11 illustrates the zone of transition from -50° F air temperature to +28° F sea water temperature. The chosen steel must be sufficiently tough at these temperatures to preclude brittle fracture. However, as recently noted by Rolfe et al in Ships Structure Committee Report SSC-224 [26], the combination of toughness, stress

	USS ATKA HTS [22] (averages of 4 samples)	PRESENT-DAY HTS [24]
Tensile Strength	81,000 psi	88,000 max.
Yield Point	56,000 psi	47,000 min.
Elongation, %	27	20
Charpy V-notch energy, ft-lb	14 to 26 @ +40° F	100 @ +40° F (typical)

Table II. USS ATKA shell plate versus present-day HTS level, and flaw size together determine the resistance of a structure to brittle fracture.

- Fatigue Strength. As in any structural design, material selection and detail design go hand-in-hand in determining the resistance of the ice-breaker structure to fatigue. Although longitudinal bending cycles are less than for most merchant vessels, Figure 12, and ice impact loads on a given area occur relatively infrequently [2], the stress levels experienced by an ice-breaker are often severe. Resistance to fatigue as a basic material characteristic assumes secondary importance when compared with the low-cycle fatigue strength of structural details.

- Ease of Fabrication and Repair. Icebreaking vessels typically have heavy plate and dense framing systems and therefore require a relatively large number of construction man-hours, many of them spent within areas of difficult access. It is to the owner's advantage to use a material that will not further increase fabrication difficulties due to more stringent forming or welding requirements.

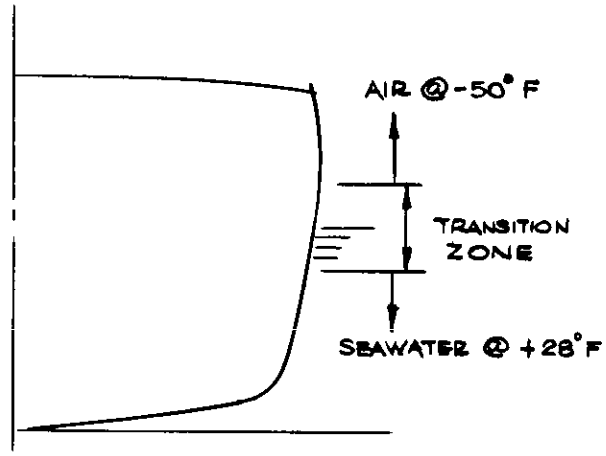


Figure 11. Minimum temperatures acting on polar icebreaker hull

- Yield Point and Ultimate Strength. Adequate tensile properties are necessary to insure that the structure will withstand the basic loads while not imposing high weight penalties and fabrication difficulties.

- Cost. Less is best, but since material characteristics must be optimized, cost may be subordinate to other requirements.

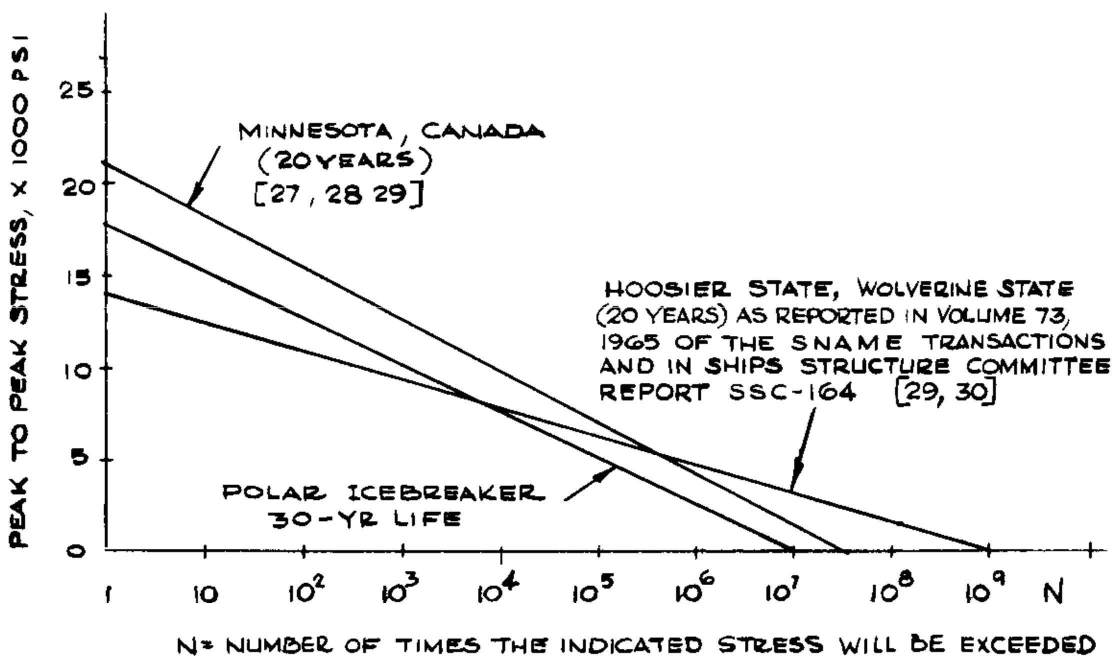


Figure 12. Cumulative frequency distribution of polar icebreaker and merchant vessels, from [2]

- Corrosion Resistance. Since all low-carbon steels tend to exhibit similar corrosion properties in salt water, this parameter will not generally influence the steel selection. Of greater concern are the corrosion properties of the heat-affected zone at the shell seams, where accelerated corrosion has been noted on WIND Class and GLACIER hulls. Insufficient data of this type was available in 1970 to aid in steel/filler metal selection, and the Coast Guard has since initiated a 10-year program to study salt-water corrosion on welded samples of icebreaker plate.

Hull Steel Candidates

The search for the hull steel that best met the above requirements led to two prime candidates, HY-80 and ASTM-A537 [Table III]:

HY-80 (MIL-S-16216)

The advantages of HY-80 for icebreaker hull structure are obvious. Cross-rolling provides it with essentially equal properties in all directions, ideal for impact loading normal to the plate. Its low-temperature toughness is outstanding. Its yield strength of 80,000 psi minimum appears to provide considerable potential for weight reduction. But HY-80 has clear disadvantages also. Cross-rolling, alloying, and extensive testing requirements increase its procurement cost to 3.5 times ABS Class B (1964 data as reported in SNAME T & R Bulletin No. 2-11 [32]). Compared (for now on an equal-thickness basis) with A537, it is less attractive for welding because:

- A greater preheat/interpass temperature is required.
- Lower allowable heat input results in lesser deposition rates.
- A higher-strength, more costly electrode is necessary.
- Joint preparation is more difficult due to its greater hardness.

Not only do these factors tend to drive cost upward, but they also increase the difficulty of repair work and make such work in remote shipyards less desirable. Furthermore, because of fatigue, buckling, and other considerations, full advantage cannot be taken of the higher yield stress to achieve weight reduction. Tensile strength, a measure of overload-absorption capability, is not significantly greater than that of A537. In the final analysis, too little weight reduction and too much increased cost plus the ready availability of ASTM-A537 were the reasons for the elimination of HY-80 as a hull material.

ASTM-A537

The steel described by ASTM Specification A537, adopted in 1965, is a low-carbon pressure vessel steel available in either a normalized (A537A) or a quenched and tempered (A537B) condition. Steel of this chemistry was, in 1970 as now, seeing wide use in the marine field, typical applications being in carriers of liquified petroleum gas (LPG), "all-hatch" ships, and offshore drilling and production platforms in temperate and arctic regions. It had been considered a leading candidate for polar icebreaker hull steel since the inception of the design project due to its excellent low-temperature toughness, good tensile and fatigue properties, and relative ease of welding. Its material cost was about 1.4 times that of ABS Class B as reported in SNAME T & R Bulletin No. 2-11 [32].

Fracture mechanics methods indicated that A537B plate had sufficient Charpy V-notch energy in thicknesses through 1-1/2 inches to insure through-thickness yielding before fracture, in accordance with Rolfe's criteria of [33]. In addition, the method of [34] was used to verify that a built-in surface flaw would not propagate to a dangerous size over a 30-year lifetime.

Steel Selection and Application

The tensile properties of ASTM A-537 are achieved by allowing carbon content to a maximum of 0.24%. The Coast Guard decided, however, that resolution of the toughness-tensile characteristics trade-off should favor the former and investigated a carbon reduction to achieve this. The work of Roper and Stout published in Ships Structure Committee Report SSC-175 [35] and elsewhere [36] indicated that a reduction to 0.17% would provide thick-plate (to 4 inches) A-537 material with toughness far superior to that of ABS Class C while tensile strength was reduced to about 80 KSI, and that underbead cracking was eliminated by using low-hydrogen electrodes.

During the polar icebreaker preliminary and contract design phases, J. W. Kime and R. A. Yuhas of the Coast Guard met with major steel manufacturers to determine the optimum chemistry of an A537-type steel for this application. They then developed a new specification limiting carbon to 0.16% and increasing the Charpy V-notch requirement to 20 ft lb @ -60° F transverse (base metal and as-welded) compared with 15 ft lb @

	HY-80 MIL-S-16216	ASTM A-537A (NORMALIZED)	ASTM A-537B (QUENCHED & TEMPERED)	CG-A537M (QUENCHED & TEMPERED)
TENSILE REQUIREMENTS				
TENSILE STRENGTH	(NO REQMT)	70-90 KSI	80-100 KSI	70-90 KSI
YIELD POINT	80-95 KSI	50 KSI (MIN)	60 KSI (MIN)	50 KSI (MIN)
% ELONGATION IN 2"	20 (MIN)	22 (MIN)	22 (MIN)	22 (MIN)
TOUGHNESS REQUIREMENTS				
UNWELDED CHARPY V-NOTCH FT-LB *	50 @ -120° F	15 @ -75° F + (LONGITUDINAL)	15 @ -75° F + (LONGITUDINAL)	20 @ -60° F (TRANSVERSE)
DROP-WEIGHT				NO-BREAK @ -60° F (-70° F NDT)
WELDED CHARPY V-NOTCH				20 @ -60° F (TRANSVERSE)
DROP-WEIGHT				NO-BREAK @ -60° F
CHEMICAL REQUIREMENTS (LADLE ANALYSIS)				
CARBON, MAX	0.18	0.24		0.16
MANGANESE	0.10 - 0.40	0.70 - 1.35 **		0.90 - 1.50
PHOSPHORUS, MAX	0.025	0.035		0.035
SULFUR, MAX	0.025	0.040		0.040
SILICON	0.15 - 0.35	0.15 - 0.50		0.15 - 0.35
MOLYBDENUM	0.20 - 0.60	0.08 MAX		0.08 MAX
CHROMIUM	1.00 - 1.80	0.25 MAX		0.25 MAX
NICKEL	2.00 - 3.25	0.25 MAX		0.25 MAX
COPPER	0.25 MAX	0.35 MAX		0.35 MAX
TITANIUM	0.02 MAX			
VANADIUM	.03 MAX			

* MINIMUM, AVERAGE OF 3, 10 MM X 10 MM SPECIMEN

** FOR 1-1/2-INCH PLATE

\+ BY ASTM A593; FOR 1-1/4-INCH MAX THICKNESS

NOTE: ALL REQUIREMENTS ARE FROM SPECIFICATIONS CURRENT IN 1970.

Table III. Comparison of hull steels considered for POLAR Class

-75° F longitudinal for the ASTM material. This specification was titled "CG-A537M" to indicate a modified A537 steel for Coast Guard icebreaker application. Steel company pricing methods were such that the identical steel would cost less if purchased under a new specification than as modified A537B.

Steel to specification CG-A537M saw its most essential application in the transition region (Figure 11), where both extreme impact loading and low temperatures will occur. At higher locations, low temperatures will be the dominant environmental hazard, complicated by unavoidable structural discontinuities; on the underbody, impact loading at water temperature will prevail. The use of CG-A537M was extended to include the basic hull envelope and other materials were incorporated as shown by Table IV.

Since icebreakers are historically unable to retain bottom paint, a corrosion allowance was added to the icebelt and all other underbody steel, including the rudder, consisting of a 1/4-inch addition beyond that required for strength. A current Coast Guard R & D study is attempting to determine whether present-day materials have potential as abrasive-resistant underbody coatings for icebreaker application. Such a discovery would have significant impact on the cost of future icebreakers. POLAR Class ships will, when new, carry around some 175 tons of sacrificial steel.

DETAIL DESIGN

Although data of the type presented schematically in Figure 13 [37, 27] would be most useful to the structural engineer involved in the selection of details for any ship design, the method that it implies has not been rigorously developed. The limitations are chiefly in the extrapolation of curve A, the cumulative distribution of frequency, to some curve B, the ship equivalent of the S-N curves C, D, and E. Such a transformation involves altering the fatigue load spectrum of curve A into a representation of number of load applications versus constant mean stress level. Although Nibbering [27] explains one such method, he is careful to emphasize its approximate nature.

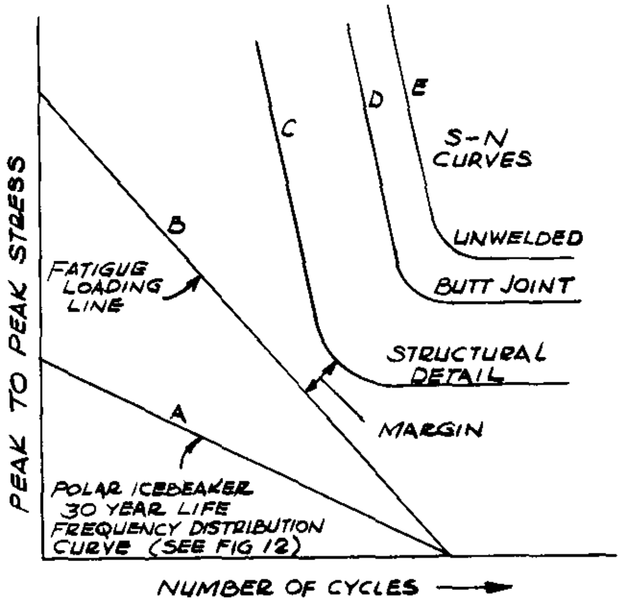


Figure 13. Schematic fatigue design curves, from [2, 27, 37]

A further limitation on the use of this method appears to be the lack of

APPLICATION	REQUIRED MATERIAL
Low-temperature	
Shell including Icebelt	CG-A537M
Ice frames	CG-A537M
Other shapes, Fabricated	CG-A537M
Rolled	CG-A537M or ASTM-A537A or ASTM-A537B
Flight Deck	HY-80
Other Weather Decks (Main & 01)	CG-A537M
Internal Structure Adjacent to Shell or Weather Decks (plate)	CG-A537M
(rolled shapes)	ASTM-A537A or ASTM-A537B
Surrounding Large Deck Openings	HY-80
Interior Structure Not Subject to Low Temperatures	ASTM-A131
Superstructure	ASTM-A131

Table IV. POLAR Class steel types

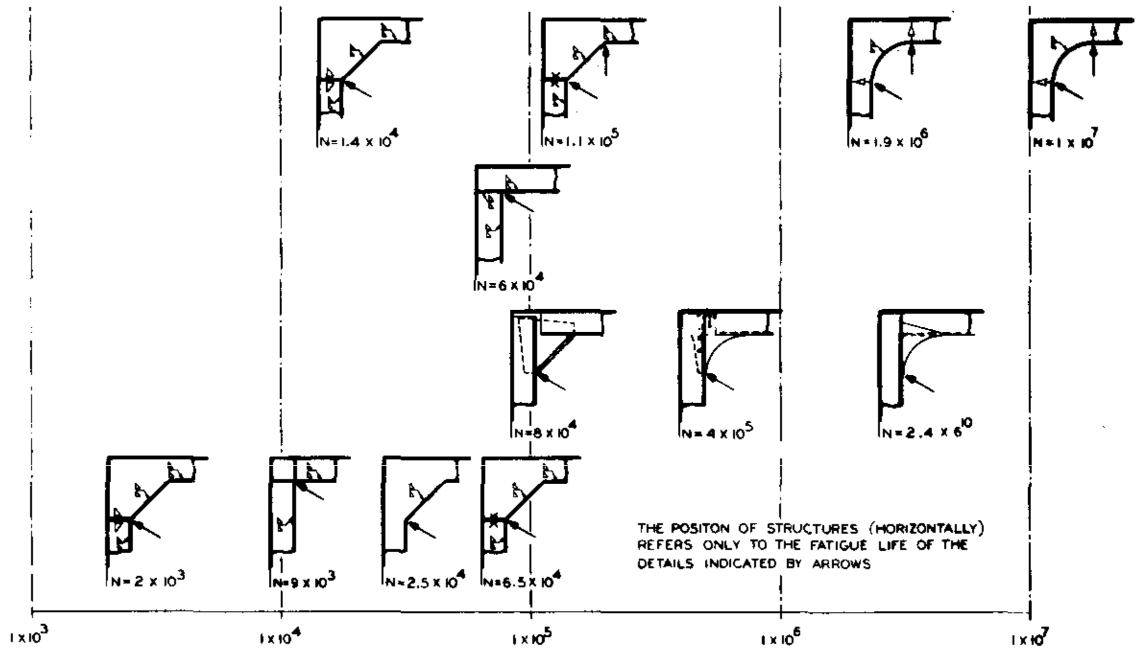


Figure 14. Relative fatigue resistance of structural details, from [27]

S-N curve data for specific material/ detail combinations under loading conditions identical to those aboard ship.

The data of [38] (Figure 14) was useful to the extent that it provided a relative ordering of various joint designs in terms of fatigue resistance. Despite the low cycle predictions, the most fatigue-resistant details were selected for structure which would be expected to absorb impact loads, keeping in mind the usual icebreaker hazards - extreme loads, low temperatures, the remote operating domain - and those normally expected such as material flaws, construction-related structural notches, and corrosion. Figure 15 illustrates typical joint designs provided on POLAR Class vessels.

A tenet of sound structural design is to insure that instability failure does not prevent a member from absorbing its design load. Lateral supports on all heavily loaded framing are incorporated into the POLAR Class structure to preclude such failure.

Additional detail considerations worth noting are that hull frames are continuous through decks and that longitudinal structure is continuous through bulkheads. A general requirement, in fact, is that plate not be loaded through its thickness in order to eliminate the possibility of lamellar tearing.

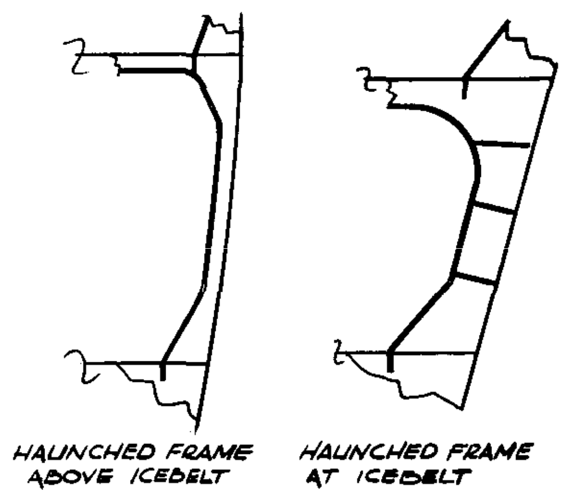


Figure 15. Typical POLAR Class shell framing details

COMMENT

More extensive use of icebreakers and ice-navigating ships is forecast for the coming years. Various rule-making bodies have responded with additional or more severe structural requirements. The Canadian Arctic Shipping Pollution Prevention Regulations require icebreaker hulls capable of withstanding pressures as great as 1,500 psi without exceeding the yield strength of the hull structure. The American Bureau of Shipping has outlined requirements for new

Ice Classes requiring shell design ice pressures of up to 234.5 psi for navigation in "extreme ice conditions". The latter requirements alone are difficult enough to attain. They are comparable to or exceed the actual strengths of many of today's "polar" icebreakers. The need for further research was never more apparent than now if these requirements are intended to specify the true load-bearing capability of a ship's hull.

As Figure 16 indicates, current data on the world's icebreakers continues to point to a correlation between ship displacement and plating thickness. Curves A and B describe older and more recent trends respectively for arctic icebreakers, while curve C shows the present trend for Baltic and Great Lakes icebreakers. Note that only curve C tends to indicate a limit on plate thickness.

Neither WIND Class ships nor GLACIER have sustained damage that can be attributed solely to plating failure. Deformation between frames is not apparent even after thirty years of Arctic service accompanied by 15 to 25 percent plating deterioration. It therefore appears that perhaps this plate has never been loaded to the point that further increases in thickness will be justified. This issue has immense impact when one considers that one-eighth inch of shell plate on POLAR STAR costs some 90 tons and one-quarter-million dollars. Assuming adequate strength and ductile behaviour of the material, how much plastic deformation should be tolerated - structurally or even aesthetically - under extreme loads?

It appears that the design of framing can likewise be improved. Figure 17 indicates the relative strengths of shell and framing for representative ships, both existing and hypothetical, and compares the load-carrying ability of plating designed plastically, plating designed to yield, and framing designed to yield. The consistent differences between shell and framing strength, viewed in the light of Coast Guard experience, appear to require additional study. The framing strengths shown assume that decks and bulkheads provide unyielding support for transverse frames and longitudinal girders. This is an optimistic assumption, particularly for icebreakers where decks and bulkheads are not necessarily normal to the shell in the areas of greatest load. Furthermore, the framing strengths of Figure 17 do not account for the reduction in load-bearing capacity caused by frame instability under heavy loads. This is due to shapes which are inherently unstable because of section asymmetry (such as inverted angles) or to frames which, even though symmetrical,

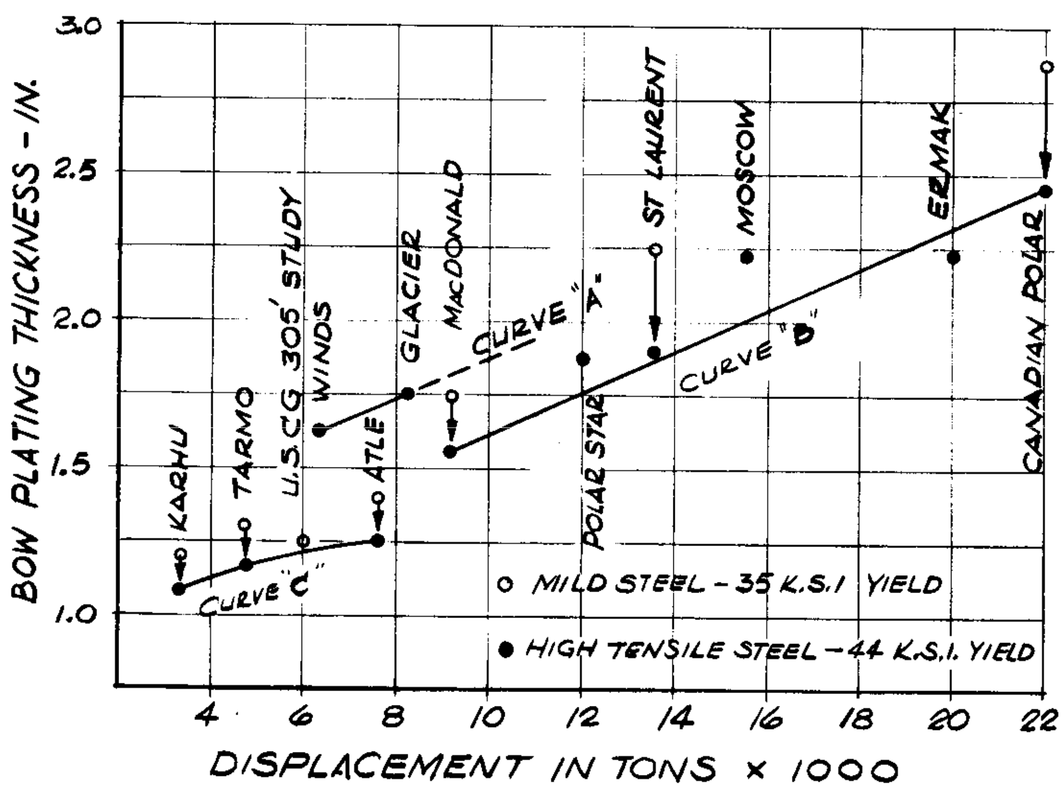


Figure 16. Icebreaker shell plate thickness as a function of displacemnt

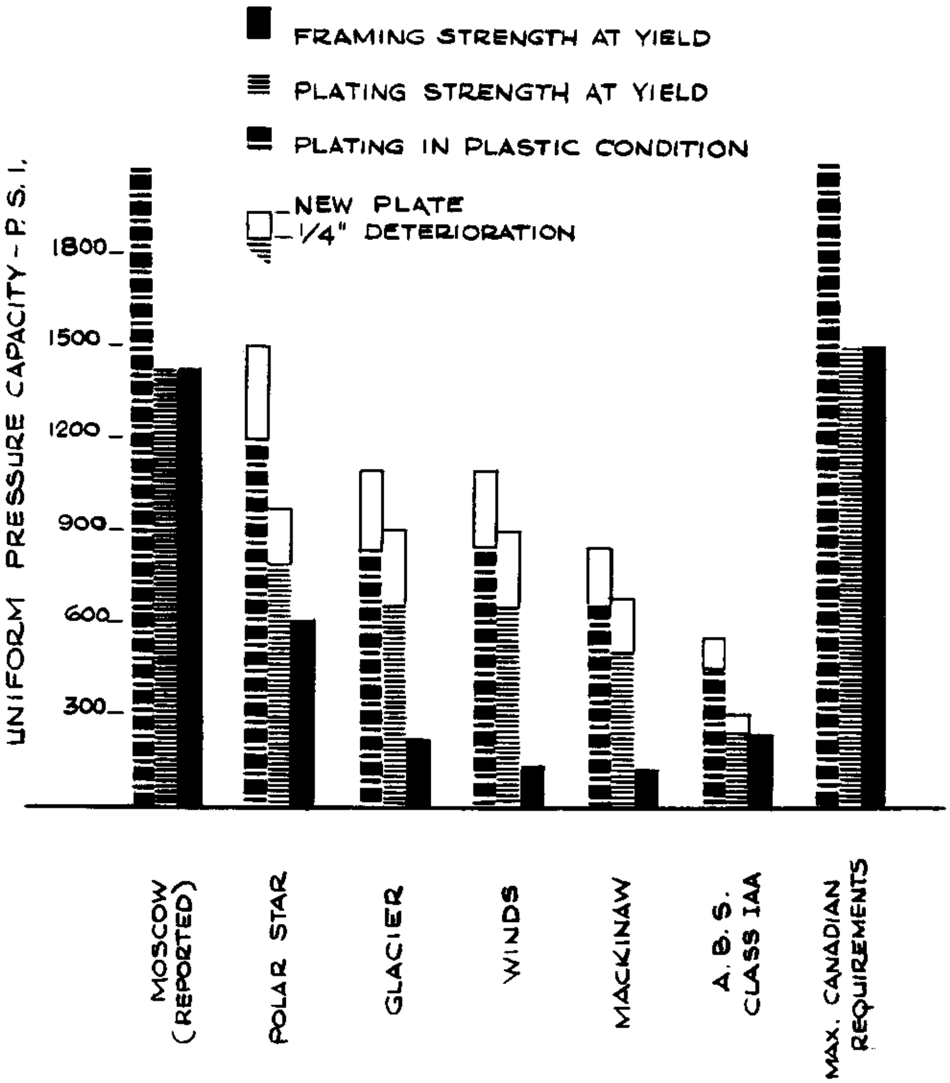


Figure 17. Relative strengths of icebreaker bow shell structures

cannot be installed normal to the shell. The enlarged detail of Figure 18 illustrates the situation as found in varying degrees throughout the length of any ship, particularly those transversely framed, yet its effect on the ultimate strength of the plate-stiffener combination has yet to be assessed through the application of research.

Structural fabrication cost must be further addressed, including consideration of weight, shipyard capabilities, and framing system options available to the designer. Typical might be a study of framing systems such as that shown by Figure 19. This illustrates the conceptualized framing of an icebreaker of about 300 feet overall length, and shows longitudinally framed grillages in the less contoured areas of the hull, allowing more extensive use of standard rolled shapes. While somewhat heavier, the design minimizes the costly custom fabrica-

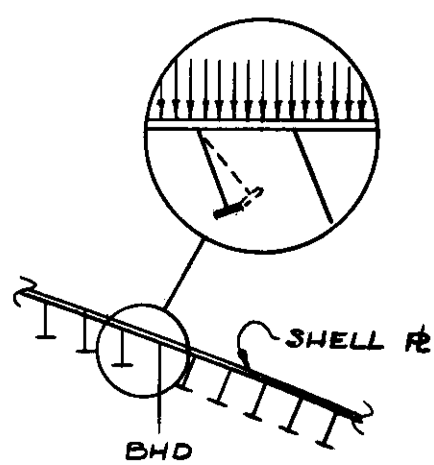


Figure 18. Shell framing not normal to the plate. Combined beam and plate section modulus does not represent the true ultimate strength in this situation.

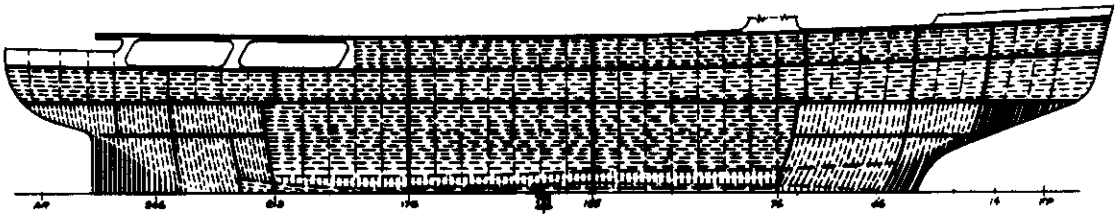


Figure 19. Conceptual longitudinal framing system with cant frames fore and aft

tion of frames necessary with a transverse framing system.

SUMMARY

This paper has only highlighted the extensive research effort that attended the POLAR Class structural design. The resulting cutters, although yet unproven, are expected to accomplish the Coast Guard's missions with the utmost in structural reliability. This work provides a foundation upon which future designs, with new and additional requirements, will be based.

Changing design criteria are imposing new constraints on all ship designs. Cost ceilings, once flexible, are now unyielding. Prevention of pollution is mandatory. More reliable performance is expected as new missions are added while manning levels are reduced. The conscientious designer can no longer rely on intuition or on the extrapolation of past concepts to solve the new problems. He must actively and objectively seek out sound bases for his choices. The POLAR Class design is but one example of his dependence on the research establishment for the necessary answers to his inevitable questions.

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