SSC-206

PERMISSIBLE STRESSES AND THEIR LIMITATIONS

SHIP STRUCTURE COMMITTEE

1970

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SECRETARY
SHIP STRUCTURE COMMITTEE
U.S. COAST GUARD HEADQUARTERS
WASHINGTON, D.C. 20591

1970

Dear Sir:

Mr. J. J. W. Nibbering has presented a lecture, entitled *Permissible Stresses and Their Limitations* during a post-graduate course on naval architecture at the Delft University of Technology that appeared as Report No. 141 of the Ship Structures Laboratory, dated April 1969. This report is being reprinted by the Ship Structure Committee, with the permission of the author, because it represents an important synthesis of some of the current research being conducted by the Committee.

Sincerely,

W. F. Rea, III
RADM, U. S. Coast Guard
Chairman, Ship Structure Committee

SSC-206

Special Report

PERMISSIBLE STRESSES AND THEIR LIMITATIONS

by

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Washington, D.C.
U.S. Coast Guard
1970

ABSTRACT

Various aspects of capability and demand of ships such as extreme loads, cyclic loads, plastic design, crack design, collapse and damage are discussed in an attempt to make a synthesis. It is explained that whenever permissible stresses are used in structural design, they should be bounded by probability-concepts, deformation criteria and critical crack lengths. For an acceptable risk factor, the margin between capability and demand seems to be substantial in ships. Yet drastic reductions in structural weight will only be possible if the principle of "fail safe" design is adopted in shipbuilding to the same extent as in aeroplane-building.

I. INTRODUCTION

Permissible stresses have for a long time been used as a flag that had to cover a large variety of cargo. In them were incorporated large parts of the more modern concepts of capability and demand. In fact capability and demand are still largely described in terms of stresses. But it is more and more realized that (permissible) stresses can never stand alone. In the first place they have to be accompanied and bounded by probability-concepts. Secondly permissible deformations or deflections have to be incorporated. Thirdly permissible crack-length will become a parameter of which the importance will become as large as permissible corrosion.

Apart from this, permissible stresses are of restricted value in plastic design, both where the formation of full plastic hinges is concerned as of partly developed hinges. It will be argued in section 3a of this paper that only the latter condition is of significance for ship structural design.

A discussion on the above mentioned aspects of capability would be impractical without considering loads. Therefore, the first part of this paper is devoted to loads with special emphasis on extreme loads. The latter is a subject which is difficult to grasp because the relevant information is scarce. An attempt is made to select figures for the extreme stresses which will roughly occur about once in a thousand shipyears.

The estimation of reliable long-term distributions has for a long time been the goal of many investigators all over the world.

It seems that not everyone has been fully aware of the fact that the value of this work for practice depends on whether a fatigue-problem in ships exists or not. This will be discussed in section 3b. It will be seen that while nowadays the danger of small cracks is a significant factor in structural design, in future the danger of large cracks will become a principal factor. The growth of these cracks is governed by the local stress-field in the immediate vicinity. It can satisfactorily be described with the aid of fracture-mechanics. The risk of unstable extension of these cracks will become decisive in establishing the capability of ships.

It will be clear that when handling this problem, again realistic values for the extreme nominal stresses, which can be expected with a certain low probability, are indispensable.

II. EXTREME LOADS IN SHIPS

In the past the values used as permissible longitudinal wave bending stresses were specifically intended to be applied in combination with standardized loads simulating wave bending. When practical experience proved that the method had to be adjusted, this was initially effectuated by changing the permissible stresses. Later it was tried to modify the standard wave into a more realistic expression. Originally the standard wave height was proportional to the ship's length (L/20); later the exponent of L was reduced, first to 0,5 ($1{,}1\sqrt{L}$) and later to 0,3. Theoretically more justified are formulas of the type $c.Le^{x}$, where c is a constant and in which x, according to Nordenström[1] should be equal to -L/885.

It is important to know if these improved standard loads give a realistic approximation of the maximum wave bending moment that can be met by a ship. But the difficulty is: what is realistic? Should the standard waves represent the most severe conditions a ship can ever meet in her life? Certainly not, it is hardly possible to define such a load in an absolute sense. Only a few indications are available. Dalzell[8] has created in a towing tank wave systems which gave rise to bending moments of a magnitude up to three times the standard wave moment. Getz expressed the opinion that the height of the highest waves of ships' length conforms to L/7.[3] The point is that it does not matter if these situations are physically possible or not; what matters is whether there is a fair chance that they occur. Thanks to the many investigations made in this field we know that this chance is extremely small. Perhaps one out of all ships now in service will suffer once in her life an extreme bending moment of the order of three times the standard wave moment.

It will be clear that it is senseless to use something so extraordinary as a calculation norm for each and every ship!

Many investigations have proved that the standard wave methods nowadays in use represent on the average rather well the maximum wave bending moments ocurring in ships (fig. 1). From this, one might conclude that about half of all ships will have to sustain higher loads in their lives. This is illustrated in fig. 2a and 2b.

Figure 2a is taken from a paper written by

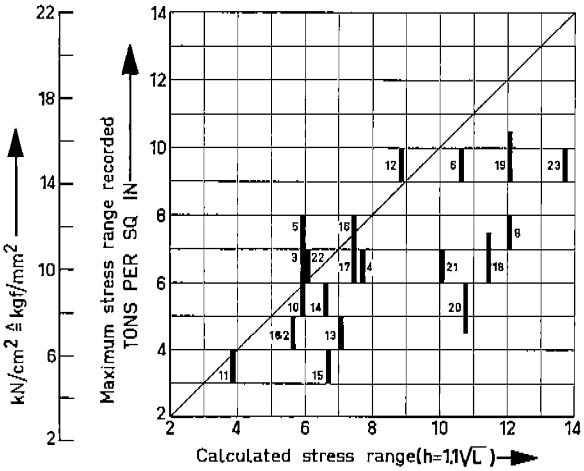


Fig. 1. Comparison of Calculated Stresses with Measured Maxima. (Johnson, Larkin[6])

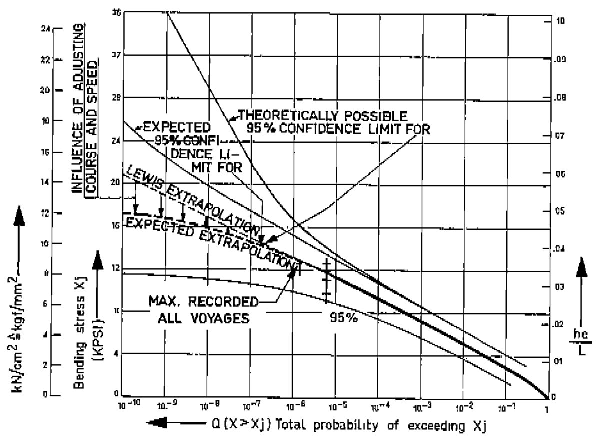
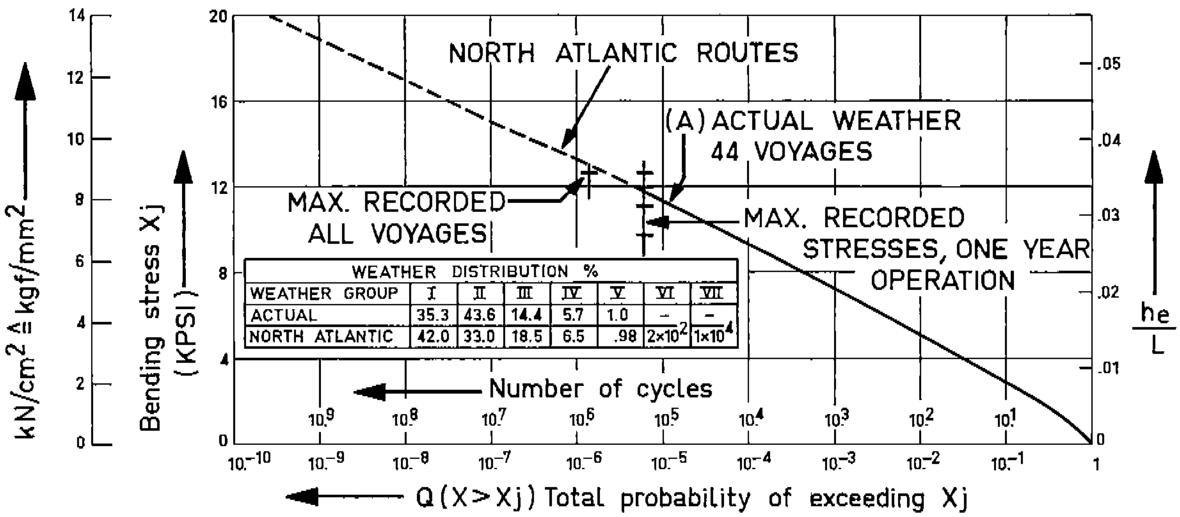


Fig. 2b. Original and Modified Long-Term Distributions with Respectively Theoretical and Supposedly More Realistic Confidence-Limits.



WEATHER DISTRIBUTION %							
WEATHER GROUP	I	II	III	IV	V	VI	VII
ACTUAL	35.3	43.6	14.4	5.7	1.0	–	–
NORTH ATLANTIC	42.0	33.0	18.5	6.5	.98	2×10^{-2}	1×10^{-4}

Fig. 2a. Long-Term Probability of Exceeding Peak to Trough Stress Values in Several Assumed Weather Distributions. *S. S. Wolverine State* and *Hoosier State*. (Lewis[2])

Lewis.[2] The subject was mainly long-term distributions of bending moments, while the following observations concern more extreme loads. The method by which the curve has been calculated is essentially sound and has been explained brilliantly in the paper. Yet for the dotted part of the curve a reservation must be made. The continuation of the full part of the curve might be different from what is indicated. This is not only based on considerations given further down, but also on fig. 7 of Lewis' paper made by Hoffman in which are shown the highest expected bending moments in 20 years North Atlantic service for several weather groups. The maximum peak to peak stress is 16 Kpsi, while for the same 20 years it is 17 Kpsi in fig. 2. Also in the appendix III of the 1967 report of the ISSC-committee 2bII it is shown in fig. 3,4 that the extreme loads measured in severe sea-conditions are much smaller than the calculated ones. The same is valid for comparisons made by Aertssen.[17] The differences are so large that even when allowance is made for the fact that Lewis' calculations are more refined than the latter-mentioned ones (heading and angular spread of energy have been taken into account) the tendency will presumably remain. One reason may be that a ship in a storm is intuitively navigated more favourable by the crew than is normally expected.

It is necessary to realize that even if the dotted part of the curve would be a good estimate of what on the average can be expected for the type of ship concerned, there is always a large possibility that it turns out otherwise. This can be taken into account by indicating confidence limits as shown in fig. 2b.

These are tentative because although much work is being done in this field the approaches differ appreciably. Therefore the following has mainly the purpose of an illustration.

At the end of the region covered by measurements, -- that is at about $Q = 10^{-6}$ -, the theoretical upper 95% confidence limit might deviate some 20 to 40% from the dotted line.[4] This means that there is a chance of 1:20 that 20 to 40% higher bending moments occur than suggested by the curve. A ship's life conforms to $Q \approx 10^{-8}$. There the theoretical upper 95% confidence limit may run perhaps 100% higher than the wave-stresses line. It will be seen further that this is unduly pessimistic but it demonstrates nevertheless that talking about the maximum load of a ship is not correct. It must be added how often the ship will meet that load and how accurate that statement is.

The consequence is that a permissible stress "sec" has no sense either. The lower the probability that a load used as a design load occurs, the higher the permissible stress can be taken. It will be clear that in this procedure any need of safety factors disappears, especially when "capability" in the same way as "demand" would be defined in terms of probabilities.

It has been stated that the standard wave methods actually in use represent loads which on the average occur once in the lifetime of a large number of ships.

The consequence of the foregoing discussion would be that about one out of 20 ships might meet a load twice as large. At this stage some sober reasoning is required in order to arrive at engineering solutions. As long as nobody has the opportunity to take measurements on an appreciable number of ships during their whole lifetime, statisticians will not allow bringing the confidence limits much closer together than has been indicated. One cause might be that in a prediction not all variables can be taken into account, especially human factors. But is it necessary to require that each individual British, Swedish, American, Norwegian, Belgian, Japanese, French, Russian and Dutch institute gathers sufficient material before a reliable estimate about the extreme loads for the ships they investigated can be made?

A different approach can be of help. If one could combine all measured data, it would certainly cover a few hundred -- and after statistical treatment -- a few thousand shipyears. Then it is possible to state that the largest wave bending moments ever measured in all these ships, will give a fairly accurate approximation of the maxima which can be met once in a thousand shipyears.

Well then, as far as the author knows the highest vertical wave bending stresses (peak-to-trough) ever measured are about 17 kN/cm^2. (Bennet[5]; Johnson, Larkin[6]).

It is interesting to note that in both cases mentioned the stresses were still 2 kg/mm^2 lower than calculated for a standard-wave height 1 ,$1\sqrt{L}$ (fig. 1). Nevertheless they are real extremes because for most ships the measured maxima were in the order of 10 kN/cm^2.

From this it can be concluded that firstly the confidence limits of the long-term distribution of fig. 2 should not deviate so much as indicated and again that the dotted extrapolated part of the stress-curve is too pessimistic.

A casual confirmation of this is shown by the point "max. recorded all voyages" in fig. 2, which is situated a little below the stress-curve. Of course a much more refined analysis based on the idea of combining results of as large a number of instrumented ships as possible can be made by discriminating between types of ships, trades, weather etc. This will result in different extremes for different cases. But for the purpose of this paper the figure of 17kN/cm^2 is thought to be sufficiently accurate. Another point is that it only applies to vertical bending; so it should be corrected in order to take into account horizontal bending and slamming. Especially with respect to the latter this is very difficult. It is however highly improbable that whenever an extreme vertical wave bending stress occurs the horizontal bending stresses and the slamming induced stresses are also extreme and the peaks of the three components of the load coincide.

Therefore for the same chance of once in a thousand shipyears it is assumed that a figure

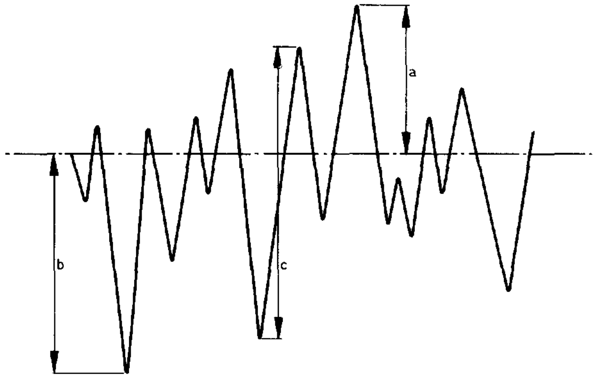


Fig. 3. Correction to Peak to Trough Values.

of say 20 kN/cm^2 will represent for the moment the extremes of the combination of vertical and horizontal bending stresses and slamming stresses. As very little information about ships in very severe storms is available, this value will further be compared with results of measurements obtained for a 23.000 tons bulk-carrier which has weathered a storm of magnitude Beaufort 11-12 for two days. The ship was instrumented by the Delft Ship Structures Laboratory. It sailed in ballast from Rotterdam to Port Churchill - Canada. The output of two strain gauges on both sides of the deck has been recorded on photographic paper and on punched tape for periods of 5 minutes every hour. Two distinct maxima were found, both equal to 12kN/cm^2. This is much smaller than the above-mentioned 20 kN/cm^2. In one of these cases no slamming had occurred, so that the value was due to combined horizontal and vertical wave bending only. In the other one slamming had clearly contributed with 3 kN/cm^2 to the total of 12 kN/cm^2. A further statistical analysis of the recordings showed that it is very unlikely that this value was exceeded during the stormy period by more than 2 kN/cm^2. Consequently the highest maximum stress for combined horizontal and vertical wave bending and slamming will be 14 kN/cm^2 for the observation period of 2 days. This value again is appreciably smaller than the previously derived 20 kN/cm^2 what confirms that the latter is not likely to be an underestimation and can be used for further analysis.

The value 20 kN/cm^2 represents a peak-to-trough value (fig. 3). The sum of the largest individual wave bending tensile stresses and the largest compressive ones during a storm-period can be some 10% larger (fig. 3). This leads to 22 kN/cm^2. It is assumed that a value of about 12,5 kN/cm^2 represents the maximum sagging component that can occur. The maximum hogging one will be smaller, but this is neglected for the sake of simplicity. Finally an extreme still water bending stress of 7,5 kN/cm^2 is added which results in an absolute extreme tensile peak of 20 kN/cm^2. The chance of meeting higher still water stresses of course is greater than once in a thousand ship-years, but what matters is that the combination of the 12,5 and 7,5kN/cm^2 has no higher probability of occurrence than once in a 1000 ship-years.

Of course this analysis has not the pretention of being refined. Local stresses have not been considered. On the other hand for most ships a lower value than 20 kN/cm^2 will be representative but higher ones are very unlikely. For the purpose of this paper it is of primary importance to avoid "underestimation" and that condition is certainly fulfilled when 20 kN/cm^2 is used in the following paragraphs on capability. There it also will become clear why the emphasis in this section has rather been laid on stresses than on effective wave heights.

Finally for research-people a conclusion may be added. It seems that the time has come to supplement the search for extreme bending moments by investigations directed to the problem how to avoid extreme bending moments and movements in extreme sea-conditions.

With such information it will be possible to provide futural ships with small computers fed by stress- and movements-indicators, which either depict suitable course-speed combinations for the crew or guide the ship automatically.

Experiments in model tanks can be of great help. In fig. 4a and 4b a simple analysis made by the author on recently published test-results by Maniar and Numata[15] illustrates the point. In fig. 4a the indications max (+) and max (0) and the comparison values 1, 1,25 and 1,7 have been added to the original diagram. Moreover fig. 4b gives simple arithmetical means of the original plots. Although this treatment of the results is theoretically not correct, it gives a good impression of what benefit can on the average be expected from changing speed or course. The hogging moments prove to be particularly sensitive to the latter. Both fig. 4a and b show that in extreme bad conditions navigating in the direction of the waves is clearly advisable. But it must be admitted that in fig. 13 of the before mentioned paper by Lewis[2] this tendency is much less pronounced which confirms the need for extensive research.

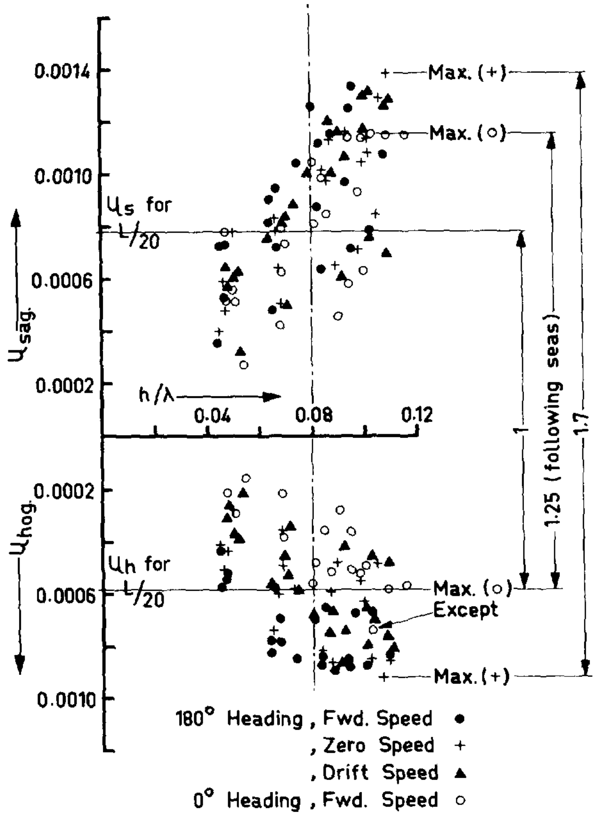


Fig. 4a. Bending Moments Variation with Wave Steepness (N.M. Maniar; E. Numata[15]) (Indications "Max" (O), (+) and Comparison Numbers 1, 1.25 and 1.7 on the Right are added).

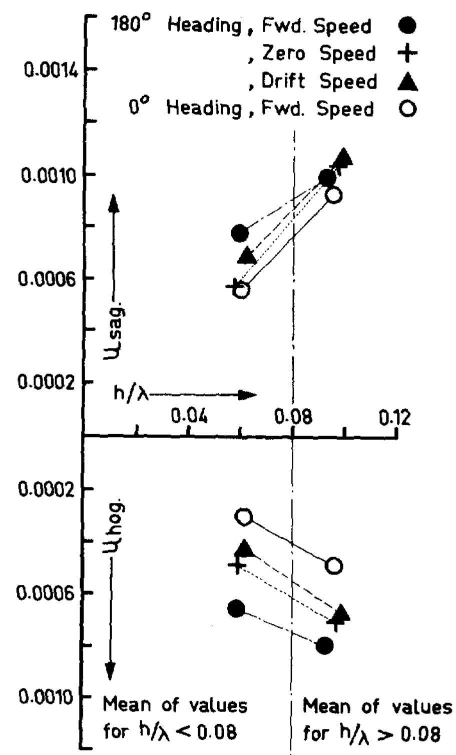


Fig. 4b. Arithmetical Mean of Data of Figure 4a.

III. THE CAPABILITY OF SHIPS

a. Compressive strength.

The capability of ships has for a long time been expressed mainly in terms of stress. Other factors like deflections, vibrations and corrosion also played a role, but not as substitutes for stresses but rather as complements to them. Concepts like permissible plastic deformations from a plastic-design-point-of-view and permissible crack lengths from a crack-design-point-of-view were not included. Another new idea that capability similarly to demand, should be defined in terms of chance, is also essential. Capability has its own variability due to fluctuations of yield point, plate thickness, unfairness of plating, quality of design and welding, etc. Unfortunately (in a sense!) the influence of these variables can hardly be judged on account of practical experience, because few structures collapse and when they do, the relevant loading conditions are mostly unknown.

Notwithstanding this, damage reports of course are of great help. For the rest structural test laboratories often are the only resource for arriving at realistic design criteria.

The domain where the significance of permissible stresses is rather trivial is that of plastic design.

It is well known that the ultimate bending load a beam can carry is appreciably larger than $M = W \times \sigma_y$. For a beam with rectangular cross section the load necessary for the development of a full plastic hinge is $1{,}5\ W.\sigma_y$. The total energy needed for creating that situation is many times larger than the elastic part of it because at plastic hinges appreciable deformations occur. For ships it is doubtful whether the capacity for deformation is in the same order of magnitude. This will be discussed in this section.

A ship is not a massive beam, but a hollow

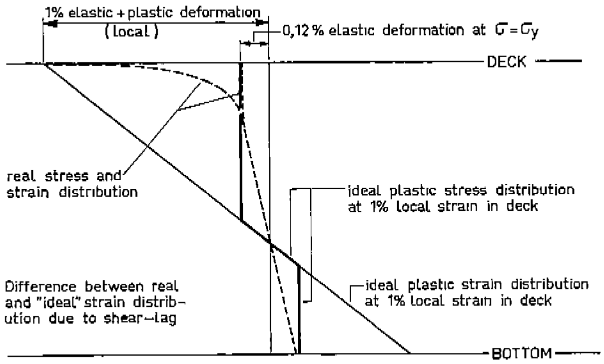


Fig. 5. Stress and Strain Distribution in the Side-Plating When 1% Strain Has Developed Over a Narrow Transverse Strip of Deck- Plating.

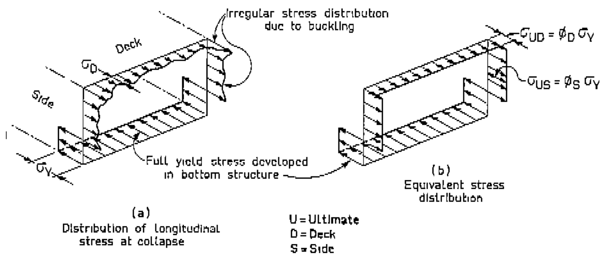


Fig. 6. Ultimate Stress Distributions-Buckling Included-Sagging Condition. (Caldwell[9])

one. Due to that the plastic reserve strength is much smaller than the 50% mentioned before. For a big tanker it will amount to 15% and for a ship with large openings in the deck, like bulk-carriers and containerships, some 30%. However, even these low figures are still too optimistic to be used for design calculations for two reasons. In the first place, the total deformation of a ship after the formation of a full plastic hinge is unacceptably large. Secondly the capacity for tensile deformation of most structural details, as well as the compressive strength of stiffened plates is generally insufficient for the formation of an ideal plastic hinge. Consequently it is necessary to indicate which deformations can and may occur. Permissible deformations then replace permissible stresses. In that case it is better to speak of limited plastic design instead of plastic design. Based on tests carried out in the Ship Structures Laboratory[7] it has been suggested to the ISSC-committee 1967 "Plastic design" that deformations larger than 1% are not likely to develop in ships prior to fracture or alternatively without a great reduction in compressive strength. This might seem a large value if compared to the elastic strain in a structure at a nominal stress equal to yield point ($\varepsilon = \frac{\sigma_y}{E} \approx 0{,}125\%$ for mild steel). But it should be realized that there is a big difference between the case of 1% strain developed over a length of 0,5 m in a ship's deck* over say 10 m. In the first case no plastic hinge will be formed; only a small part of the side plating will find itself in the plastic condition. This is explained in fig. 5. The total deflection of the ship is only a few percents greater than in the situation where the nominal stress in the deck just approaches yield point.

* In the following only "deck" is mentioned although often "deck or bottom" is meant.

This forms a big contrast to the case when a large part of the length of a deck, say some 10 m, is plastically deformed by 1% over its entire breadth. Then the deflection of a large ship will be about two times the maximum elastic deflection. An additional favourable factor is, that in practice these deflections will not develop the very first time an extreme load of the required magnitude occurs. The time during which the load is maximum, is too short, especially when a part of the load is due to slamming. However as mentioned before, it is very unlikely that a plastic deformation will develop over an extended length of a ship. It will always start at one or another "weak chain" in the form of a locally less efficient structural design detail, a corroded area, a section containing openings or with unfair, and therefore less effective plating etc. Due to that the situation of fig. 5 represents what really can be expected and the plastic reserve strength energy of a ship is much smaller than generally assumed.

It will also be clear now that the requirement that 1% plastic deformation should be able to occur without fracture or collapse at critical places is really a minimum.

Before the other part of the problem will be discussed, viz., on what tests or theory the mentioned 1% is based, an extension of the classic plastic design theory developed by Caldwell needs attention.[9] His basic idea was that the behaviour of a compressed stiffened panel at the moment the buckling starts is very similar to that of a tensile bar at the moment yielding starts. In both cases the deformation can be increased greatly when only a small raise of the external load is effectuated. Of course many parameters are involved like beam spacing, plate thickness etc., but that is not essential now. Fig. 6 derived from Caldwell's paper shows the as-

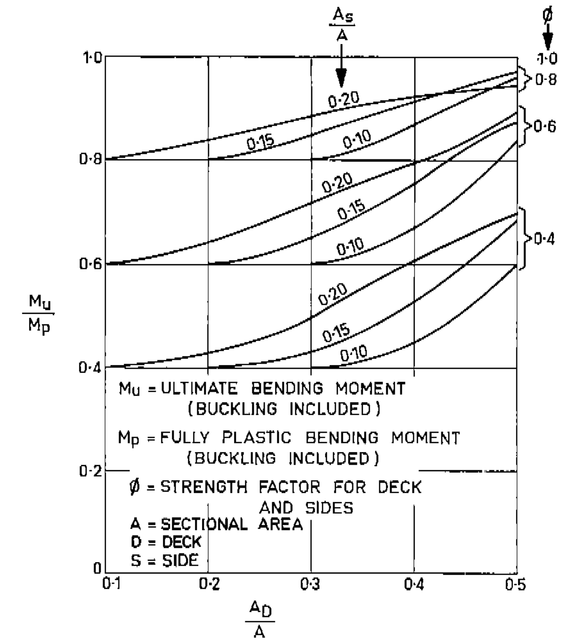


Fig. 7. Effect of Buckling on Ultimate Longitudinal Strength. Single Deck Ship. Sagging Condition. (Caldwell[9])

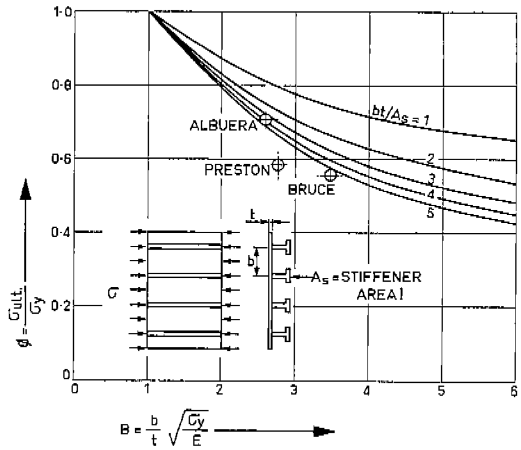


Fig. 8. Ø-Values for Stiffened Plating. (Faulkner[9])

sumed stress distribution in an extreme sagging condition. The bottom is in a state of complete plastic straining; in the deck only the corners are in full plastic compression, the rest is in a situation of elastic or plastic buckling at nominal stresses much lower than yield point. Caldwell has proposed to consider the average of the compressive stresses in side and deck as a fictitious compressive yield stress (fig. 6b). The relation between f_{UD} (U = ultimate, D = deck) and yield point f_y is called $Ø_D$ and is generally smaller than 1. When this $Ø_D$ is known, the collapse load of a ship ($M_{ult.}$) can easily be calculated with the aid of fig. 6b, (Provided of course that a full "plastic" hinge has developed!).

Fig. 7 shows the reduction in collapse load in relation to the ideal one ($Ø_D$ = 1) as a function of the buckling strength of the deck. For well designed large ships Ø is always larger than 0,8 and will often approach 1.

In the discussion on Caldwell's paper D. Faulkner has given a very useful contribution with respect to the Ø-values. This is shown in fig. 8. The diagram is easy to handle. Interesting is that the results of the only experiments ever carried out with complete ships, conform well to the curves (*Albuera, Preston, Bruce*). The Ø-values for these ships are surprisingly low. For the *Preston* and *Bruce* it will be due to the fact that they were built before World War II and were provided with transverse framing. The *Albuera* is more modern. The plating was rather thin which may partly explain her low strength. But the primary factor may be an unsatisfactory design of structural details. An example of how shipbuilders and other structural engineers have failed in this respect is given in fig. 9. It is a full-scale specimen compressed to collapse in the Delft Ship Structures Laboratory. The test conditions were slightly more unfavourable than in reality because the bulkhead could rotate freely and the longitudinal edges of the bottomplate were unsupported. On the other hand the ends of the specimen were fully clamped and lateral loads were absent. Fig.10, in which the test-result is shown, demonstrates that the behaviour of the *Albuera* is not so peculiar as often is thought. The maximum load the test piece could sustain was equal to 410 tons, being two thirds of nominal yield load; thus Ø = 0,67. This conforms rather well to the Ø = 0,71 for the *Albuera* in fig. 8. With the aid of figure 7 it is found ($M_{ult.}$)/($M_{plast.}$) = 0,79 (for AD/A=0,3 and AS/A=0,15). Thus far everything seems to conform with Caldwell's suppositions. However, the situation of fig. 6, being a fully developed hinge, will not have been reached at all in the ships concerned. It requires deformations in deck and bottom in the order of percents, while the overall deformation of the specimen of fig. 9 at the moment the load was maximum, was only 0,2%. Locally, -which means close to the bulkhead, it was more, say 1%, but it has already been explained before that plastic deformations in such a narrow strip are not able to bring the whole transverse section of the ship in a

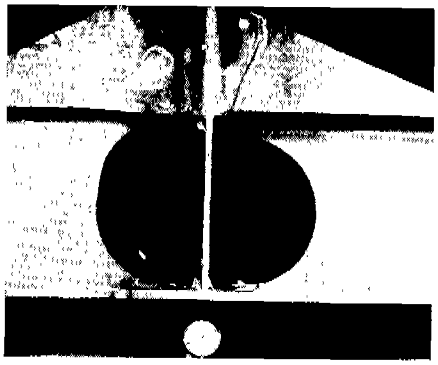


Fig. 9a.

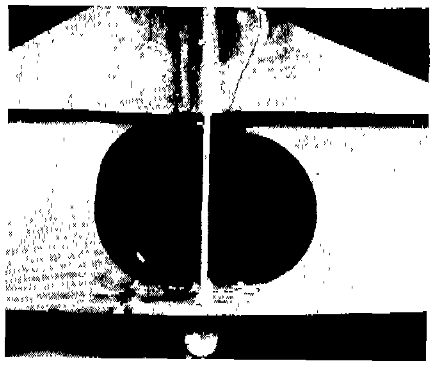


Fig. 9b.

Specimen at 400 and 410 Tons Compressive Load. Note Small Deformation at 400 Tons.

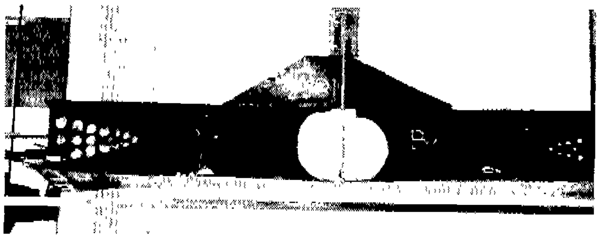


Fig. 9c. Specimen After Testing. Note Deformation of Bottom Plating.

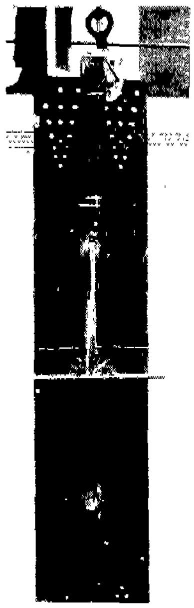


Fig. 9d. Specimen After Testing. Note S-Shape of Each Half of Through-Bracket.

as shown in figure 10 instead of as in figure 6. The section is very uncompletely plastic and the relation between the collapse load and the ideal plastic load will be lower than according to Caldwell. $(M_{ult.})/(M_{pl.}) \simeq 0,77$ instead of 0,79. More important is that the total energy to collapse is not several times larger than the elastic part of it, but only a few percents.

The low collapse load of the specimen of fig. 9 deserves some additional attention. Three factors are involved.

a Vertical bending of the specimen as a whole as a consequence of the shift in position of the neutral axis at the bracketed part.

In the unsupported part of the bottomplating at both sides of the transverse bulkhead, yielding (bending) started already at 250 tons.

b When the load is increased a vertical plastic hinge starts developing in the free parts of the bracket near the bulkhead. This is due to a second form of internal bending caused by shifts in the position of the pertaining neutral axis in a horizontal plane.

c The rigidity of bracket and upper part of the frame in a horizontal plane is rather small so that at about 400 tons the events described in b lead to horizontal, plastic buckling (fig. 9d). At 410 tons the specimen has attained its maximum load.

An unpleasant phenomenon was that during the continuing compressing of the specimen after the maximum load was attained, the load quickly decreased. At a total deformation of 0,3% about half of the maximum load was left (fig. 11). It must be concluded that Caldwell's hypothesis that the behaviour of ship structures under compression is similar to ideal plastic behaviour is not always justified, at least not for structures of the tested type.

The test-result demonstrates how easily the carrying capacity of astructure can be impaired by using brackets, asymetrical sections, etc. For a continuous T-frame, the collapse load would have been much larger; some 95% and not 67% of the optimum. In ships constructed accordingly the plastic reserve might amount to some 15 to 30% of the load at which the nominal stresses approach yield point. Of course this is only considered from the view-

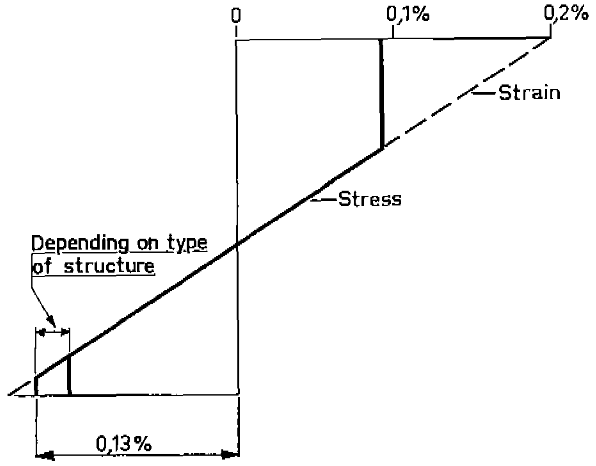


Fig. 10. Real Elasto-Plastic Strain Distribution in Contrast to that of Figure 6.

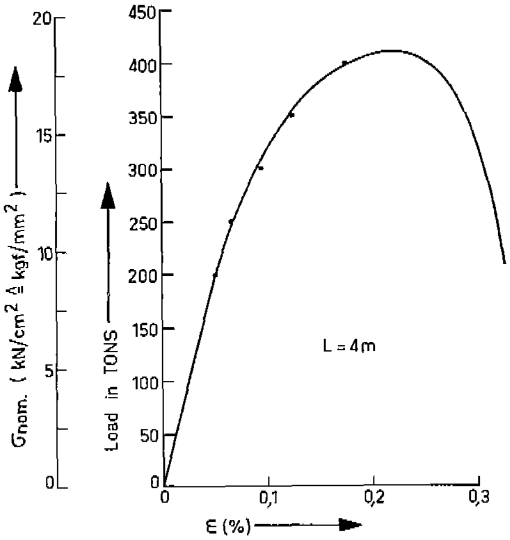


Fig. 11. Load-Deformation Curve of Compression Test.

point of compressive strength; the situation for tensile loading will be discussed later.

In section 2 of this paper an extreme longitudinal bending stress of 20 kN/cm^2 (20 kgf/mm^2) has been supposed to occur once in a thousand shipyears. When this value is compared to the yield point of Mild Steel being 26 kN/cm^2 (26 kgf/mm^2) there is a remarkable margin left. When the ideal plastic collapse load is taken in view the margin is still larger.

It can be doubted whether this is really necessary. Even for the worst designed ships, with $(M_{ult.})/(M_{plast.})$ as low as 0,77 and $(M_{ult.})/(M_{\sigma y}) \simeq 0,9$ ($M_{\sigma y} = W \times \sigma\ y$) the collapse stress is still 0,9 x 26,00 = 23,40 kN/cm^2 (kgf/mm^2).

This is clearly in excess of what is required particularly when it is remembered that plastic collapsing always needs more time than extreme loads are normally working.

The foregoing is another demonstration of the fact that where capability is concerned it should also be defined in terms of chance. Not all ships are badly constructed, so the chance that such a ship is the one that will meet the extreme of 20kN/cm^2 must be smaller than once in a thousand shipyears. The practical conclusion from the foregoing is that one should not worry about the compressive strength of the hull of current ships. The extreme loads which they can withstand are much larger than what is met in practice.

b. Tensile strength.

The foregoing can be interpreted in such a way that the classification societies do not take into account the mentioned collapse modes, because the danger of other modes of failure like brittle fracture or the damages caused by fatigue constitute a more real danger.

As far as brittle fracture is concerned, tests in the Ship Structure Laboratory[7] with bottom longitudinals of the type shown in fig. 9 have led to the idea that local plastic deformations in the order of magnitude of 1% can develop before fracture starts. Unfortunately this averaged over a few meters length amounts to not more than 0,2% (fig. 12). The nominal stresses at fracture were about 20 kN/cm^2 (kgf/mm^2). Now it is possible to introduce a $\emptyset$-factor for the tensile strength of panels in the same way as before for the compressive strength. $\emptyset_D$ then is equal to (20,00)/(26,00) = 0,77. $(M_{ult.})/(M_{pl.})$ will be about 0,83 and this corrected for the fact that in the transverse section concerned only an incomplete plastic hinge has developed, leads to 0,81; $(M_{ult.})/(M_{\sigma y})$ will be 0,93.

The collapse stress would be 0,93 x 26 ≈ 24 kN/cm^2. Similar to what has been concluded before for the compressive strength of ships, it seems that the margin between capability and demand with a risk factor of once in a 1000 years, being 24 - 20 = 4 kN/cm^2 is larger than necessary. However in the present case the situation is clearly less safe, due to additional unfavourable influences.

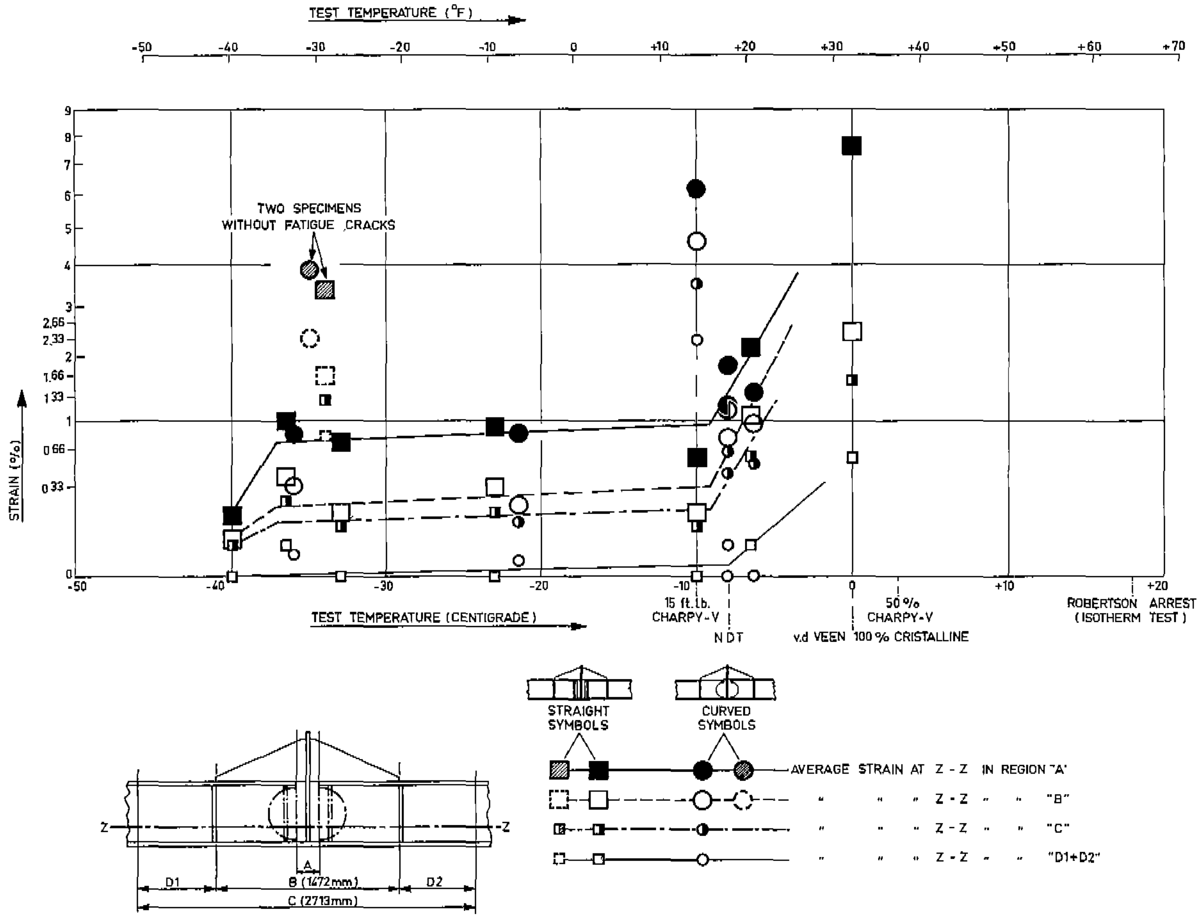


Fig. 12. Local and Overall Strains of a Specimen Containing Fatigue-Cracks at the Moment a Brittle Fracture Started.

a In the first place as soon as a brittle fracture starts, the capacity of the structure for taking up deformation energy is immediately exhausted; the chance of complete collapse through fracture is large.

b The high frequency of slamming stresses, which was beneficient in case of compression can be disadvantageous for tension.

c The ultimate tensile strength of a panel is more liable to scatter than the ultimate compressive strength. Structural and welding defects might reduce the above-mentioned fracture stress and-strain substantially.

The author's opinion that the danger of brittle fracture in ships must be eliminated by using crack-arresting steels has been explained last year at the "Construction-day."[10] It seems to conflict with the fact that 90% of all ships in the world move regularly and undamaged in conditions where the temperature is lower than the crack-arrest temperature of their steels.

The explanation is included in the foregoing. The nominal stresses mostly are so low that with present-day quality of design and workmanship brittle fractures cannot initiate and consequently there is no need for arresting. On the other hand it has been made clear that the nominal tensile stresses in ships are not too far away from the actual permissible limit. Consequently without the use of crack-arresting steels (and higher tensile steels) it will not be possible to raise the limit substantially.

The modern normalized Niobium-treated steels meet the indicated requirements very well, provided that they will not be welded with welding methods giving excessive heat-input (one-run Electro-Gas and Electro-Slag systems). For in that case, - as tests in the Delft Ship Structures Laboratory with 34 and 46 mm thick plates of St. 52 have proved-,cracks, initiated in the heat-affected coarse grained zone adjacent to the weld, do not deviate from this zone as usual and stop in the unaffected plate, but run parallel to the weld. The main cause of their remaining in the heat-affected zone is that the residual stress-pattern is different from that in multilayer-welds. In these cases one apparently cannot rely fully upon good crack-arresting properties of the steel.

So much for the brittle fracture problem. It is quite possible that within 10 or 20 years it has disappeared from shipbuilding. Then the level of permissible stresses will to a large extent be determined by fatigue considerations. In fact it does so already nowadays together with brittle fracture, buckling of bulkheads and webs of deep frames and bottom-damage due to slamming. It seems that not everyone is aware of this fact. There are even investigators, dedicating their time to wave bending moments, who are not much interested in fatigue. Yet 90% of their work would be superfluous when fatigue did not constitute a serious problem. For, what is the use of statistical material, calculation methods, model tests and what else, if the frequency-distribution of the wave bending moments would be of no use. Then only the extreme values would be of interest as once proposed by Yuille[11] and much of the actual research would have to be directed otherwise. However in [12] it has been shown to what extent fatigue is a problem in shipbuilding. One conclusion was that there is little danger that large cracks develop in actual ships. Therefore the pertinent investigations in the Delft Ship Structures Laboratory have been devoted to the question if the presence of small fatiguecracks enlarges the risk of brittle fracture[7]. In the foregoing it has already been mentioned that the answer was confirmative. Later it was found that the danger of fatigue is larger than even fatigue-experts are aware of. Some of the specimens mentioned before were subjected to a combination of small static loads and light transverse impacts. The result was really alarming because fractures occurred without any previous local plastic deformation. As a consequence of these results the laboratory is now working on the propagation and damping of shock-induced stress waves in structures and on slamming-induced stresses in ships. When in future the brittle fracture problem is eliminated, the stress level will no longer (partly) be governed by the requirement that fatigue-cracks must be kept small. Larger cracks can certainly be allowed from the viewpoint of human and structural safety. (Of course damage due to leakage also plays a role but this is excluded from the present discussion.) It is a favourable circumstance that fatigue-cracks propagate very slowly in ships' structures. Due to this they can be controlled easily. Even in aeroplane building, where permissible crack lengths are much smaller, and consequently less easily detectable, this philosophy of "fail-safe" design has become generally accepted and put into practice.

The question now is how large may cracks grow in ships without leading to an unstable fracture? The answer can be given with the aid of fracture mechanics. This is a new branch of applied mechanics of which many people have not yet taken notice. Its importance can be judged by the vast amount of literature published in the last 10 years especially in USA, Japan and Great Britain. An introduction to the field is given in [15]. The basic idea is that when a unit extension of a crack in an infinitely long and wide plate under tensile loading is effectuated, e.g. by sawing, an amount of elastic energy comes available which is used for deforming the material adjacent to the new crack-tip. The larger the crack, the more energy is set free per unit crack extension. At a certain length more energy is set free than would be needed for rupturing the material at the crack tip over one unit of length and an unstable fracture starts. The point is illustrated in fig. 13a, b, c, d, where especially the influence of plate length is considered. The elastic energy per unit crack extension is called G (strain energy release rate) and is equal to $G = \frac{\pi a \sigma^2}{E}$; a = half cracklength plus half of the length of an eventual plastic zone at the crack tip. The energy needed for fracturing the material one unit of length at the crack tip is called G_c. It is equivalent to the area of stress-strain curve of an imaginary small test bar situated at the crack tip. Thus G_c is a material property and is called fracture toughness. For steel it is very much dependent on temperature and strain rate. The G_c belonging to a low-stress brittle fracture is tenths of times smaller than that for a shear fracture. Hence a ship either fractures brittle or does not fracture. Shear fractures are very seldom found. In future the situation might become different. When larger fatigue-cracks are permitted, the risk of unstable shear fracturing might become real. How real this danger is can be estimated in the following way. For a 30 mm thick mild steel plate a lower value for G_c (shear fracturing) is about 5,00 kN/cm. The nominal stress at which an existing crack with a length of 1 m becomes unstable is equal to

$$\sigma = \sqrt{\frac{E.G_c}{\pi .a}} .$$

a = half crack length plus half size of plastic zone = r_y.

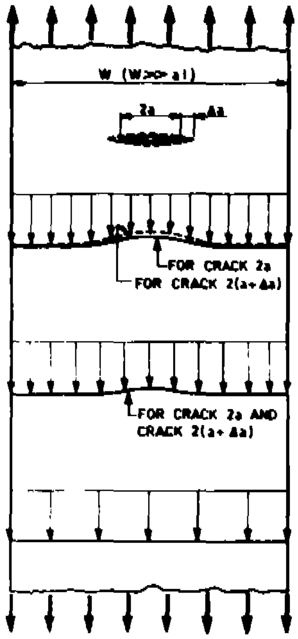


Fig. 13a. Stress Distribution in Long Plate Before and After Crack Extension.

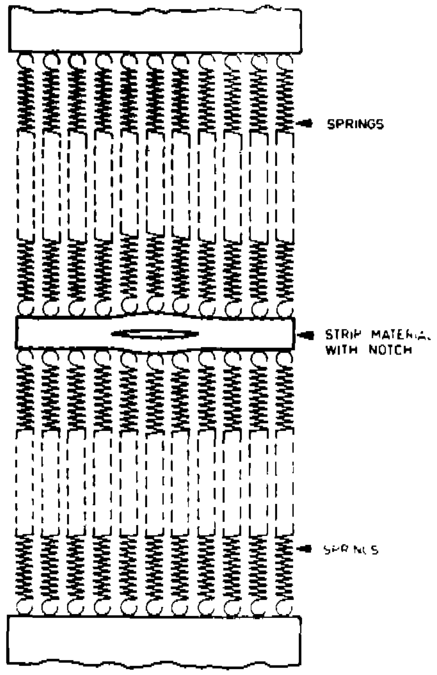


Fig. 13b. Stress Distribution at Loaded Edge of Short Plate, Before and After Crack Extension.

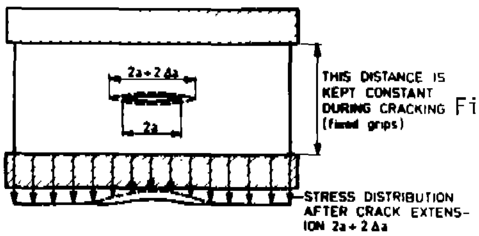


Fig. 13c. Model of Long Plate with Crack.

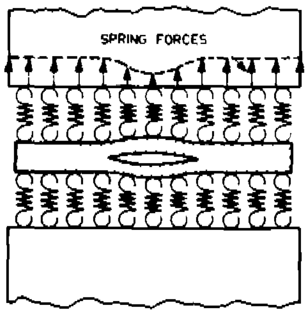


Fig. 13d. Model of Short Plate with Crack.

Models of Fig. 13c & 13d are much simplified if compared with real plate, because (horizontal) connections between springs are left away.

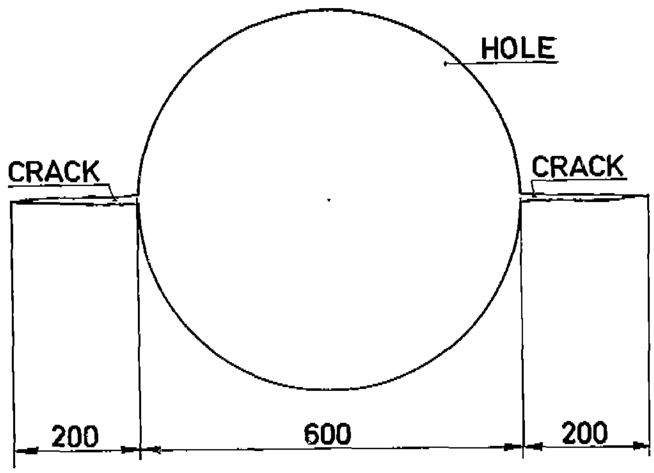


Fig. 14. Crack-Lengths 200 mm. Effective Crack-Length 1000 mm.

$$r_y \quad \frac{a\sigma^2}{2\sigma_y^2} \longrightarrow \quad \sigma \simeq \sqrt{\frac{E.G_c}{\pi\left(a + \frac{a\sigma^2}{2\sigma_y^2}\right)}}\ .$$

For $\sigma_y = 28$ kN/cm^2 (kgf/mm^2) $\longrightarrow \sigma \simeq$ 22,50 kN/cm^2.

So an unstable fracture may develop at a nominal stress of $0{,}8\sigma_y$! For some higher strength steels G_c can even be smaller than for mild steel. When $\sigma_y = 35$ kN/cm^2 and G_c = 4.50 kN/cm the critical stress σ_c is equal to 22.20 kN/cm^2 being 64% of yield stress.

One might object that a crack of 1 m in length is abnormally large. In the first place it is only a matter of getting accustomed to it. Secondly it should be realized that the effective crack-length for the situation depicted in fig. 14 is also equal to 1 m, despite the fact that the real cracks are only 20 cm long. It should be observed that the above used G_c-values are excessively low. For most modern steels it is two and more times larger.

Nevertheless from the foregoing it will have become clear how important "crack-design" in shipbuilding will become the more the level of stresses is raised and the more higher strength steels are used. Consequently prolongation of all the efforts directed to the determination of long term distributions of wave bending moments and intensification of fatigue-investigations on welded structures, particularly under programmed loading are both of great importance. For the latter the resistance to crack propagation is a more important aspect than that to crack initiation.

A number of the problems discussed in this paper cannot be satisfactorily solved without the use of advanced calculation methods like finite-difference and finite-element methods. Shock-induced phenomena belong to these. Existing theories are only applicable to uniform, prismatical bars loaded axially and transversally. Use of a finite-difference method for impulsive loading of a ship as a whole has been made by St. Denis and Fersht.[14] A second problem is the elasto-plastic behaviour of the ship's hull in case of small permissible plastic deformations (limited plastic design). A third field is that of fracture mechanics. The actual knowledge cannot easily be applied to complicated structures particularly not when appreciable plastic deformations occur at crack-tips. As an illustration fig. 15a shows the model of Argyris[13] for a cracked plate and the solution for the plastic zones as a function of load (fig. 15b).

IV. CONCLUSIONS

1 The highest wave bending moments ever recorded in ships at sea are smaller than the moments which on the basis of observed extreme waves and model tests could be expected.

2 There is no argument for designing the structure of ships on the basis of theoretically possible, but in practice apparently always avoided extremes.

3 Research on how ships can most effectively avoid extreme moments and movements in extreme sea conditions is of prime importance.

4 Plastic design theory gives an overoptimistic picture of the collapse strength of a ship, especially from the viewpoint of deformation energy.

5 Notwithstanding this, the margin between demand and capability is even for inferiorly designed ships so large, that the risk of collapse due to insufficient compressive strength of the longitudinal material is extremely small. Only excessive deterioration of the structure can lead to failure.

6 With regard to collapse by brittle fracture the margin between demand and capability is also wide. However secondary considerations have led to the conviction that the longitudinal strength of ships is still largely determined by the risk of brittle fracture.

7 The full benefit of the use of higher strength steels can only be obtained by careful design of structural details. Besides shipbuilders will have to accept the philosophy of "fail-safe" design which means that cracks are allowed up to a critical length.

8 Permissible stresses are dependent on crack length. In practice this will be reversed in such a way that permissible crack lengths are prescribed as a function of chosen stresses.

9 Thorough information about the cyclic loads acting on ships is indispensable for estimating the moment that cracks start as well as for estimating the speed of crack growth.

10 Finite difference and finite element methods will have to be used for the solution of many of the current and future problems in ship structural design like: shock phenomena, elasto-plastic deformations at underdeveloped plastic hinges, critical crack lengths in complicated structures.

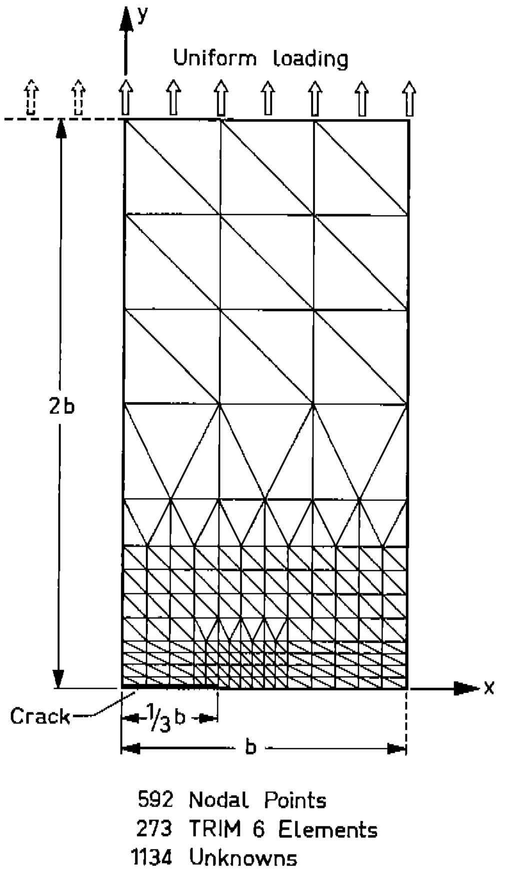


Fig. 15a. A Plate with a Semi-Crack in the Middle, Loaded Uniformly at the Ends. Only One-Quarter of the Idealized Plate is Shown (Using Trim 6 Elements). (Argyris, Patton[13])

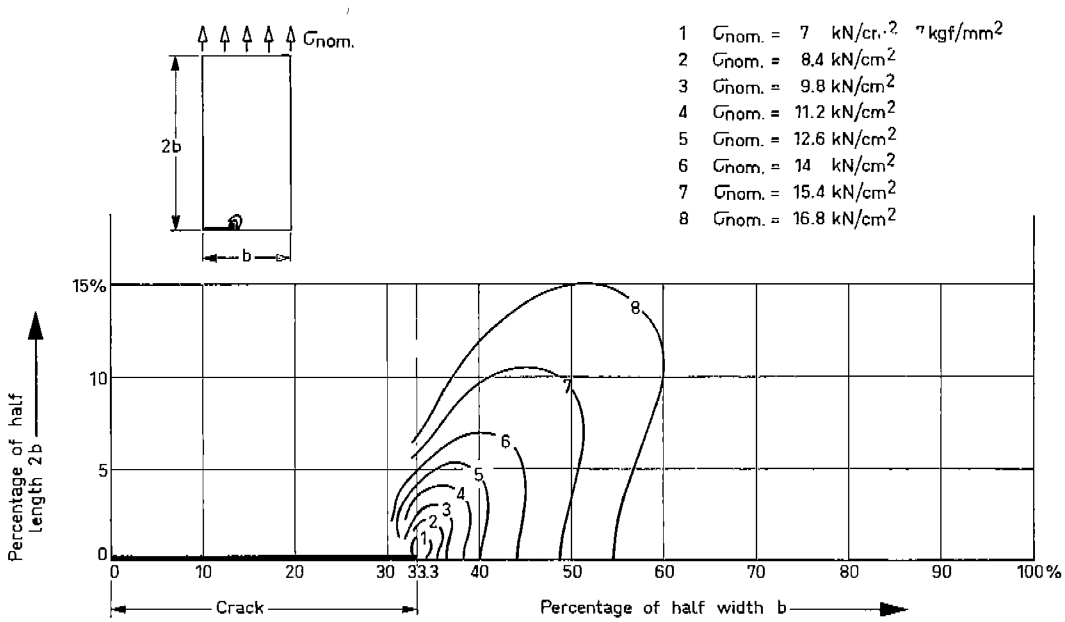


Fig. 15b. Development of the Area of Plastic Deformation in the Cracked Plate with Increasing Applied Load. (Argyris, Patton[13])

References.

1 Nordenstrom, N.
Statistics and wave loads.
Gothenburg, May 1964.

2 Lewis, E.V.
Predicting long-term distribution of wave-induced bending moment on ships hulls.
Trans. S.N.A.M.E. 1968.

3 Getz, J.R.
Longitudinal strength and minimum weight.
Scandanavian Ship Technical Conference, Oslo, Oct. 1960.

4 Proc. of 1st I.S.S.C. 1961. Glasgow.
Report of committee "Response to wave loads".

5 Bennet, R.
Stress and motion measurements on ships at sea.
Rep. no. 15, Swedish Shipbuilding Research Foundation,
Göteborg 1959.

6 Johnson, A.J.; E. Larkin.
Stresses in ships in service.
T.R.I.N.A. 1964.

7 Nibbering, J.J.W.
An experimental investigation in the field of low cycle fatigue and brittle fracture of ship structural components.
T.R.I.N.A., vol. 109, Jan. 1967.

8 Dalzell, J.
An investigation of midship bending moments in extreme irregular waves.
S.S.C. Rep. 155, Jan. 1964.

9 Caldwell, J.B.
Ultimate longitudinal strength
T.R.I.N.A., vol. 107, Jan. 1965.

10 Nibbering, J.J.W.
Considerations on the use of higher strength steel in ships. (In Dutch).
De Ingenieur, no. 26, 1968.

11 Yuille, J.M.
Longitudinal strength of ships.
T.R.I.N.A., vol. 105, 1963

12 Nibbering, J.J.W.
Fatigue of ship structures.
Neth. Ship Res. Centre TNO, Rep. 55 S;
I.S.P., Oct. 1963.

13 Argyris, J.H.: P.C. Patton.
Computer oriented research in a university milieu.
Appl. Mech. Rev., Dec. 1966.

14 St. Denis, M.; S.N. Fersht.
The effect of ship stiffness upon the structural response of a cargo ship to an impulsive load.
S.S.C. Rep. 186, Sept. 1968

15 Nibbering, J.J.W.
Fracture mechanics and applications for the structural engineer. (In Dutch).
Constructiematerialen, Nov.-Dec. 1968.

16 Maniar, N.M.; E. Numata.
Bending moment distribution in a mariner cargo model in regular and irregular waves of extreme steepness.
S.S.C. Rep. 190, Nov. 1968

17 Proceedings of 3rd I.S.S.C. 1967. Oslo.
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UNCLASSIFIED

Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) Ship Structure Committee	2a. REPORT SECURITY CLASSIFICATION Unclassified 2b. GROUP

3. REPORT TITLE

PERMISSIBLE STRESSES AND THEIR LIMITATIONS

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)
Special Report

5. AUTHOR(S) (First name, middle initial, last name)

Nibbering, Ir. J. J. W.

6. REPORT DATE 1970	7a. TOTAL NO. OF PAGES 15	7b. NO. OF REFS 17

8a. CONTRACT OR GRANT NO. None b. PROJECT NO. c. d.	9a. ORIGINATOR'S REPORT NUMBER(S) SSC-206 9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)

10. DISTRIBUTION STATEMENT

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY None

13. ABSTRACT

Various aspects of capability and demand of ships such as extreme loads, cyclic loads, plastic design, crack design, collapse and damage are discussed in an attempt to make a synthesis. It is explained that whenever permissible stresses are used in structural design, they should be bounded by probability-concepts, deformation criteria and critical crack lengths. For an acceptable risk factor, the margin between capability and demand seems to be substantial in ships. Yet drastic reductions in structural weight will only be possible if the principle of "fail safe" design is adopted in shipbuilding to the same extent as in aeroplane-building.

DD FORM 1 NOV 65 1473 (PAGE 1)

S/N 0101-807-6801

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14 KEY WORDS	LINK A		LINK B		LINK C	
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