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Speed and Strength of Large Tankers.

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Speed.

From a classification man it will usually be expected that he says something about strength of ships. Before I do this to-day I like, however, to say a few words about speed, from a point of view of general interest.

Table 1 is a compilation of main data for some typical tankers of all sizes divided into five groups according to their Froude numbers, which are given in the last column of the table.

The very first ship of Table 1 is one of the three small wooden sailing ships which in 1877 were converted at a small Norwegian yard to tankers for carrying oil in bulk. Five years later the same owner converted two other similar ships in the same way. These five sailing ships were probably the first tankers in the world to carry oil in bulk.

It will be seen from the first group that the 14,500 tons motortanker built in 1928 had the same Froude number as the sailing ship of 1874. Similarly the second group shows that the 3200 tons iron steam tanker "Glückauf" of 1886 had a speed corresponding to the speed of the 85,500 tons "Universe Leader" of 1956. The third group shows that 12 knots for the 12,600 tons "Mosli" of 1935 corresponds to 16½ knots for the 104,500 tons tankers to be built at the Kure Shipyard in Japan. And as seen from the fourth group 13½ knots for a 15,000 tons ship corresponds to 16 knots for a ship of 35,000 tons deadweight and 19 knots for the 106,500 tons tankers ordered from the Bethlehem yard at Quincy. The "Vesta" of nearly 20,000 tons deadweight in the fifth group is an exceptionally fast tanker judged from her Froude number.

In Table 2 a few modern dry cargo ships of different sizes have been put together for comparison with the tankers. Their Froude numbers vary from 0.22 to 0.27, while the Froude numbers of the tankers vary from 0.14 to 0.21 with the majority of modern tankers at about 0.18.

Table 1.

TANKERS

Ship	Built	Main dimensions Lbp x B x D	DW in tons	Draft	Propulsive machinery	BHP	Speed in knots	$\frac{BHP}{DW}$	$v/\sqrt{gL}$
LINDESNÆS	1874	152'6" x 29'6" x 18'10"	ab.670	17'4"	Sail		6		0.146
HAAKON HAUAN	1935	407'0" x 56'0" x 33'6"	9870	27'2"	Diesel	2600	10	0.263	0.147
FINNANGER	1928	475'0" x 64'6" x 36'6"	14530	28'9"	Diesel	2800	10½	0.193	0.143
GLÜCKAUF	1886	300'0" x 37'3" x 23'2"	ab.3200	17'0"	Steam recipr.	850	9	0.265	0.155
ORKANGER	1928	458'0" x 59'10" x 34'9"	11610	27'8½"	Diesel	3160	11	0.272	0.153
UNIVERSE LEADER	1956	815'0" x 125'0" x 61'3"	85515	46'1½"	Turbines	19500	15	0.228	0.157
MOSLI	1935	465'0" x 60'9" x 33'11"	12600	26'9"	Diesel	3450	12	0.274	0.165
(Project)		900'0" x 135'0" x 67'6"	104500		Turbines	27500	16½	0.263	0.163
EIDANGER	1938	480'0" x 68'4" x 36'0"	14800	28'2"	Diesel	4100	13½	0.277	0.183
WORLD UNITY	1952	625'0" x 86'0" x 45'9"	31745	34'6½"	Turbines	13700	15½	0.431	0.185
FERNCREST	1955	645'0" x 87'0" x 47'6"	34800	35'5"	Diesel	12500	16	0.359	0.187
ESMERALDA	1957	685'9" x 97'5" x 49'9"	40800	35'11"	Turbines	17000	16½	0.417	0.187
EVGENIA NIARCHOS	1956	725'0" x 97'2" x 52'0"	47150	37'10"	Turbines	20500	17	0.435	0.188
(Project)		920'0" x 134'0" x 66'6"	106500	49'0"	Turbines	43000	19	0.404	0.187
VESTA	1957	530'0" x 71'9" x 40'3"	19875	31'0½"	Diesel	10000	16¼	0.50	0.210

18.2.58
D/AN

Table 2.

D R Y   C A R G O   S H I P S

Ship	Built	Main dimensions $L_{bp} \times B \times D$	DW in tons	Draft	Propulsive machinery	BHP	Speed in knots	$\frac{BHP}{DW}$	$v/\sqrt{gL}$
VELARDE	1957	300'2" x 45'3" x 27'5"	2670	18'0"	Diesel	3800	15¼	1.42	0.26
TOPEKA	1949	400'0" x 55'0" x 38'0"	7550	25'6"	Diesel	5200	15	0.69	0.22
RISANGER	1951	450'0" x 61'0" x 39'2"	9955	28'7"	Diesel	8000	17	0.80	0.24
BRAZILIAN REEFER	1953	375'0" x 54'0" x 33'9"	3900	22'11"	Diesel	6720	17½	1.72	0.27
OKLAHOMA	1949	440'0" x 59'0" x 37'6"	6900	25'1"	Diesel	11000	19	1.60	0.27
SEATTLE	1947	465'0" x 64'0" x 39'10"	9085	26'8"	Diesel	14000	19½	1.54	0.27

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D/AN

The brake horsepower per ton deadweight is given in the second last column of the two tables. This figure is probably easier to visualize than the Froude number. For the tankers it varies from 0.2 to 0.5 with the majority not above 0.4, while for the dry cargo ships it varies from 0.7 to 1.7 with about 1.2 as a fair average.

What this means can be seen from the diagram Fig. 1, where BHP per ton deadweight is used as ordinate and speed as abscissa. Each curve represents a constant deadweight, i.e. increasing ship dimensions and decreasing block coefficients with increasing speed.

With BHP/DW = 0.4 a 20,000 tons tanker would have a speed of 15¼ knots, while with BHP/DW = 1.2 a 20,000 tons dry cargo ship would have a speed of 20 knots and a 20,000 tons tanker 20½ knots. With BHP/DW = 0.4 a 100,000 tons tanker would have a speed of 18 knots, while with BHP/DW = 1.2 a tanker of the same deadweight capacity would run at nearly 25 knots.

Remembering that tankers are at sea some 300 - 320 days a year compared with 200 - 250 days for dry cargo ships, it seems surprising that the tankers are run at such a low speed. It is, however, not surprising that private owners who time-charter their tankers to oil companies keep the speed as low as the charterers will accept, as long as they are not sufficiently paid for extra speed with the accompanying increase in ship simensions and capital cost. But that the oil companies keep the speed of their tankers as low as they do n must be due to their way of calculating the most economic speed.

If by most economic speed one means the speed by which the ton/mile of oil cargo is carried at the least possible expenses, one gets a low speed, which should practically correspond to the speed at which a private owner would try to run his ship when the freights are so low that they may just cover his expenses. But if in addition to expenses one also calculates with an income due to an average freight rate and by most economic speed means the speed at which the ton/mile of oil is carried with the maximum return on the invested capital, the speed should usually be higher.

With the present practice tankers and dry cargo ships cannot both be run at the most economic speed. Something must be wrong somewhere. It is not suggested that tankers should immediately be speeded up to the same level as dry cargo ships. But the difference in speed between the two types is so large that there might be a possibility of choosing something in between.

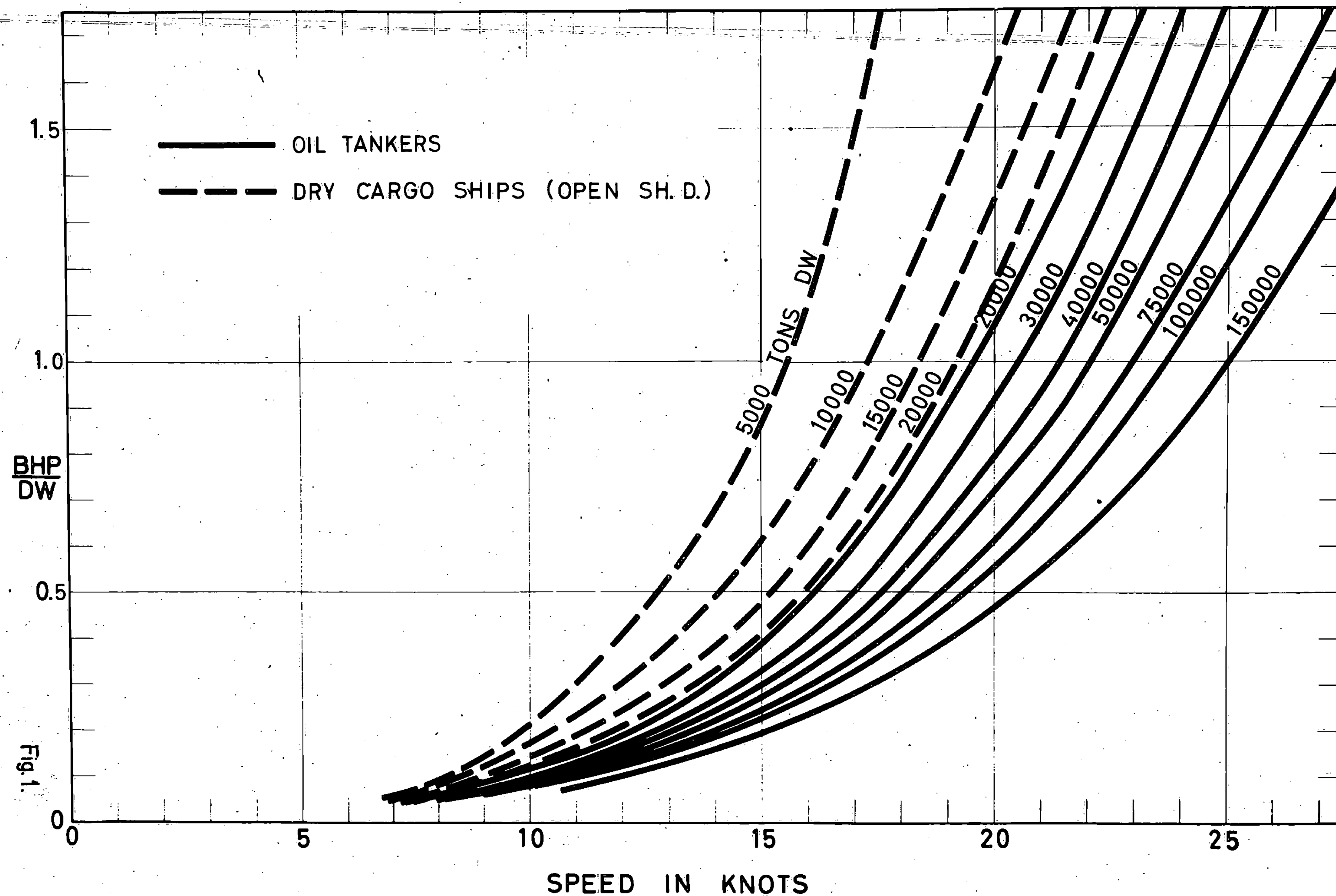

OIL TANKERS
DRY CARGO SHIPS (OPEN SH. D.)
5000 TONS DW
10000
15000
20000
20000
30000
40000
50000
75000
100000
150000
BHP/DW
1.5
1.0
0.5
0
0
5
10
15
20
25
SPEED IN KNOTS

Fig. 1.

Strength.

Why do tankers break?

During the first 40 years of this century I estimate that as an average about one ship per year has broken in open sea, taking the total for the whole world. During World War II the number increased heavily. For a while broken ships became an epidemic among the ships welded together in U.S.A. during the war. After the war the average can be estimated to about two broken ships per year. Ships built in America during the war are still among them. A good many of the broken ships have been tankers.

A total of seven ships belonging to the Norwegian merchant fleet have broken in open sea, the first one in 1939, the last one in january 1958. They have all been tankers, ranging in size from 11000 to 15000 tons dw. all riveted or at least partly riveted, and all built with transverse framing. They all broke in very rough sea and in fully loaded condition, which means that the maximum bending moment was in sagging with compression in deck and tension in bottom. The break was exactly amidships, through center cargo tank No. 5, the tank aft of the pump room.

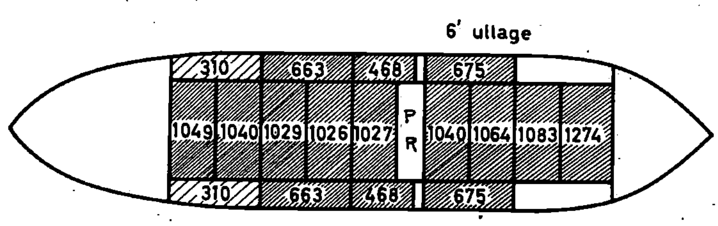
Calculation has shown that these tankers were not safe against buckling of deck plating between two adjacent beams at the compressive stresses to which the deck has probably been exposed. The failure has probably started with a fairly sudden collapse of the deck plating in buckling between two beams, resulting in such a weakened section that the bottom plating has been torn off in tension, and half a wave length later, when the ship was exposed to hogging the deck plating was torn off in tension.

Especially from American side the objection has been raised against this theory that buckling of the deck will only reduce the section modulus a very small amount, not sufficiently to explain why the bottom plating should be torn off so suddenly. Personally I believe that under circumstances when buckling occurs right across the deck the buckling stress may suddenly be reduced to say less than a third of the theoretical critical stress with the result that there will be practically no resistance left in the deck, and this should reduce the section modulus sufficiently to explain the failure. For columns it has been proved theoretically as well as experimentally that under certain circumstances they can snap off at a much lower load than expected. No doubt the same can happen also with plates.

The failures mentioned have shown clearly that it is wrong to build tankers larger then say 400 feet in length with 'thwartship stiffening in deck and bottom. Since about 1950 it has also never been done, as far as I know.

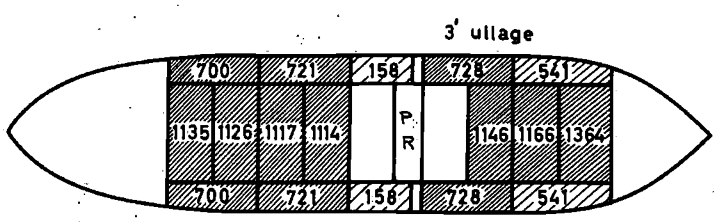
The tanker which broke in January this year was the first and up till now the only ship classed with the Norwegian Veritas which has broken in the open sea. She was 20 years old and riveted. The upper part of fig. 2 shows how she was actually loaded when she broke, while the lower part shows how she might have been loaded if one had taken care to reduce the bending moment as much as possible. The trim of the ship is the same in both cases. The actual distribution of cargo resulted in a still water bending moment of 33,600 ton-meters, while the proposed distribution gives a still water bending moment of only 4000 t.m. In both cases must be added one and the same wave bending moment which for very rough sea has been calculated to about 57,000 t.m. The total bending moment in heavy storm has still been about 50 % larger than necessary with a careful distribution of cargo.

1. VED HAVARI.



STILLEVANNSMOMENT = 33600 TONNM.

2. FORSLAG.



STILLEVANNSMOMENT = 4000 TONNM.

Fig. 2

This shows that it is not easy to teach all sailors how to distribute the cargo. Many of them still seem to believe that it is wrong to have an empty cargo tank amidships and prefer to keep the empty tanks at the ends. For large tankers it is therefore essential to have a loadicator or other instrument on board to show how the cargo should be distributed to give a reasonable bending moment.

Most of the ships which have broken during and after the war belong to an entirely different category than the Norwegian broken ships. The tankers have been longitudinally stiffened and have had sufficient strength judged from ordinary bending calculations. Most of them have been welded. They have never broken in warm weather, but always at temperatures near zero or a little below zero centigrade.

One explains this by a temperature dependent property of the steel material. This property will not be detected by usual tensile and bending test, but will show up if the test pieces are given a notch or other means of stress concentration. The property also depends upon the chemical analysis and the method of production of the steel. The trouble with the American war-built ships was to a great extent due to their steel containing too much carbon and too little manganese. To-day one requires above certain thicknesses that the steel must be killed by silicium and a little aluminium, and where the finishing temperature in rolling is not kept at about 930°C the steel should also be normalized to keep the grain size down and get an even product. Usually only Siemens Martin steel and electro-steel are accepted for ships, but some of the new methods of producing steel in converters by blowing with oxygen may be accepted after very thorough tests. It is necessary to control the content of nitrogen, slag inclusion and grain size, take Baumann prints, watch the ageing in notch tests etc. Ordinary Thomas steel has never been accepted for ships and is still more dangerous now when the ships are being welded.

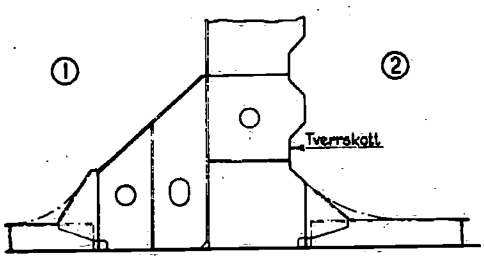
The phenomenon of brittle fracture is also closely connected with the design of structural details and the workmanship. Sharp corners and other reasons for stress concentrations, straight brackets, doublers, holes of shapes most common in shipbuilding, too abrupt changes in the dimensions of girders etc. must be avoided. If it is unavoidable to make a hole in the deck or the shell of a tanker it is wise to consider how such a hole would have been designed if it were not a ship but a pressure vessel, and remember that

that the stresses in deck and bottom of a tanker may be considerably higher than they are in a pressure vessel. The designers of ships can do much to avoid stress concentrations and socalled hard spots where cracks easily develop. But even with a very thorough inspection and many X-ray pictures and using only certified welders, it is difficult to entirely avoid spots of bad workmanship. In any case it is unrealistic to rely upon the workmanship being 100 % perfect in every spot and corner of a complicated ship. But every unperfect spot of a weld is equivalent to a stress raising notch which will always be a latent danger for the initiation of brittle cracks. Therefore one has to specify notch tough steel and increase cost by insisting upon soft designs of details.

When discussing the value of one or more riveted seams as crack arrestors one should remember that one does not make a welded ship riveted by fitting a few riveted seams. The structure is still rather monolitic. Carefully made riveted seams have, however, been able to stop small cracks, but cannot be relied upon stopping big cracks which have already developed considerable energy and run very fast. And if seams are riveted the plate edges must not be sheared and the rivet holes not punched, because such cold working is ageing the material and many cases are known of cracks having started at such strain aged edges or holes. This oldfashioned workmanship consists a danger, making riveted seams crack initiators instead of crack arrestors.

Six years ago two American T2-tankers broke in the same storm east of Cape Cod, which lies on the coast between New York and Boston. A Norwegian T2-tanker classed with the Norwegian Veritas but exactly like the Americans, was in the vicinity and had to stand the same storm. When she came into port it was found that all those bottom longitudinals marked with a small circle with numbers 1 or 2 inserted in fig. 3 had broken where marked. All these longitudinals were in line with the vertical web girders on the 'thwartship bulkheads. Practically every longitudinal of this type in cargo tanks 3-8 had broken. None of the other bottom longitudinals were broken, although they were all of the same dimensions as the broken ones.

The longitudinals which broke always did so close to a bulkhead where they had been connected to the vertical webs by large brackets, as shown to the left in fig. 3. We attributed this mainly to the large difference in stiffness between the vertical webs and the longitudinals. After this lesson we strengthened all



SNITT VED VERT. BÆRER 10'-0" & 25'-0" FRA ℄.

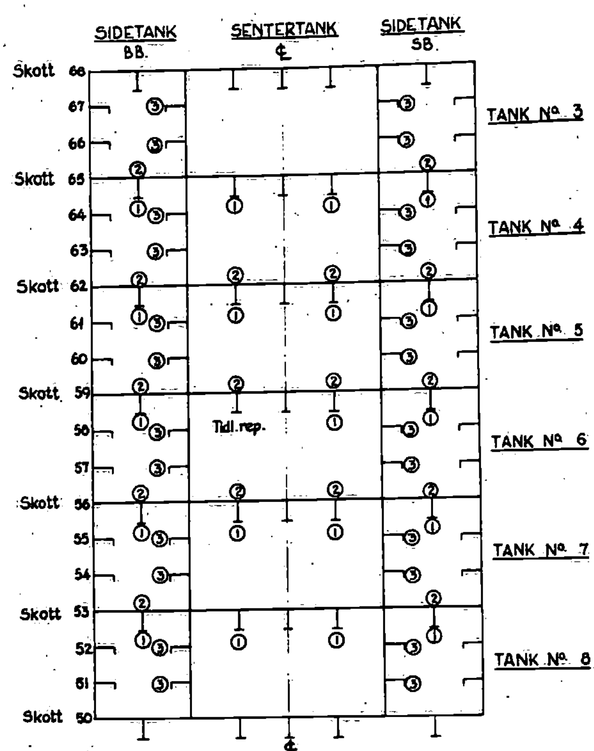
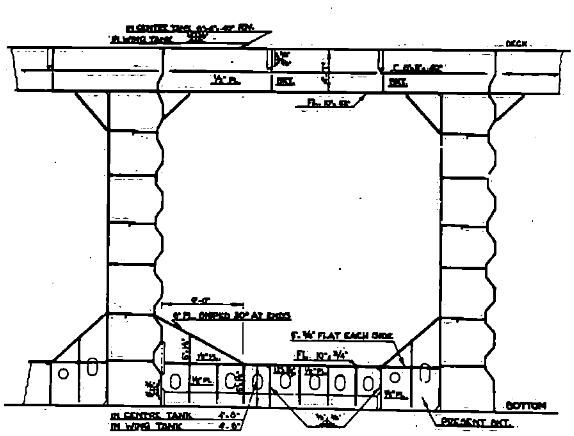
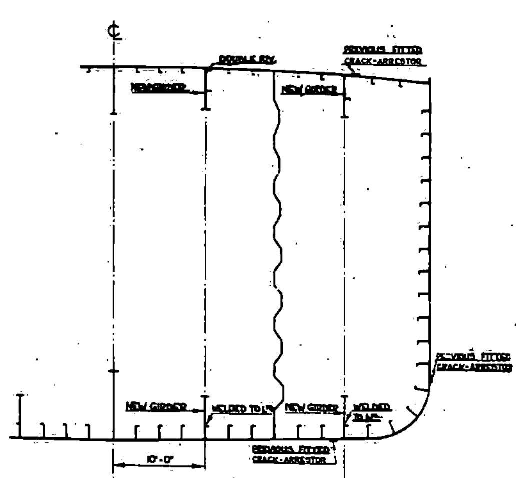


Fig.3



NEW DECK & BOTTOM GIRDERS 10'-0" & 25'-0" FROM ℄.



SECTION

Fig.4

our T2-tankers by adding a web to these longitudinals as shown in fig. 4, thereby increasing their stiffness. Nothing else was done. And since then nothing has happened with our T2's, except the usual leakages in the corners between the longitudinal and 'thwartship bulkheads, an unremediable disease common to all T2-tankers. Other classification societies strengthened their T2's by adding 15 % to the total section modulus, which means about 20 % of all longitudinal steel, a much more expensive job, and it does not seem to be safer, because the original section modulus was not less than for other tankers.

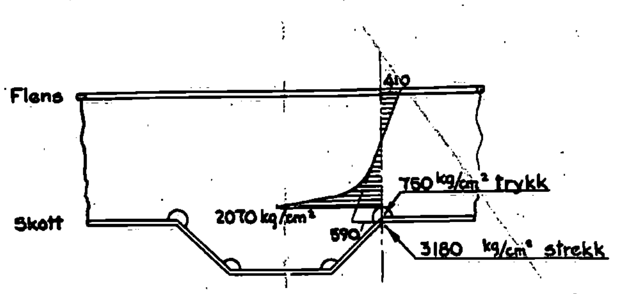


Fig.5

From figs. 3 and 4 can also be seen that the girders on the corrugated bulkheads in the T2-tankers were fitted on one side of the bulkhead only. This is a design which has given much trouble and many people do not like corrugated bulkheads because of this trouble. As to one-sided girders fig. 5 gives en example of stress measurements made by our Norwegian Institute of Ship Research. It shows that at the inner bulkhead corner there is a stress concentration factor of about 4. When calculating the girder strength nothing of the bulkhead plating can be included as flange. The correct way of designing a girder for a corrugated bulkhead is to make it symmetric with the bulkhead near the neutral

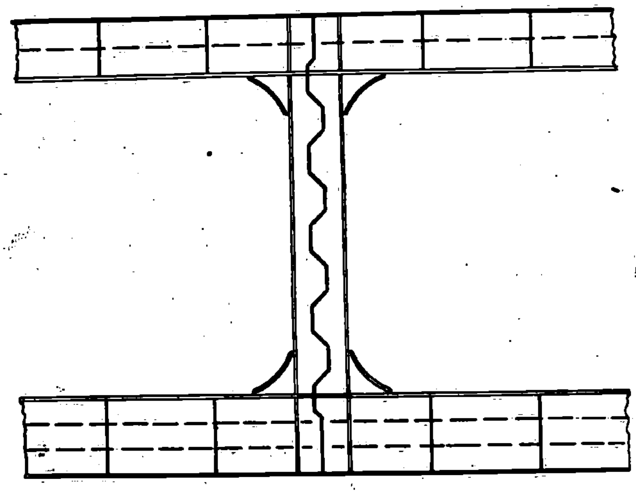


Fig.6

Axis where the stress is low, as shown in fig. 6.

Another point in connection with corrugated bulkheads is that not only the section modulus, but also the moment of inertia of the corrugations should be controlled. This is advisable to keep not only the stresses, but also the deflection at a reasonable limit. When in addition details in the connection between corrugated bulkheads and other bulkheads are watched for continuity, there should be no fear for the corrugated bulkheads.

Fig. 6 shows also the longitudinal centreline girders at bottom and deck. Longitudinal stiffening of these girders has been suggested because such stiffening will give better safety against buckling, with less weight, than will the orthodox vertical stiffening.

Fig. 7 is a picture which has been shown very often of the T2-tanker "Schenactady" which broke in two during the war when lying at the builders' quai before delivery. The picture is usually taken as an example of the possibility of brittle failure even with no bending stress. But it should be noted that the two halves of the ship lie at a large trim angle against each other, which is a proof that the ship must have been exposed to a large bending moment when the failure occurred. Right aft and forward the ship is touching the bottom of the harbour, otherwise the trim angles would have been still larger. Officially the max. stress due to the still water bending moment caused by careless distribution of ballast

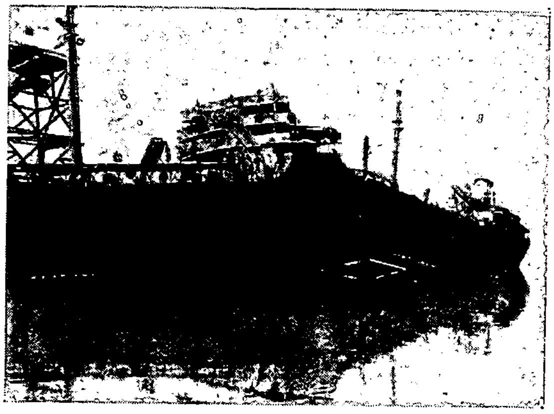


Fig.7

has been given as 760 kg/cm^2. Judging from the large trim angles of the two halves it may have been larger. To this stress must be added temperature stresses caused by reduction of the air temperature during the winter afternoon, while the water temperature remained the same until the ship suddenly broke late in the evening.

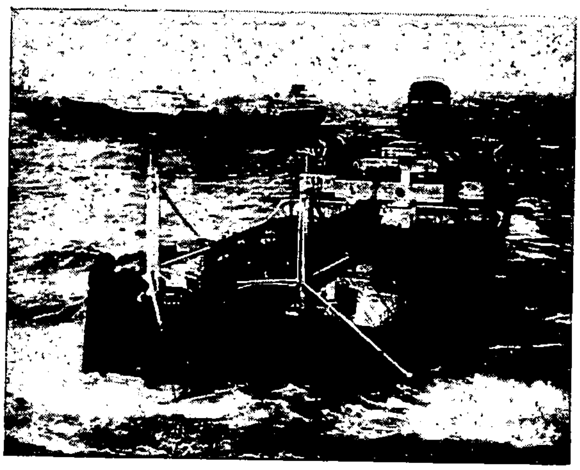


Fig. 8

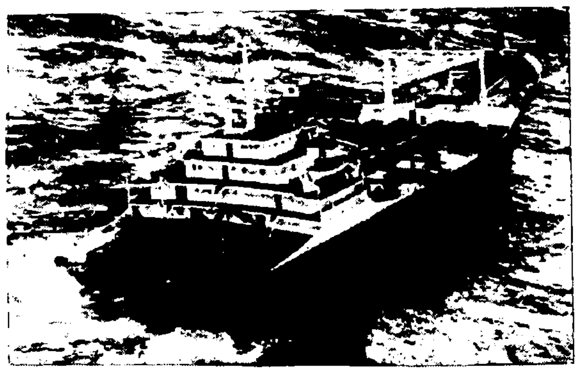


Fig. 9

Figs. 8 and 9 show the two halves of the 32,000 tonner "World Concord" which broke in two in the Irish Sea in November 1954, only one year old. The two halves both lie on even keel which means that there has been practically no still water bending moment when she broke. But the wave bending moment was probably high. The fracture was brittle throughout. The steel material had very little notch toughness and was very uneven with coarse grain owing to the rolling having been finished with a much too high temperature.

She broke close to a 'thwartship bulkhead amidships. Only 50 mm from the bulkhead was a welded butt running right across the ship. The butt can be seen in fig. 10. It is undoubtable a mistake to have two weldings so close together, such a design accumulates too much energy in residuary stresses.

Fig. 11 shows that no trace was left of the two longitudinal bulkheads at their connection to the 'thwartship bulkhead. The longitudinal bulkheads had been made intercostal and the case shows that fillet welding is not a good connection. Longitudinal bulkheads should be made continuous, at least the upper and lower strakes of them, because the highest stresses are in the longitudinal direction of a ship.

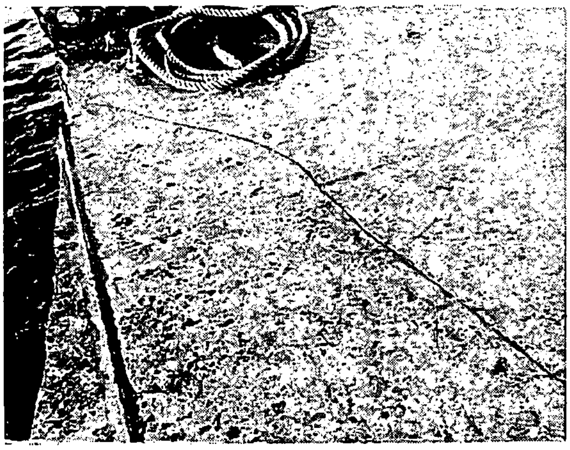


Fig.10

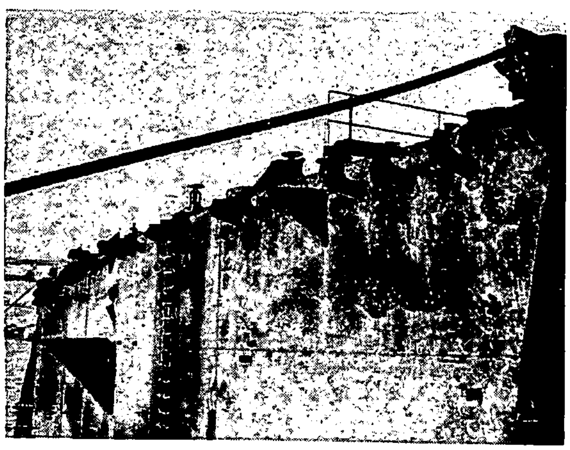


Fig.11

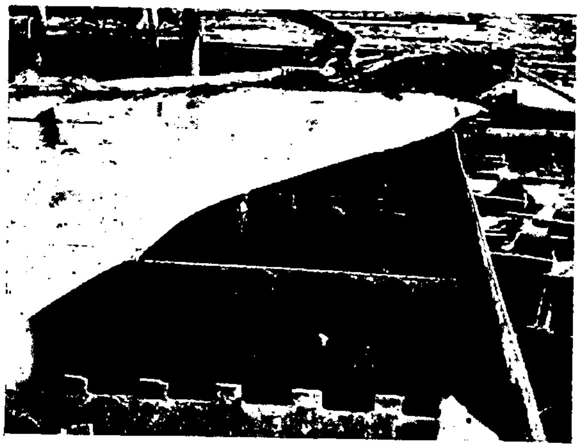


Fig.12

Fig. 12 shows how the main crack along the bulkhead has stopped for a moment. The crack has then jumped several meters and started again at a hard spot at the deck girder where it could be seen from the herringbone pattern that it had run in both directions. From this picture it can also be seen that the fillet welding between the scalloped longitudinals and the deck plating has not been strong, none of the small lips between the scallops having broken.

From fig. 11 it can also be seen that all the longitudinals had broken some distance from the bulkhead, at the end of the brackets. The details can be seen better in fig. 13, which is a short distance picture of two of the longitudinals. They have broken at the end of the brackets because this end represents a hard spot, especially because the brackets have been overlap welded. The lesson is that one should try to avoid overlapping of this nature.

Fig.13

Norwegian Veritas' Rules.

In the Norwegian Veritas we have for some years spent a great deal of thought and money on the question of building rules for very large tankers. More than a year ago we sent out a preliminary draft as a recommendation how to build large tankers, and nearly half a year ago our Committee adopted our Special Rules for the Construction of Oil Tankers of Length greater than 200 meters. A complete edition of our revised Rules is now being printed and will be available for sale in about 3 months. I shall not go into details here, but believe it may be of interest to mention some main points in connection with the development of the Rules.

In most longitudinal strength calculations it has been customary to use a wave height of 1/20 of the wave length. This has been shown by the lower dotted straight line in fig. 14, in which several observed wave heights have been marked. It will be observed that all the observed points given in the diagram lie above the straight line H = L/20 for wave lengths less than 250 meters. The upper curve covering the largest observed data has been suggested by the British Admiralty Ship Welding Committee. In the Norwegian Veritas we decided to use a somewhat lower curve given by the equation $H = 0.45\ L^{0.6}$ meters, which gives larger wave heights than L/20 for wave lengths less than 245 meters, but less heights above this length. It follows approximately the same law as the curve of the A.S.W.C. but has an amplitude of only 80 % of the latter. The largest waves are so rarely met with that we think this reduction to be justified. It should also be remembered that the value of wave height amplitude chosen for static calculations has only a relative importance, it must be seen in conjunction with the nominal stress allowed.

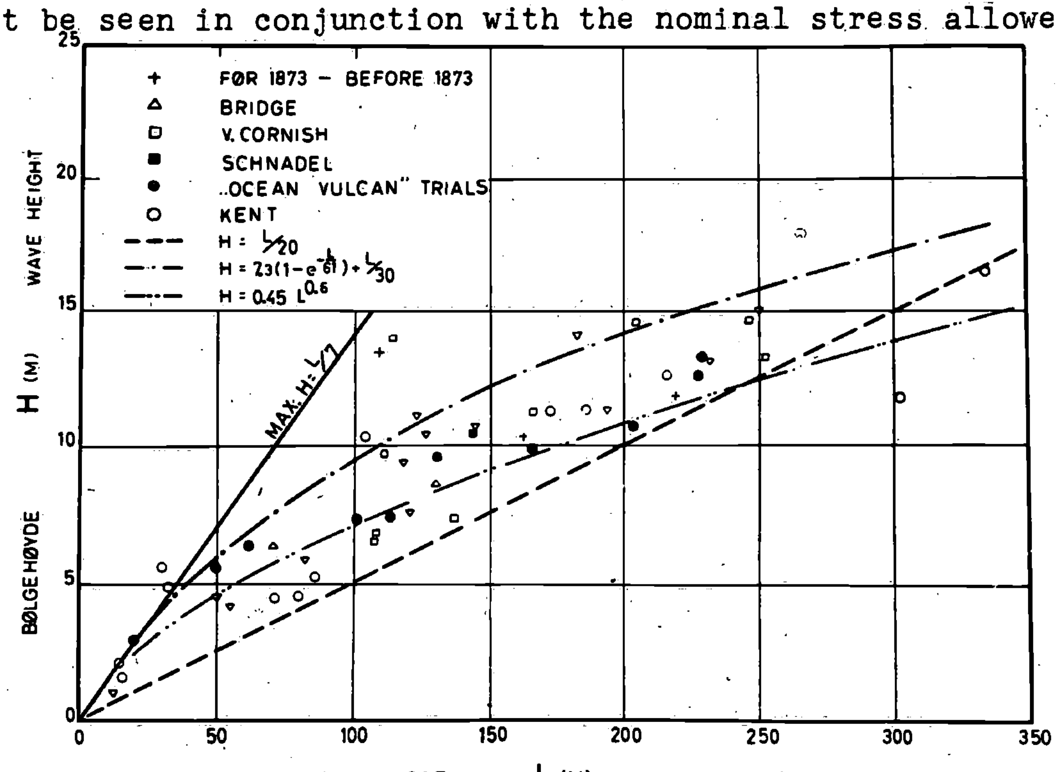


Fig.14

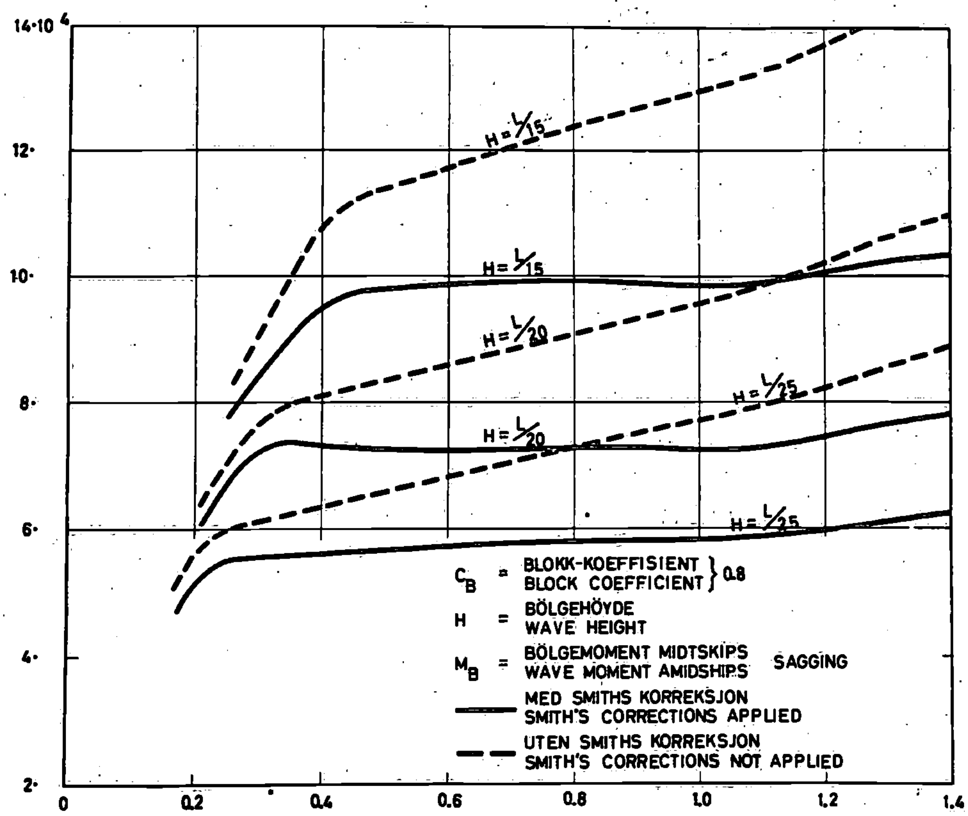


Fig. 15

For simplicity the calculation of wave bending moments are very often carried out without Smith's correction, which takes into consideration the difference in hydrostatic pressure in wave crests and troughs. Doing so one refers to the relativity of the calculation, stating that if the same procedure is used throughout this should give a correct result. That this is not so is seen from fig. 15, where the curves give the dependency of the wave bending moment on the draft of the ship. This and similar results for other block coefficients show that without Smith's correction the wave bending moment increases with increasing draft, while with Smith's correction it is independent of the draft. It is therefore necessary to include Smith's correction.

In addition to the wave height it is also necessary to know which length one should choose to get the largest wave bending moment. The result of calculations with different wave lengths have been given in fig. 16, where they have been made dimensionless by comparison with the bending moment at a wave length equal to the ship length. It is seen that if the wave height is proportional to the length, say H = λ/20, the maximum bending moment occurs at a wave length somewhat larger than the ship length.

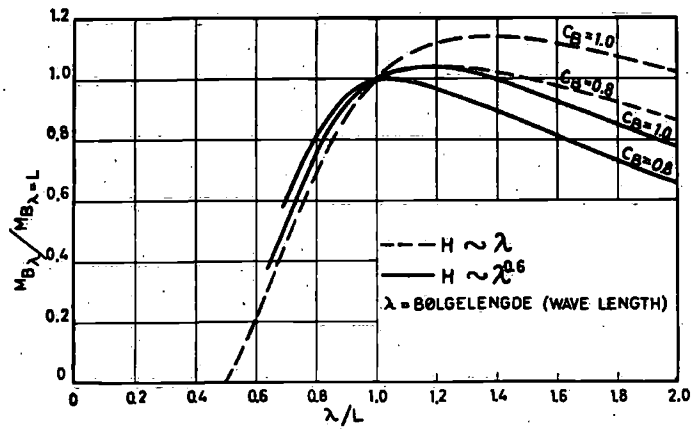


Fig.16

But if the wave height is made proportional to $\lambda^{0.6}$ the maximum bending moment occurs at a wave length equal to the ship length.

On the basis of our calculations for ships with different fullnesses we have found that we can write the wave bending moment in sagging

$$M_{Bsag} = \frac{C_B+0.8}{1.6} L^2 BH\gamma \; m_{Bsag}$$

where γ = the specific gravity of sea water,

m_{Bsag} = 1.44/100 is a wave moment constant for sagging.

The Wave bending moment in hogging can be written

$$M_{Bhog} = C_B L^2 BH\gamma \;\; m_{Bhog},$$

where m_{Bhog} = 1.55/100.

The wave bending moment is defined as the total bending moment in waves minus the still water bending moment.

Up till now we have not sufficient data to show that it is necessary to take dynamic effects due to speed and other motions in a seaway into account for the midship strength of tankers. We have, however, started a systematic research programme in collaboration with the Norwegian ship model tank to try to find out if and how dynamic effects should be taken into account.

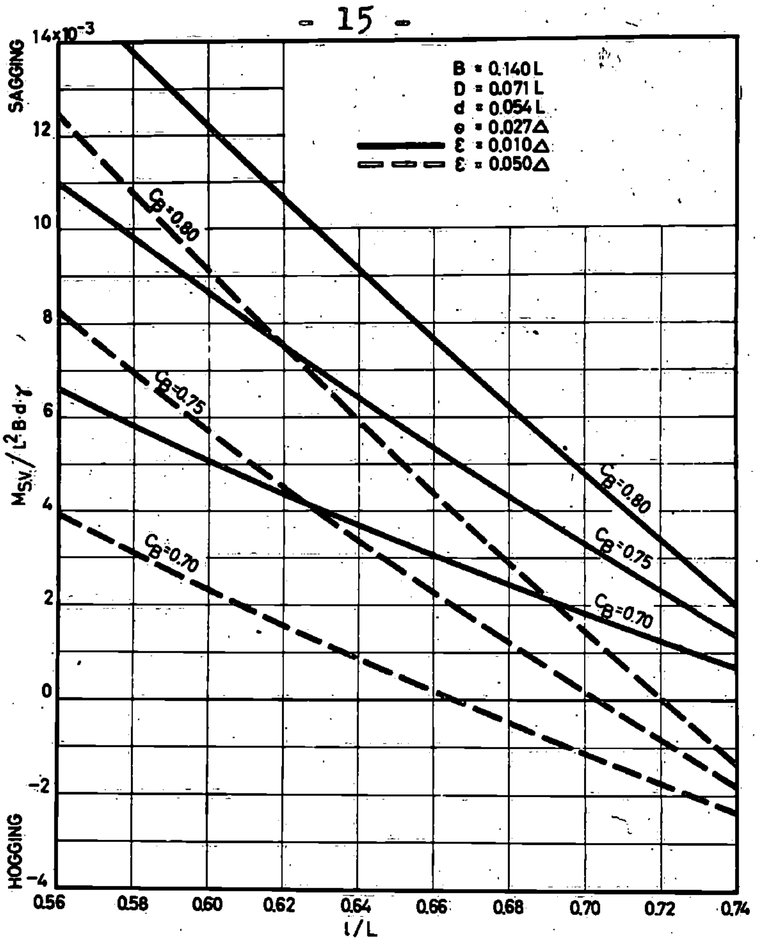


Fig.17

The still water bending moment is largely influenced by the distribution of cargo, ballast and other weights. We have standardized the calculations to give the influence of a nominal cargo tank length relative to the ship length, with block coefficient, engine room weight and bunker weight as parameters. Fig. 17 shows the result of such a calculation for a loaded ship with three different block coefficients, two different bunker weights and an engine room weight of 2.7 % of the displacement. For loaded ships in sagging we have standardized on a bunker weight of 1 % of the displacement placed aft, i.e. the upper fully drawn curves of fig. 17. If the curves are approximated by straight lines this still water bending moment can be expressed as

$$M_{sv} = 0.43\ L^2 Bd\gamma\ (C_B - 0.62)(0.76 - l/L).$$

With regard to nominal cargo tank length l the basis of this is the cargo tank length with a homogeneous distribution of cargo with the ship trimming on even keel. This is often less than the actual cargo tank length, because in many tankers the cargo tanks are carried so far forward that the ship will trim down with her nose with an even distribution of homogeneous cargo. But no captain likes to go to sea with a larger draft forward than aft. The only realistic scheme is therefore to shift the forward cofferdam so that the ship will be lying on even keel with the minimum bunkers on the voyage.

The dimensions and position of the pump room and eventual ballast tanks are taken care of as corrections to the cargo tank length. In large tankers it has certain advantages to have the pump room aft. This will, however, usually mean a larger bending moment compared with a case where the pump room is amidships. In such a case, to avoid the penalty of a larger steel weight, it is of advantage to arrange a ballast tank amidships not connected to the main cargo pipe line.

Adding wave bending moment and still water bending moment we get the total bending moment. But while the still water bending moment is always there the wave bending moment used in our calculations is very seldom obtained in practice. We therefore put more weight on the still water bending moment than on the wave bending moment. We do this by stipulating that when the still water bending moment is 60 % of the wave bending moment the total maximum stress shall not exceed 1500 kg/cm^2. This stress is, in other words, divided as 940 kg/cm^2 due to the wave bending moment and 560 kg/cm^2 due to the still water bending moment. For this case we therefore require a midship section modulus $W = M_B/940 = M_{SV}/560$. For other ratios between the two moments we put

$$W = \tfrac{1}{2}(M_B/940 + M_{SV}/560).$$

Using the approximate expressions obtained by our calculations we get

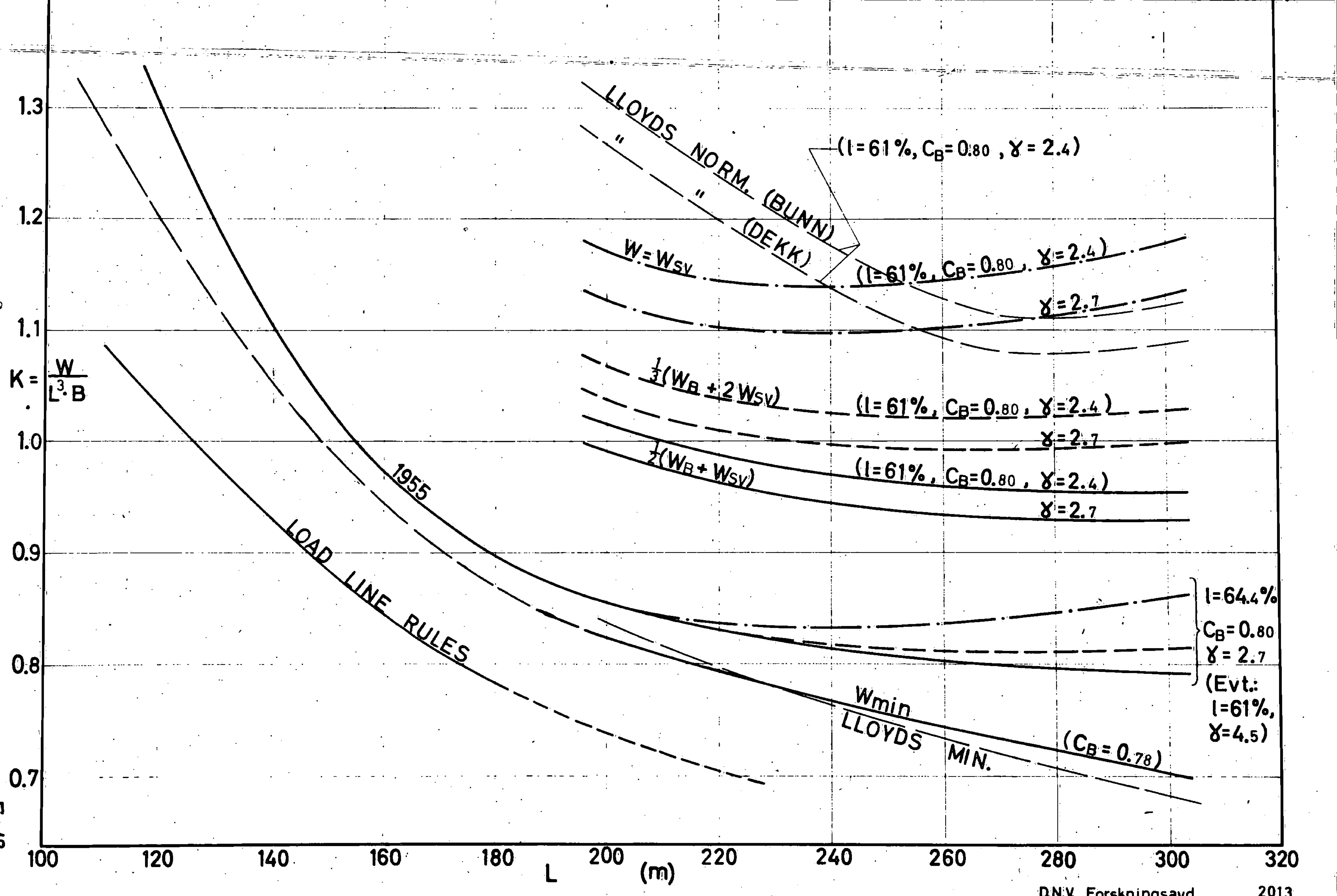
$$W = 0.22(C_B+0.8)L^{2.6}B + 38.5\ L^2Bd\ \left[(C_B-0.62)(0.76-\tfrac{l}{L})+\frac{2.7-\theta}{400}\right]\ \mathrm{cm}^3$$

where θ = engine room weight in % of the displacement and L, B, d and l are in meters.

The second part of the formula represents the influence of the still water bending moment, which is decreasing with increasing cargo tank length. But the calculation of the still water bending moment has been based on an even distribution of homogeneous cargo. The larger the tank length the larger will also be the cubic capacity and the more danger there will be that the mate will not distribute the cargo evenly, especially with very large tankers which carry crude oil with a specific gravity of not less than 0.8. We have therefore found it necessary to specify a minimum section modulus below which it is not allowed to go whichever may be the cargo tank length. Our minimum is given by the formula $W \geqq 0.44(C_B+0.8)L^{2.6}B\ \mathrm{cm}^3$ which is twice the first term of the previous equation.

In fig. 18 are given a curve for the section modulus required by the load line rules and some distance above a curve for our minimum. The two fully drawn curves somewhat higher again represent our requirements for a cargo length of 61 % of the ship length. The two uppermost curves are the requirements of Lloyd's Register for the same tank length according to a circular of last year. Lloyd's does not seem to require that the ship must be on even keel, wherefore their tank length may be larger than ours for the same ship and the requirements correspondingly reduced. In any case all shipyards will very soon learn to choose an arrangement which brings the section modulus down to minimum, and the minimum requirements are practically the same for the Norwegian Veritas and Lloyd's Register. For arrangements which do not bring the section modulus down to minimum Lloyd's Register seems to consider the still water bending moment only, while the Norwegian Veritas has considered it more correct to use a combination of the still water and wave bending moments. The influence of the cargo tank length is therefore greater with Lloyd's than with us.

In connection with design I like to mention one question which I think is rather important for supertankers. To obtain the necessary midship section modulus one must have a certain cross-sectional steel area in deck and a certain area in bottom. These



LLOYDS NORM. (BUNN)
" " (DEKK)
(l=61%, C_B=0.80, γ=2.4)
W=W_SV
(l=61%, C_B=0.80, γ=2.4)
γ=2.7
1/3(W_B + 2W_SV)
(l=61%, C_B=0.80, γ=2.4)
γ=2.7
1/2(W_B + W_SV)
(l=61%, C_B=0.80, γ=2.4)
γ=2.7
1955
LOAD LINE RULES
l=64.4%
C_B=0.80
γ=2.7
(Evt.:
l=61%,
γ=4.5)
Wmin
LLOYDS MIN.
(C_B=0.78)
K = W / (L^3·B)
1.3
1.2
1.1
1.0
0.9
0.8
0.7
100
120
140
160
180
200
220
240
260
280
300
320
L (m)
D.N.V. Forskningsavd. 2013

Fig.18.

areas consist of plating plus longitudinals. To get a sound job we think that even in the biggest ships one should aim at a plate thickness not exceeding 40 mm, and no doublers, because doublers have a tendency to cause stress concentrations and cracks and are always a nuisance in welded ships. This requirement can most easily be met by fitting a larger percentage in the cross-sectional area in the longitudinals than usual with the smaller tankers. To our opinion this is a better design from every point of view. Flat irons or slabs seem to be very suitable as longitudinals for very large tankers. They can be given a large cross-sectional area, can easily be carried through bulkheads and are stable against tripping when their height is not larger than 15 times their thickness. There is no reason why, for the biggest tankers one should not have 40 or perhaps even 50 % of the deck and bottom cross-sectional areas in the longitudinals.

I mention this because there seems to be a different view in America at the moment. The mammoth tankers with American class are built with large plate thickness and doublers and small longitudinals. The reasoning seems to be that because previous experience has been with the T2 size of tankers where the longitudinals constituted only 15% of the section modulus and the plating the remaining 85 % one should also with the much bigger tankers retain a plating or shell modulus of 85% of the total. In the T2-tankers there has been considerable corrosion of the longitudinals, which is also said to be a reason for adopting only 15 % of the area in the longitudinals. In this connection it should, however, be remembered that the T2's have been used to a large extent for light oil like petrol, whereby the corrosion will be heavy just below deck, while the super tankers are used only for crude oil, which seems to cause pitting on the horizontal lower surfaces like the bottom plating and not so much on the web surfaces of the longitudinals. Also the percentage reduction by corrosion will be much less in a slab of 30 - 35 mm thickness than it is in a web of 10 - 11 mm thickness.

Another important question for large tankers is the shear force. It has its maximum at about L/4 from both ends and this maximum divided by the cross-sectional area, i.e. the shear stress, will be much greater for large tankers than for the conventional size of ships. It is therefore necessary to consider it.

When calculating the wave shear force it is necessary to include Smith's correction just as it was shown for the wave bending

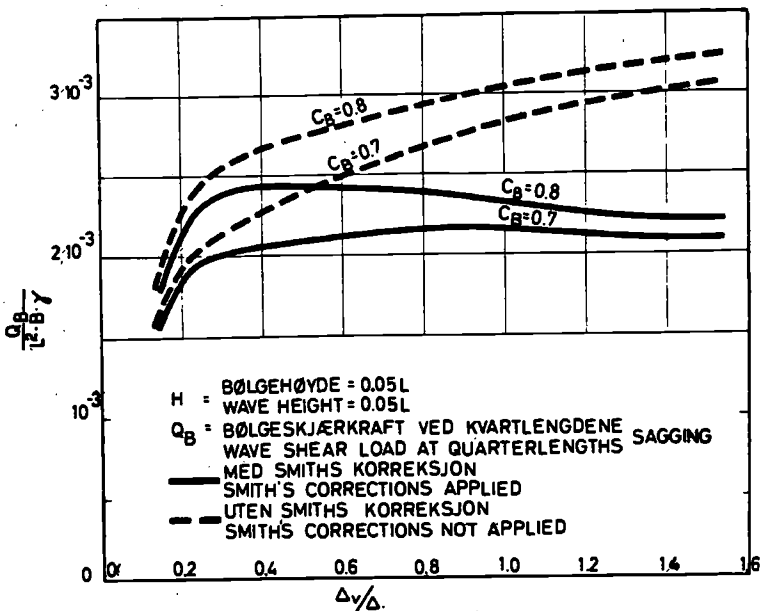


Fig. 19

moment. Fig. 19 gives the dependence of wave shear on draft with and without considering Smith's correction.

According to our investigations we can with sufficient accuracy use the following expressions for the maximum wave shear force

$$Q_{Bsag} = 0.05 \frac{C_B + 0.8}{1.6} \gamma HLB$$

$$Q_{Bhog} = 0.053 \, C_B \, \gamma HLB.$$

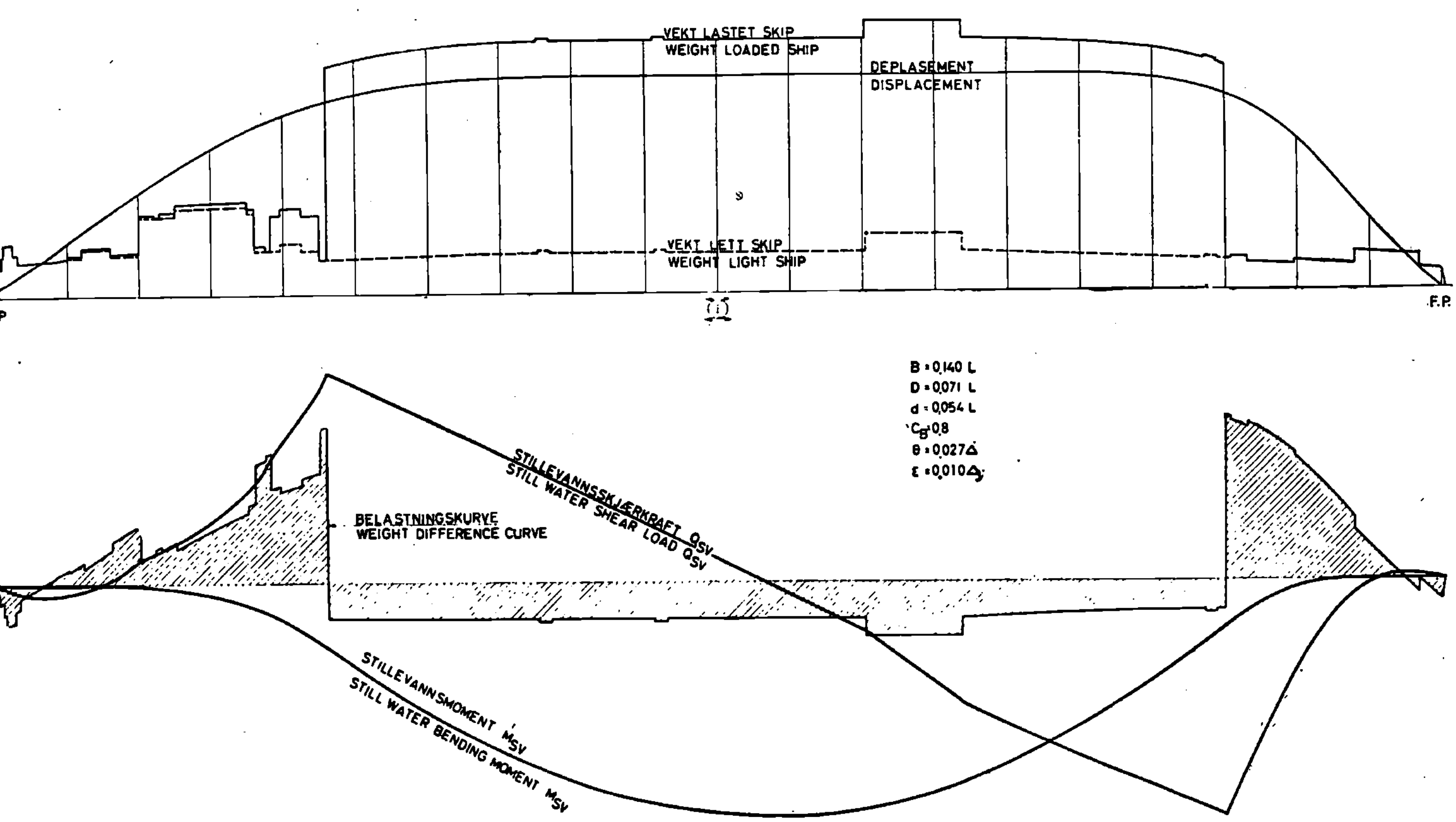


Fig. 20

Fig. 20 shows still water bending moment and shear force curves for an ordinary full tanker. The still water shear force maximum will always be at the ends of the cargo tanks, i.e. at the cofferdams fore and aft.

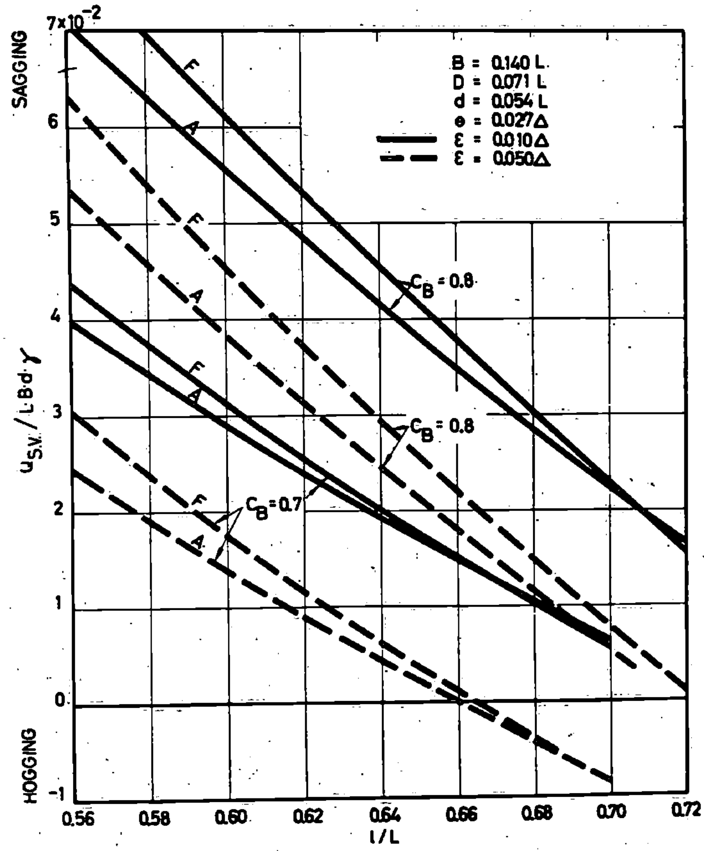


Fig. 21

Fig. 21 shows the variation of the still water shear force with the cargo tank length ratio for two different block coefficients and two different bunker weights.

The maximum still water shear force in loaded condition with a bunker weight of 1% of the displacement can be written

$$Q_{sv} = 2.1\gamma LBd(C_B-0.62)(0.76-\frac{l}{L}) .$$

It is reasonable to allow a maximum shear stress which is half of the maximum direct bending stress allowed amidships. With a similar subdivision between wave and still water stresses as for the bending stresses and with the assumption that the shear stresses are equally divided over the cross-sectional area of the vertical plating we get for this area

$$D\Sigma t = \tfrac{1}{2}(Q_B/470 + Q_{sv}/280),$$

where D is the moulded depth of the ship and Σt is the summation of the plating thicknesses of the two ship sides and the longitudinal bulkheads at the position of maximum shear. Introducing the previously given expressions and adding the correction for engine room weight we get

$$D\Sigma t = \frac{1}{65}(C_B+0.8)L^{1.6}B + LBd\left[3.84(C_B-0.62)(0.76-\frac{1}{L})+\frac{2.7-\theta}{68}\right] \text{cm}^2$$

with the minimum requirement $D\Sigma t = \frac{1}{32.5}(C_B+0.8)L^{1.6}B \quad \text{cm}^2$.

This thickness of plating is required for 10% of the ship length on both sides of the quarter length points. It is recommended to carry the longitudinal bulkheads a good distance into the engine room aft and also into the dry cargo space forward. On the other hand the thickness of the side plating amidships, where the shearing force is small has been reduced compared with previous practice and is now made $t = 10 + 0.05\,L$ mm with L in meters (> 200 m).

Often the longitudinal bulkheads are made with horizontal corrugations. Not knowing how corrugated bulkheads would take up vertical shear forces we have made comparative tests and measured the shear force deflection of plane and corrugated bulkhead models. From the results of these tests we have decided to consider a horizontally corrugated bulkhead as equivalent to a plane bulkhead with 0.8 of the thickness of the corrugated bulkhead.

With regard to local strength I may mention that we allow a maximum direct stress of 600 kg/cm^2 for bending of longitudinals in deck and bottom between supports. For longitudinals on the ship sides halfway between deck and bottom we allow 900 kg/cm^2 because here there is no stress due to the longitudinal bending of the whole ship. For intermediate positions we interpolate linearly.

All web girders are watched very closely for buckling or tripping. We have systematized the calculation of buckling stresses for rectangular plate elements so that the men controlling steel drawings at our main office can easily pick out all necessary information from diagrams.

There are, of course, a good many other details to watch when designing a large tanker. I hope, however, that the first part of my lecture where I mentioned cases of failure, together with the second part in which I have tried to give a summary of the main strength calculation, will be of some guidance when you read the rules of the Norwegian Veritas.