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THE STRENGTH STUDIES FOR THE LOCKING DEVICES OF THE BOW VISOR OF MV ESTONIA

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CONTENTS

AN ANALYSIS OF THE CALCULATION METHODS OF THE LOAD CARRYING CAPACITY OF THE LOCKING DEVICES OF THE BOW VISOR OF MV ESTONIA

1. INTRODUCTION

In the fifteenth chapter of the Final Report of the Joint Accident Investigation Commission of MV ESTONIA the calculations by the experts M. Huss, K. Rahka and J. Metsaveer on the load carrying capacity and the breaking force of the locking devices of the bow visor have been presented.. The forces affecting the bow visor are viewed as static.

The objective of this study is to present the calculations of the safe load carrying capacity and the failure force of the locking devices from a mechanics engineer's/designer's point of view.

2. LOADS AFFECTING THE BOW VISOR

2.1 The nature of the loads

The loads affecting the bow visor are mostly caused by sea loads. Besides sea forces the locking devices of the bow visor are affected by the vibration of the ship's engine and by the rolling and pitching of the vessel at sea, even when the sea loads are not directly affecting the bow visor. Both the sea load and the other loads are cyclic by nature. The cyclic nature and magnitude of the sea forces is not directly determinable and the distribution shown on Fig. 1, straight line A [1] can be used in practice.

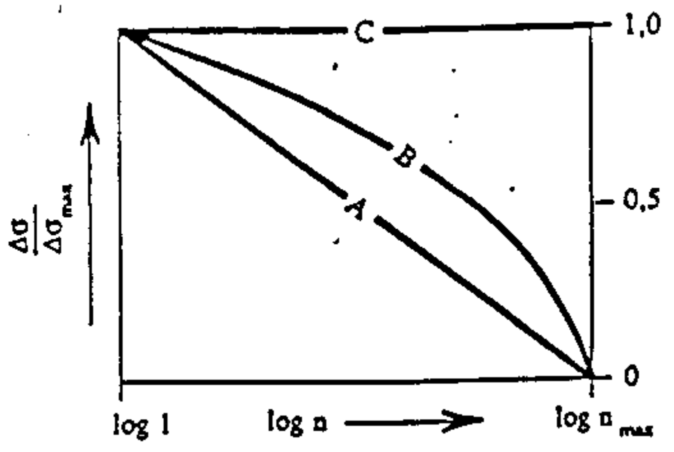


Fig. 1

where $\Delta\sigma$ — applied stress range (σ_{max}-σ_{min})

$\Delta\sigma_{max}$ — applied peak stress range within a stress range spectrum

n — the number of applied stress cycles

A: straight-line spectrum (typical stress range spectrum of seaway-induced stress ranges)

B: parabolic spectrum (approximated normal distribution of stress range $\Delta\sigma$);

C: rectangular spectrum (constant stress range within the whole spectrum; typical spectrum of engine- or propeller-excited stress range).

The theoretical calculations [3, 4] and model simulations [5] and the measurements of the pressure of sea forces on the bow visor [6] contribute to the statement that there exists alternating cyclic load from sea forces to the bow visor.

The study [4] presents a possible distribution of load on the bow visor of MV ESTONIA one hour before the accident.

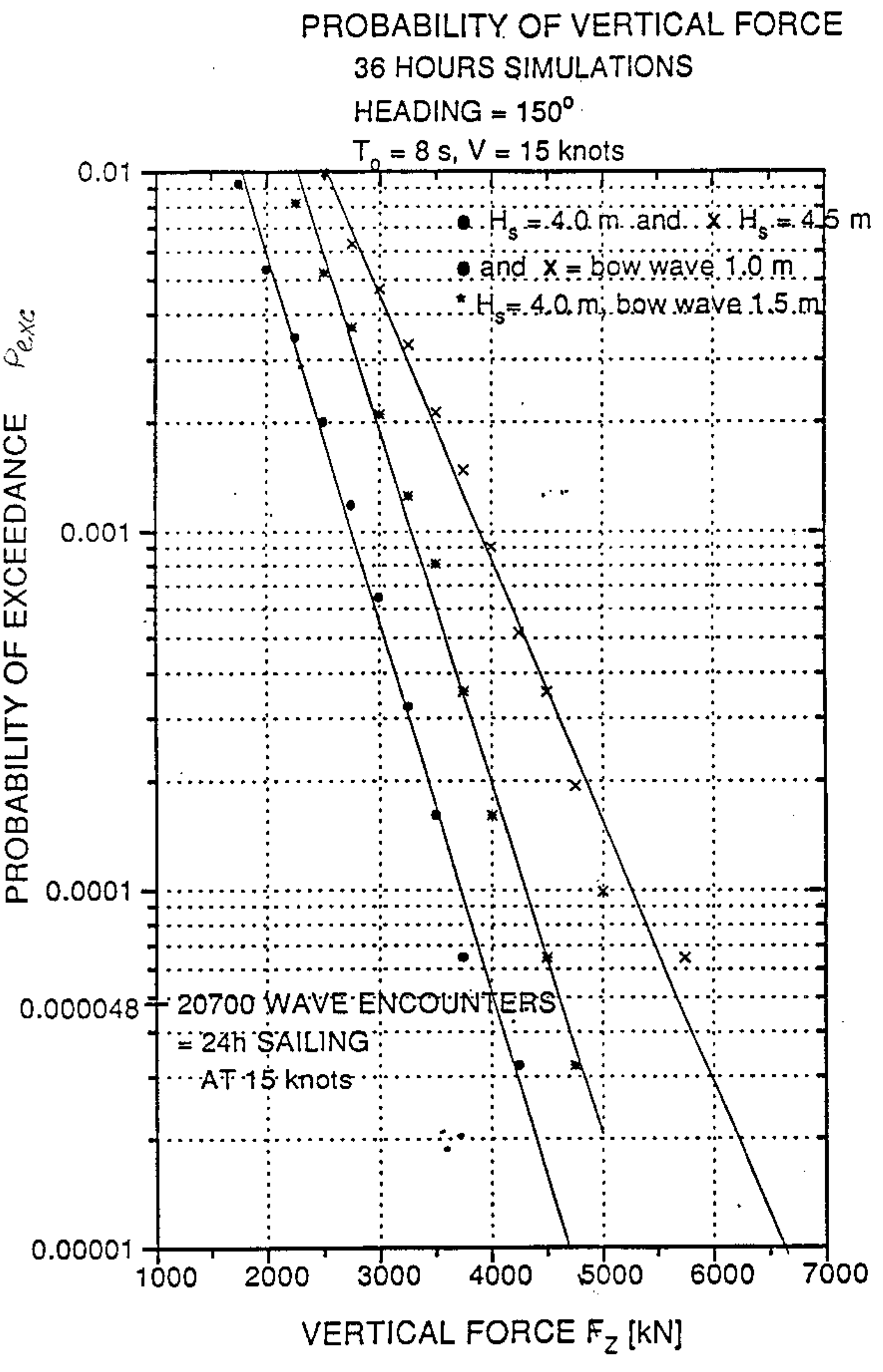


Fig. 5.8 Vertical wave loads on the visor in bow oblique seas with H_S = 4 m and 4.5 m.

Fig. 2 A

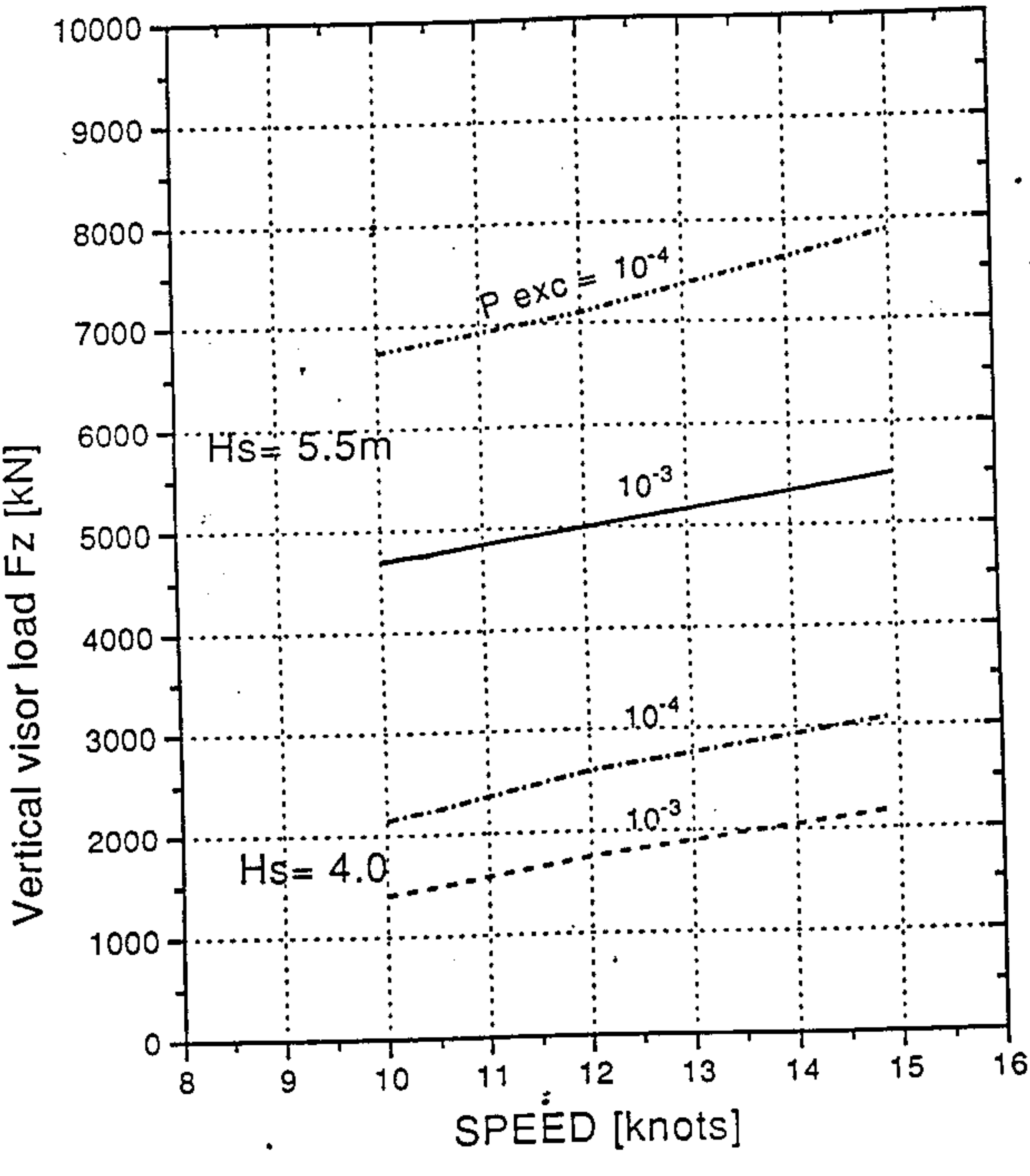
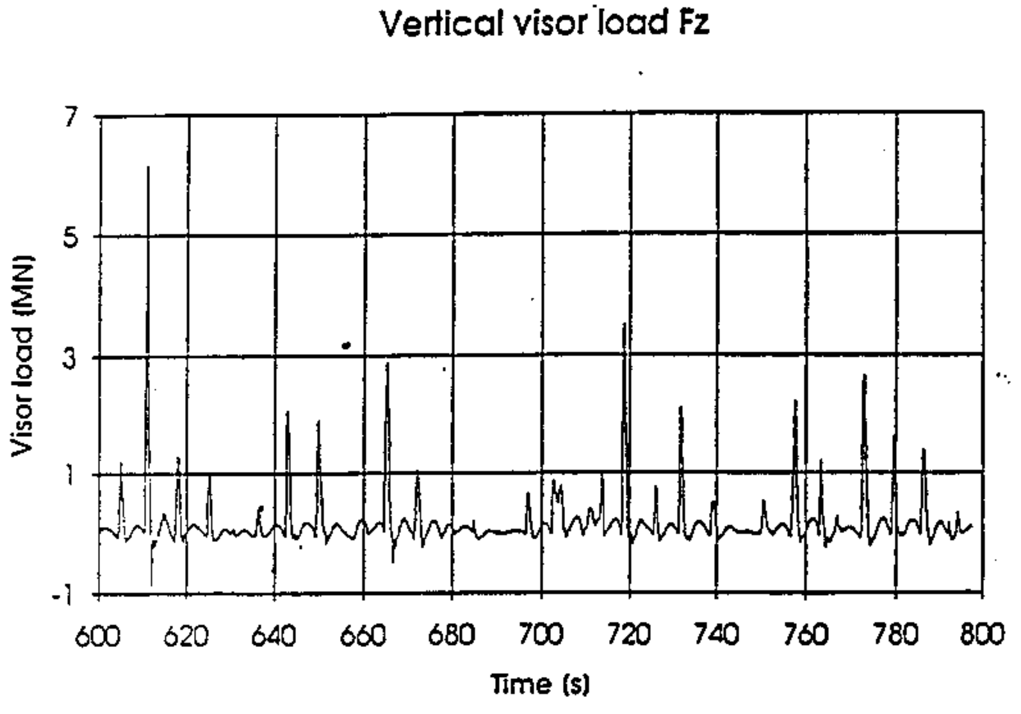


Fig. 5.9 The effect of wave height and speed on the loads of MV Estonia's visor in head seas.

Fig. 2 B

SSPA MODEL EXPERIMENTS



VTT SIMULATIONS

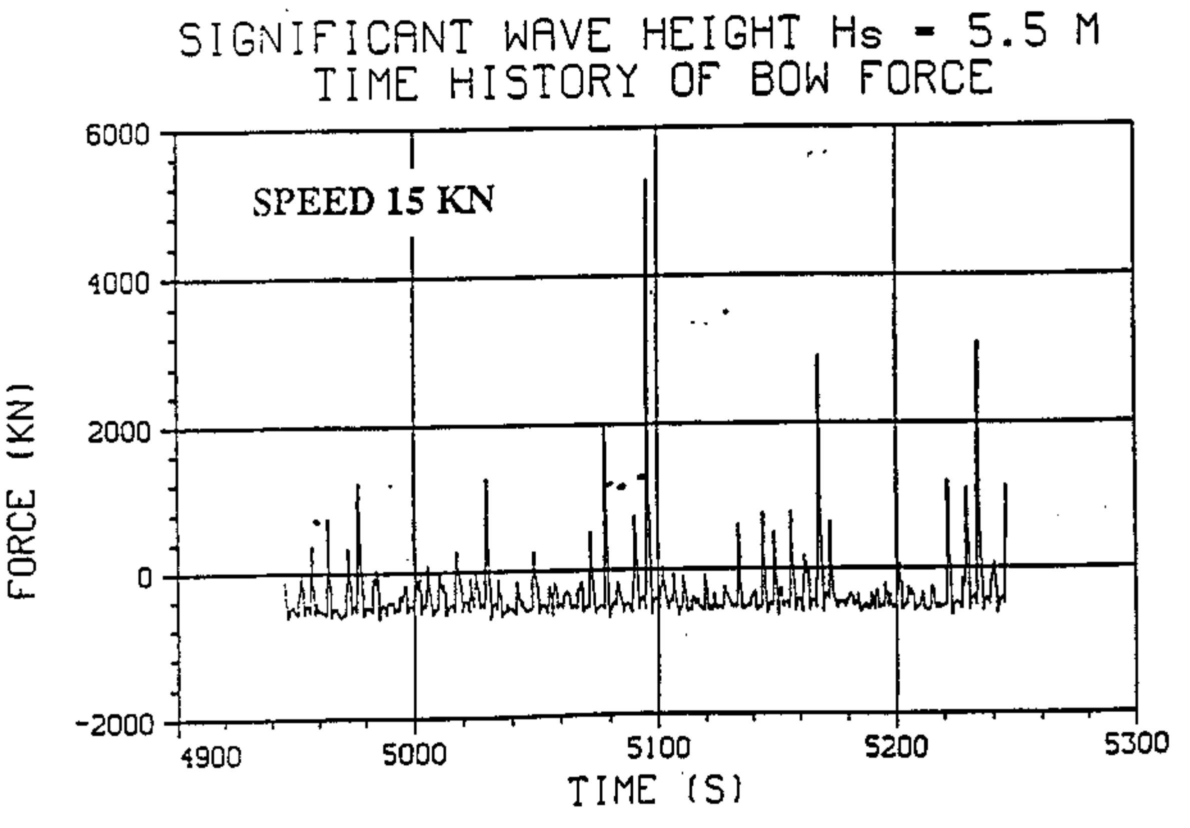


Fig. 5.10 Experimental (SSPA) and simulated records of visor loads in head seas, Hs = 5.5.

Fig. 3

TURBO TECHNIK Reparatur-Werft Dassler KG		IR 1.293-94 TS5 94-10-05 O/No. 1.293-94 Page 1/1

Inspection Report

Vessel: M/S "DIANA II"
Order-No.: 1.293-94
Port: Warnemünde
Date: 03.10.94
Subject: Bow hatch and car ramps

Inspection of bow hatch:

On deck side a crack test by "Dye-check" method was carried out on all welding areas of hatch hinge, therefore it was necessary to remove all paint etc., this cleaning was carried out by crew.

The result of crack test was good, no cracks visible.

On the hatch securing device for open position we found some welding seams on foundation of hydraulic lock bolts. This welding was possibly carried out in an emergency after damaging of bolts-foundation by collide of bolts in lock position with hatch securing eye pads.

As the second point of inspection we checked the three hatch securing-locking devices. We found them in the following condition:

The stb lockbolt shows about 20 mm clearance to the eye pad hole in fwd direction with good contact of bolt-eye pad in close position of hatch.

The portside lockbolt shows a clearance of about 40 mm to the eye pad hole in aft direction with the consequence that a movement of hatch is possible.

The center locking device is in the same condition as the port device with a clearance of about 35 mm in aft direction.

All three eye pads were welded on again, maybe after having been cut off by opening of hatch with hydraulic force by locked bolts.

As a third point we found three hydraulic cylinders with eyes welded on, two on fwd car ramp and one on port aft car ramp.

On the open portside aft car ramp it was possible to check the clearance of ramp hinge. We found most of the bolts with high clearance to the eye pads.

Comment:

On the whole we do not consider it save to run the vessel under this condition.

Wilhelmshaven, 05.10.94

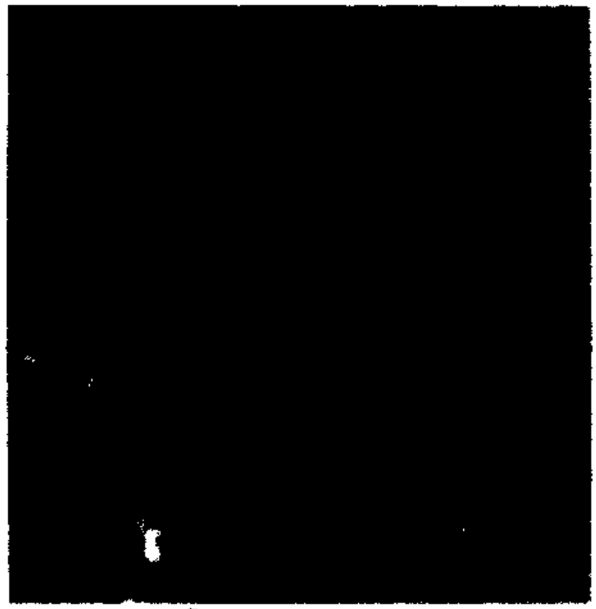
TURBO-TECHNIK
REPARATUR - WERFT
Dassler KG

i.A.

M. Gronewold

Fig. 4

ESTONIA



SB Visir Låsning
93 01 16 DIANA II

BB visir Låsning
93 01 16 DIANA II

Fig. 6

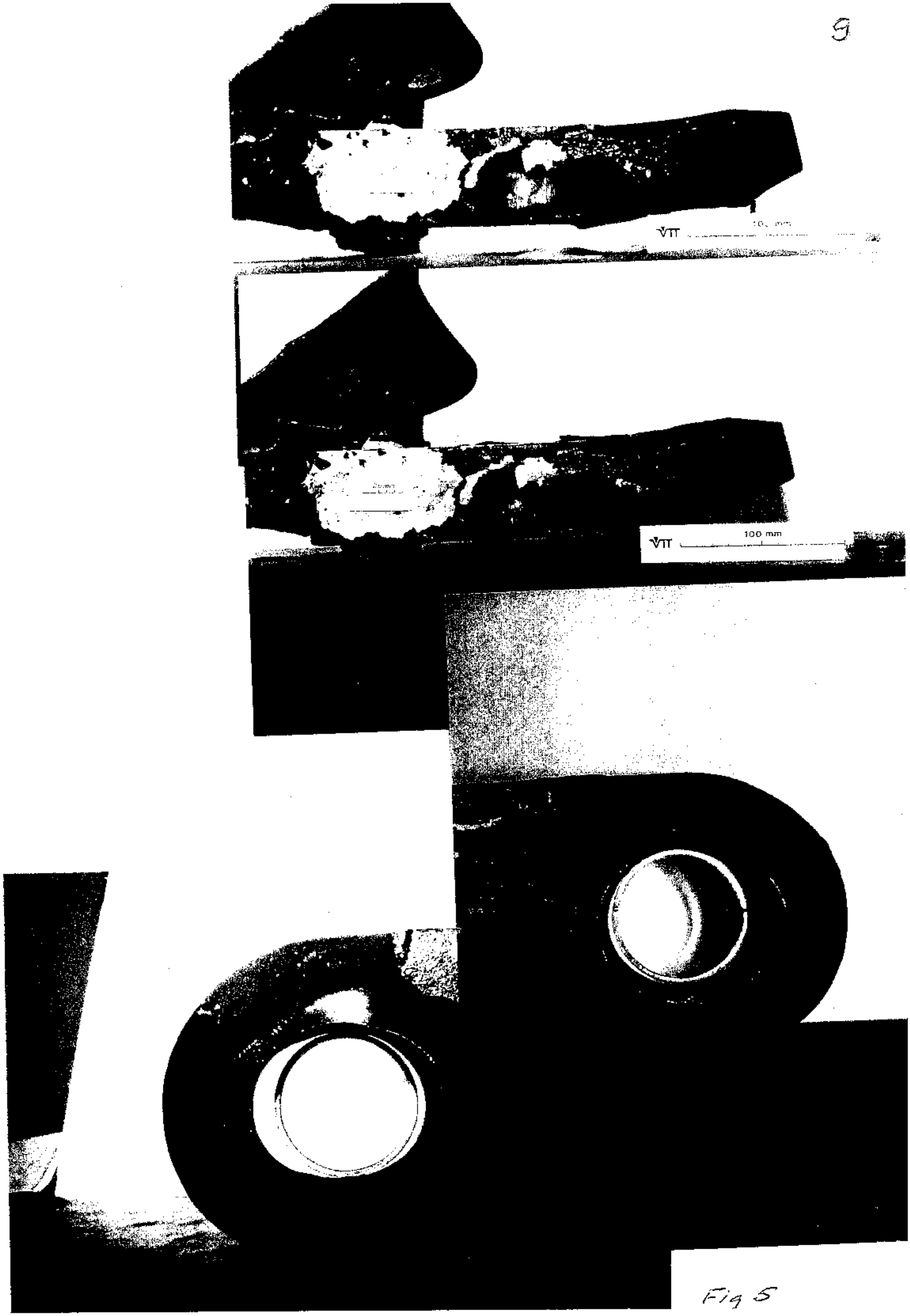


Fig 5

Bureau Veritas MARINE BRANCH DETAILED REPORT OF SURVEY

Annex No	8	to Report No	HBG/93/7-A	Page No	1	Total Pages	1		
Register Number	35 Y 002	Name of Ship	"DIANA II"						
Cert. concerned (one square only to be ticked off)	PROV	HULL X	MACH	AUT	MB	AB	RMCI		
Visa No endorsed	7	or survey of C.S.		or D.S.		items (continuation to AdE 2528)			
Survey carried out from	16/01/93	to	17/01/93						

REPORT OF SURVEY

On request by the Chief Engineer survey of the closing devices for the bow door carried out and following damages were found on the bow door.

The lug for SB lock plunges was lost.

The lug in center line (the "Atlantic lock") was bent and the weld cracked.

The lug for Port side lock plunges was bent and the weld cracked.

A minor crack at the hinge on SB side.

The girder in center line and two webs on SB side cracked.

Following repairs carried out:
The lug SB side renewed with a doubling plate on the back side.

The lug in center line faired and rewelded. The stay above the lug renewed.

The lug Port side faired and the crack chiselled and welded.

The crack at SB hinge chiselled and welded.

The cracks at the girder and the webs chiselled and welded.

Function test of the bow door carried out and found to be in order.

New name

Mar Baltica

BUREAU VERITAS EXPERT. DE GÖTEBORG PARIS 1828 REGISTRE INTERNATIONAL

Fig. 7

The significant wave height is 4 and 4.5 m and during the last hour the ship was affected by approximately 1000 waves and the possible occurrence frequency of the greatest vertical force was 1/1000 and the greatest vertical sea force has been estimated to be 5000 kN or less probably 6000 kN, as shown on Fig. 2 [4].

The theoretical calculations [4] and model simulations [5] are in agreement and have been presented on Fig. 3 [4].

In case of waves of lesser magnitude the forces affecting the bow visor are also smaller and directed towards the closing direction of the bow visor. Actually, however, the number of the affecting cycles is significantly greater.

The existence of the cyclic loads is confirmed by the inspection of the locking devices of MV DIANA II on 30. 10. 94 [7], i.e. one week after the accident of MV ESTONIA (Fig. 4).

During the inspection the following clearances were measured between the locking bolt and the eye.

The Atlantic lock — 35 mm, aft direction
stb side lock — 20 mm, fwd direction
portside side lock — 40 mm, aft direction

Such increases in clearances occur during operation **only due to alternating cyclic loads**.

The differences in the amounts and directions of the wear indicate that:

1. The distribution of loads on the locking devices is not uniform. The distribution of loads cannot be determined as the entire locking installation is not statically determined.
2. Some of the welds of the locking devices (being sensitive to cyclic loads) are subject to pressure and some to tension.

The wear of the eye in the aft direction indicates the effect of the sea forces in the direction of the opening of the bow visor and the expansion in the forward direction indicates the effect in the closing direction of the bow visor.

The existence of alternating cyclic load is also proven by the fact that the lug of the Atlantic lock of the bow visor of MV ESTONIA has hammered eyes on both sides (3.7 and 4.3 mm) in fwd direction. Hammered eyes can only be caused by a great number of impacts between the bolt and the lug (Fig. 5).

The existence of the alternating cyclic loads is also confirmed by the damages to the locking devices of MV DIANA II 16. 01. 93 (Figs. 6 and 7).

The lug of the stb side lock has been torn out of the plating of the bow visor and the portside lock was bent and the weld cracked. The photos

show a darkening of the paint at the upper edge of the left side lock, caused by corrosion. Corrosion occurs in the proximity of cracks. Consequently, there is a crack in the weld joint, which has been caused by cyclic load. As shown later, the upper edge of the side lock is under greater load and the first crack appears there.
The analogous failures of locking devices on RO-RO passenger ferries operating on the Baltic Sea indicate the existence of alternating cyclical loads and clearances between the locking bolts and the eyes. The clearances increase during operation accompanied by an increase in the lack of uniformity in the distribution of the loads, causing the failure of the lock. The incomplete lists include 8 marine accidents involving Finnish and Swedish RO-RO vessels (mainly passenger ferries) during the years 1973-1993 that were due to the damage to the locking devices of their bow visors. Four accidents occurred within a year after the construction. The conclusion to be drawn from this is that the locking system with clearances used in Scandinavia (one bottom lock/Atlantic lock and two side locks) is underdimensioned and poorly designed due to the clearances, sooner or later resulting in damage or maritime accident.
RO-RO ships built in the former USSR (years 1975 and later) have a clearance-free locking system — forced locking (a power screw and a nut). The total number of locking devices is 14, 10 of these bottom locks and 4 side locks. There have been no reports on the failure of these locking devices. The Estonian Shipping Company has 4 ships of this kind and they are operated for 20...22 years.

Conclusion.

1. The forces affecting the locking devices of the bow visor (sea forces, vibration and forces caused by the rolling and pitching of the vessel) are alternating and cyclic by nature.
2. These forces vary significantly both in extent and direction and cause different stresses (tension-pressure, bending, shear) in the components of the locking devices.
3. All the locking devices have components with weld joints which are especially sensitive to alternating cyclic loads.
4. **The cyclic nature of the loads calls for fatigue calculations when calculating the strength of the components of the locking devices.**
5. As the number of operative cycles and the magnitude of the operative forces are undeterminable, the strength calculations have to be performed for fatigue and the minimal allowable stresses should be used, providing for larger safety factors.

3. ALLOWABLE STRESSES

The main type of steel used in shipbuilding is St 37 — a mild carbon steel with good welding characteristics.
The main characteristics of this steel are the following:

ultimate stress $\sigma_u = 340 \div 470$ N/mm^2,
yield strength $\sigma_y = 235$ N/mm^2,
relative lengthening $A_s = 22\%$.

Allowable stresses are affected by the situation of the applied loads and should therefore be discussed separately.
The Bureau Veritas rules from 1977 [8] do not directly determine the allowable stresses of the attachments of the locking devices of the bow visor, but such stresses have been determined for the attachments of moving platforms.
As moving platforms, ramp and bow visor are all operating in difficult conditions, the allowable stresses valid for the attachments of the moving platforms can be used for the calculations for the attatchments of the locking devices of the bow visor.
The 1977 rules by Bureau Veritas set the minimum safety factor at 5.
The components of the locking devices must be based on direct calculations and the following allowable stresses must be used:

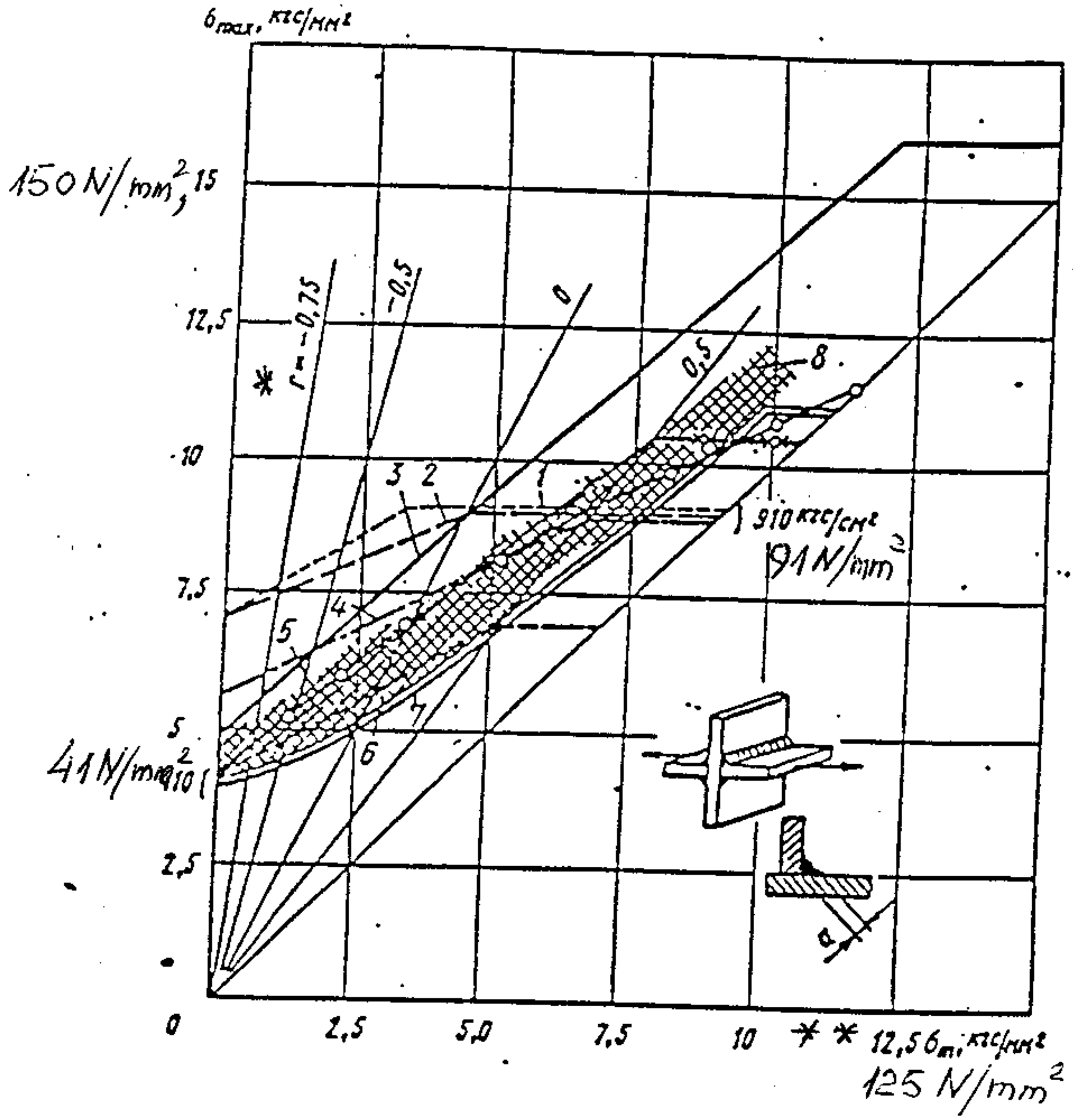
bending stresses $\sigma \leq 85$ N/mm^2,
shear stresses $\tau \leq 42$ N/mm^2,
combined stresses $\sigma_c = \sqrt{\sigma^2 + 3\tau^2} \leq$ 112 N/mm^2.

The 1987 and 1996 Bureau Veritas rules directly specify the following allowable stresses for the locking devices of the bow visor:

bending stresses $\sigma \leq 120/k$ N/mm^2,
shear stresses $\tau \leq 80/k$ N/mm^2,
combined stresses $\sigma_c = \sqrt{\sigma^2 + 3\tau^2} \leq$ 150/k N/mm^2.

The material factor k is determined by the yield point of the steel and equals

σ_Y	k
235	1
263	0.91
315	0.78
355	0.72
390	0.66



The permitted stress for unprocessesed T-weld joint of CT3 type steels (C~0,15) from different sources [10]

1. Poland
2. East Germany
3. West Germany
4. Czekhoslovakia
5. Austria
6. Switzerland
7. USSR
8. Fatigue limits

Fig. 8

$$* \quad r = \frac{\sigma_{min}}{\sigma_{max}}$$

$$** \quad \sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

The latter allowable stresses are the same as stated by in the rules of the Germanischer Lloyd [1]. In the same source allowable stresses for statically loaded weld joints have been provided. For a steel with a yield point of $\sigma_y = 235$ N/mm^2 the allowable shear stress is $\tau \leq 115$ N/mm^2.

The allowable stresses for fillet(T) weld joints for Cm3 steel (C ~ 0.15%) according to various sources is τ=41÷91 N/mm^2 (Fig. 8 [9])

For alternating cyclic loads the allowable fatigue shear stresses for a fillet(T)-weld are in the range τ = 35÷50 N/mm^2 [1].

As the shipbuilder has not determined the lifespan of the locking devices and the magnitudes, directions and working cycles of the loads affecting the locking devices are unknown, the only solution is to use the fatigue strength necessary to ensure the safe operation of the devices for the calculations.

Conclusions

1. The allowable stresses for weld joints according to different sources are in the range of τ = 35÷90 N/mm^2.
2. The allowable stresses for bending (tension) are in the range σ = 85÷120 N/mm^2.
3. The fatigue strength calculations should be the basis for design.
4. **If the operational loads are greater than the fatigue limit, the breaking of the device is inevitable.**

4. THE CALCULATION OF THE LOAD CARRYING CAPACITY OF THE LOCKING DEVICES OF THE BOW VISOR

4.1. Introduction

The objective of this chapter is to calculate the load carrying capacities and the breaking forces of the locking devices of the bow visor. Methods of engineering calculations which are recognized world-wide are used in these calculations.
The bow visor was attached to the vessel with three locking devices and pivoted around two hinges on the deck for opening and closing the visor. Thus there are five attachment points which form a statically undeterminable system. Therefore it is impossible to directly calculate the

forces affecting the locking devices. In the calculations provided by the shipyard the general load of 5000 kN (500 tons) is distributed equally (1000 kN) over the five attachment points.
The drawings provided by shipyard and by the company designing the locking devices (von Tell) are assembly drawings and they lack the dimensions of the attachments to the bow visor and to the hull. The drawings lack symbols and characteristics for welds. Some examples of the drawings by the shipbuilder (Fig. 9) and by von Tell (Figs. 10, 11) have nevertheless been included.
The bow visor was secured by a bottom lock (the Atlantic lock) and two side locks. All the three lock are of similar bolt-bushing construction. To enable the pivoting of the bow visor, there were two hinges on the deck. The calculations of the load carrying capacities and breaking forces of the bottom lock, side lock and the hinges are presented below.

4.2 The strength calculation of the bottom lock

The bottom lock is shown in Fig. 12.
The bottom locking device consists of a locking bolt 5, movable horizontally in a transverse direction guided by a bolt housing 2. In its extended (closed) position the tip of the bolt is engaged in the support bushing 1. The bolt housing was fixed to the forepeak deck by two steel lugs (II and III) and the support bushing was secured with a third similar lug (I). A mating lug 3, which is between the bolt housing 2 and support bushing 1, is attached to the visor itself. When the visor is in a closed position, the extended bolt 5 engaged the mating lug 3 and the support bushing 1.
The bolt 5 was moved in the bushings 1 and 2 by a hydraulic actuator.
The lug of the bow visor 3 was secured to the transverse beams of the bottom structure of the bow visor.
The lug had a hole for the locking bolt with an original diameter of 85 mm. During the operation of the vessel the diameter of the hole increased and after the shipwreck the hole was oval, the dimensions being 90 by 105 mm.
There are no detailed drawings of the locking devices, some general dimensions have been presented on the assembly drawings by von Tell, the supplier of the locking devices.
All of the three lugs of the bolt housing and the support bushing of the bottom lock are broken. The lug of the bow visor did not fail but tore apart the lugs attached to the forepeak deck (I, II, III) and was then separated.

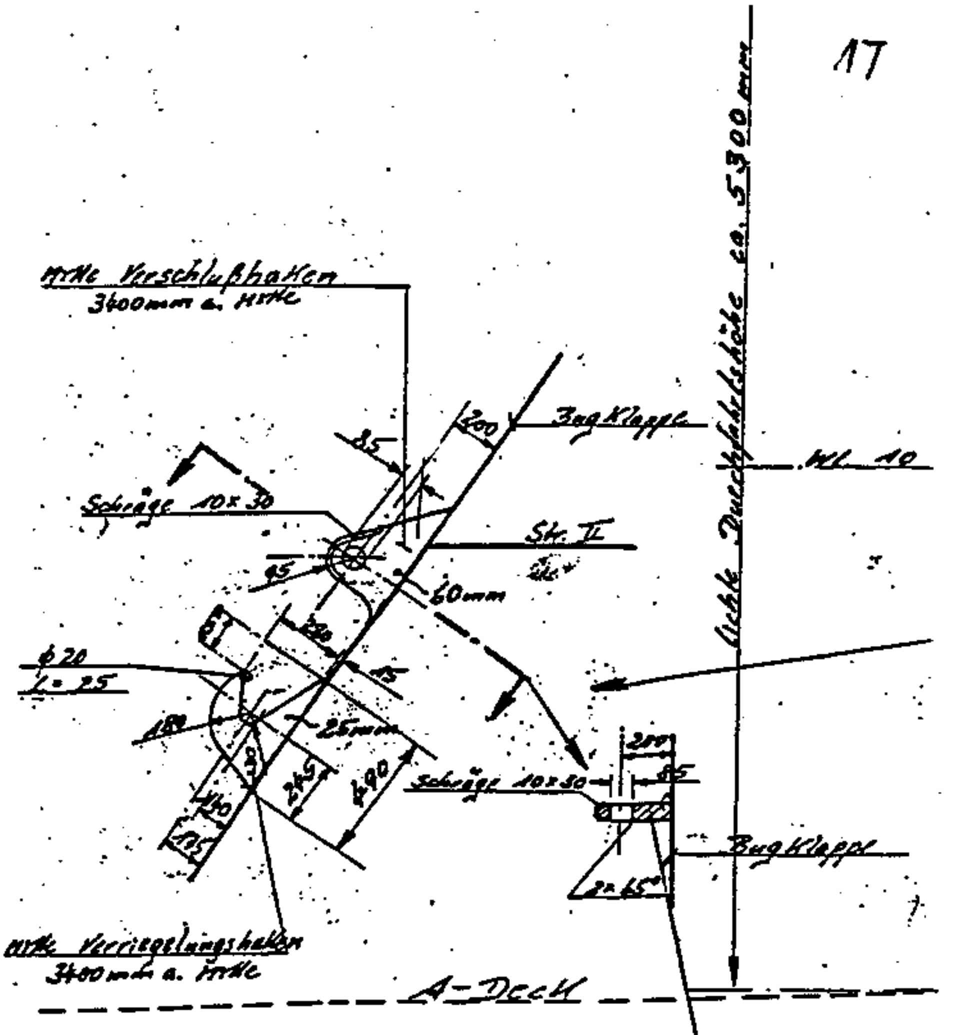

Mitte Verschlußhaken
3400mm a. Mitte
Bugklappe
WL 10
Schräge 10x30
Str. II
60mm
Φ20
L=75
25mm
Schräge 10x30
Bugklappe
2x45°
Mitte Verriegelungshaken
3400mm a. Mitte
A-Deck

Fig. 9

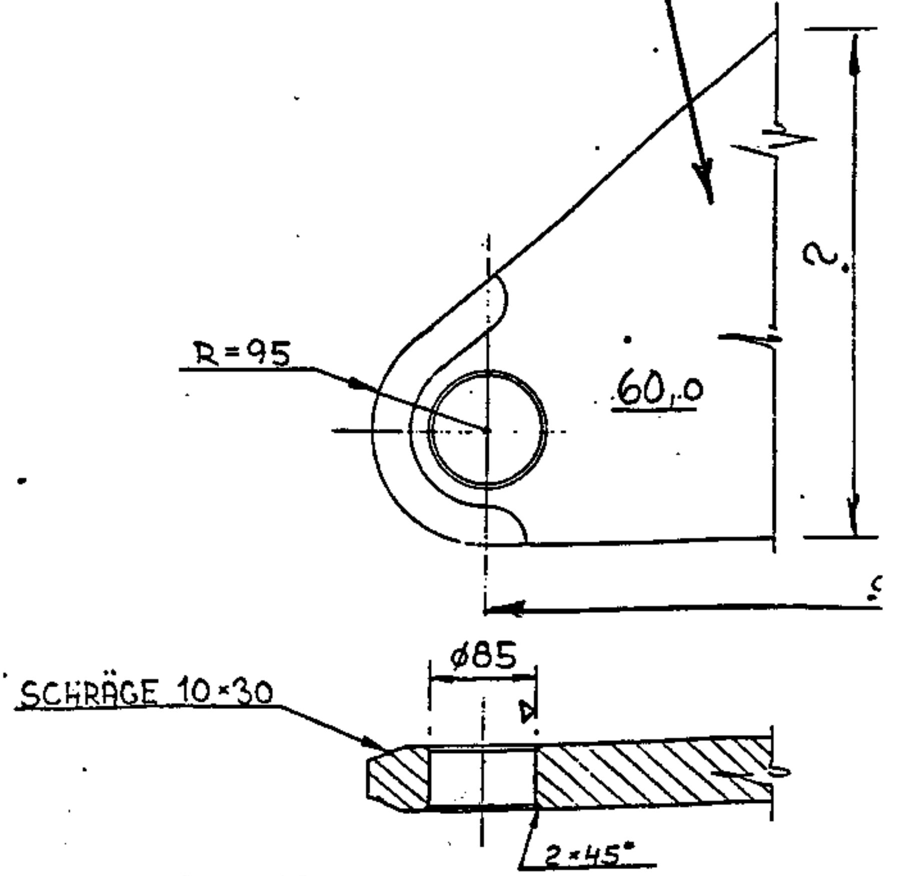
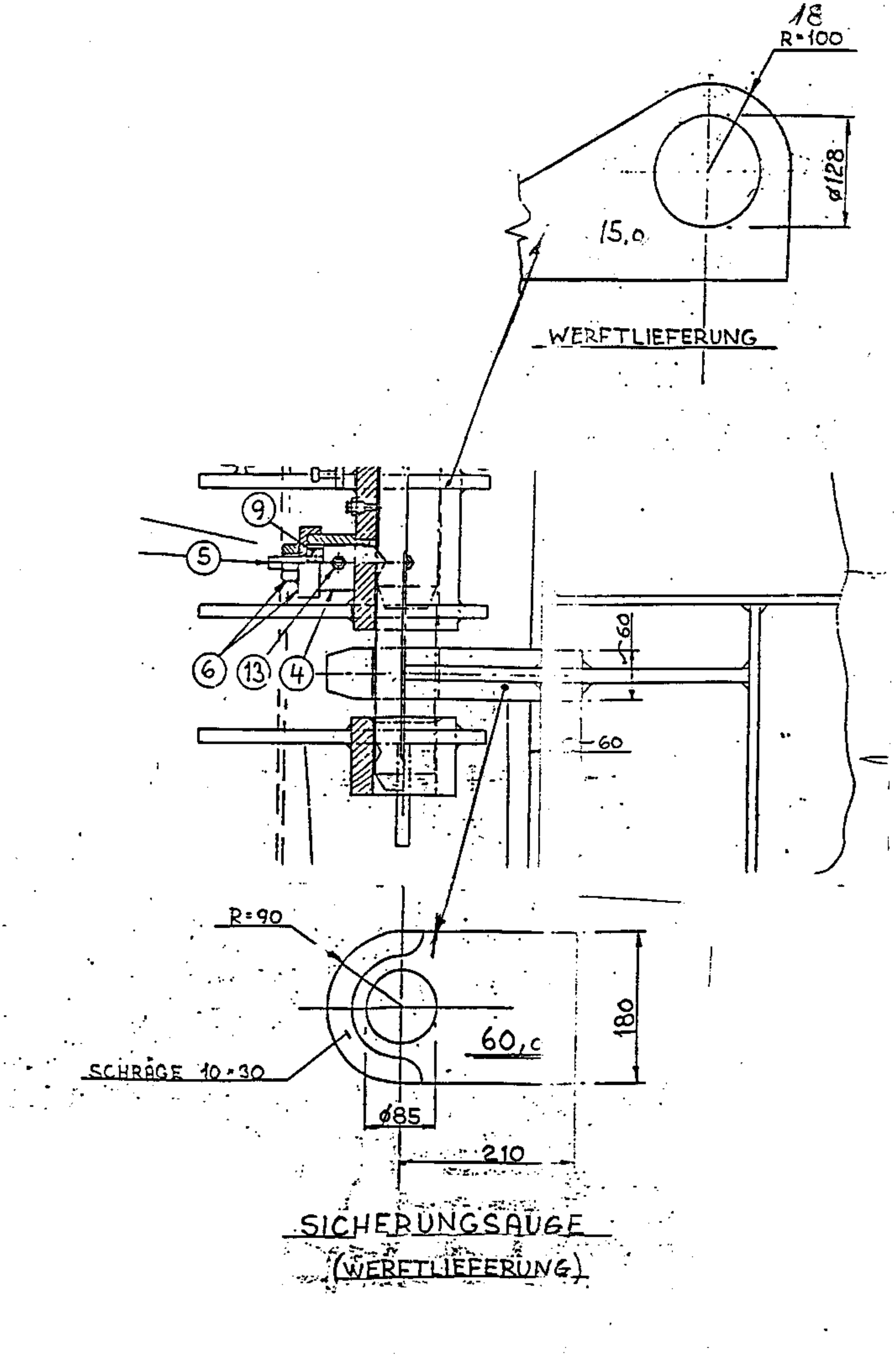

R=95
60,0
Ø85
SCHRÄGE 10×30
2×45°

Fig. 10



18
R=100
ø128
15,0
WERFTLIEFERUNG
9
5
6
13
4
60
60
R=90
60,c
180
SCHRÄGE 10×30
ø85
210
SICHERUNGSAUGE
(WERFTLIEFERUNG)

Fig. 11

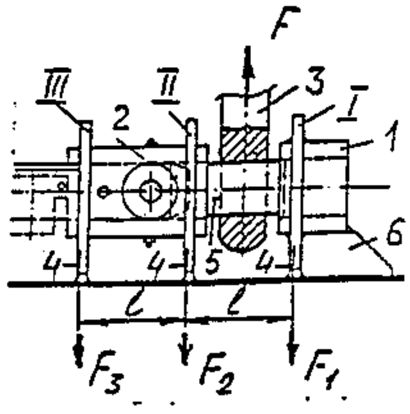


Fig. 12

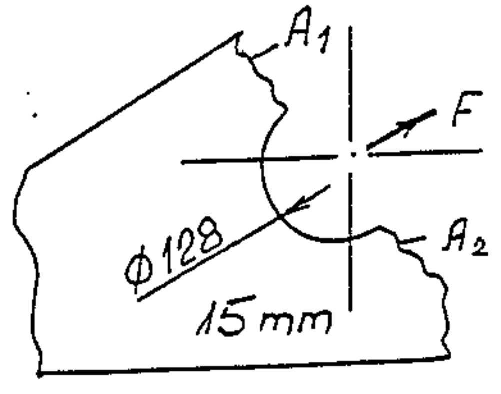


Fig. 13

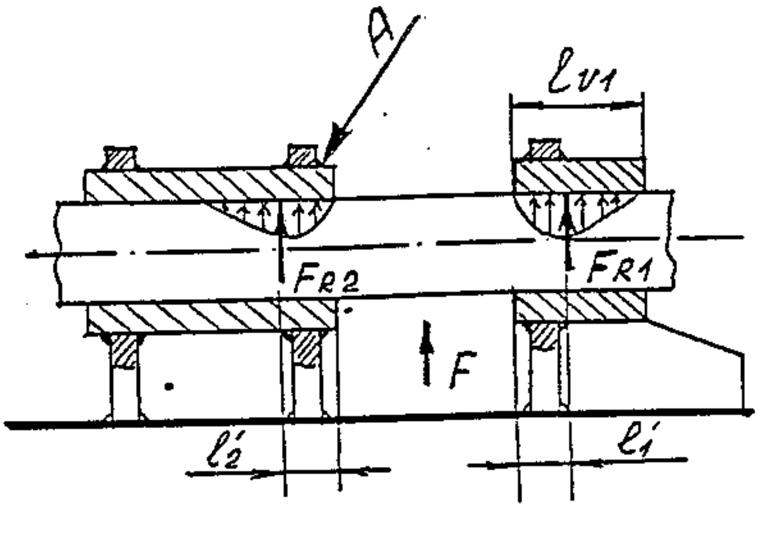


Fig. 14

The dimensions of the failed lugs are shown in Fig. 13.
The calculation scheme of the bottom lock is shown in Fig. 14.
The force F (Fig. 14) affecting the supports of the bolt 1 can be viewed as a shaft on a sleeve bearing [10].
F_{R1} and F_{R2} are the forces affecting the bushings.
The distance l_1' of the force F_{R1} can be derived from

$$l_1' = (0.25 \div 0.3) l_{v1} \le 0.5d$$

Provisionally we may assume that the reactive forces on both sides of the lug 3 are equal, resulting in:

$$F_{R1} = F_{R2} \text{ and } F_1 = F_2 = F/2 \ .$$

In that case $F_3 = 0$.
The failure of the bottom lock can be viewed as the failure of the weld joints between the bushings and the lugs (Fig. 14 arrow A) and the failure of the lugs with surfaces A_1 and A_2.
The bottom lock fails in two stages. The reason for this is the fact that the bushing in the lug has a clearance relative to the lug and this causes the failure of the weld joints in the first stage. Subsequently the load is transferred from the bushing to the lugs and the lugs fail. As the failure takes place in two stages, the load carrying capacity of the bottom lock can not be considered equal to the sum of the failure loads of the weld joints and the lugs.

The first stage of the failure:

The strength calculations for the weld joints.
The weld joints are calculated for shear stresses, hence

$$\tau = \frac{F}{2} \cdot A \quad ; \quad F = \frac{2\tau}{A},$$

where A is the failure surface of the weld joint (Fig. 13). The leg of the weld joint k is approximately 3 mm [2], hence

$$A = 2\pi \cdot d_1 \cdot 0.7k = 2 \cdot 3.14 \cdot 128 \cdot 0.7 \cdot 3 = 1688.9 \,\text{mm}^2$$

Let us determine the breaking force of the welds of the two lugs F_{wu}, when the ultimate stresses are $\tau_u = 0.6\sigma_u = 0.6 \cdot 400 = 240\,\text{N/mm}^2$, where

$$\sigma_u = 400\,\text{N/mm}^2.$$

$$F_{wu} = 2\tau_u \cdot A = 2 \cdot 240 \cdot 1688.9 = 810.672N = 810.67kN = 811\,kN$$

Now let us determine the calculational load carrying capacity F_{wp} of the weld joints if the allowable stresses are $\tau_p = 80\,\text{N/mm}^2$

$$F_{wp} = 2\tau_p \cdot A = 2 \cdot 80 \cdot 1688.9 = 270224N \approx 270\,kN$$

Let us calculate the calculational load carrying capacity of the weld joints F_{wp}, if the allowable stresses $\tau_p = 42$ N/mm^2 (BV 1977 rules)

$$F_{wp} = 2\tau_p \cdot A = 2 \cdot 42 \cdot 1688.9 = 141867.6N = 141.87kN \approx 142 \text{ kN}.$$

The calculational load carrying capacities F and the factor of underdimensioning K of the weld joints are presented in Table 1.

Table 1

τ, N/mm^2	F, kN	K = 1000 / F
42	142	7.1
80	270	3.7
240	811	1.23

The second stage of the failure:

Strength calculations for the lugs.

The lugs break with breaking surfaces A_1 and A_2 (Fig. 13).

Let us calculate the breaking surfaces, if the thickness of the lug was t = 15 mm.

$$A_1 = A_2 = (R - d_1 / 2) \cdot t = (100 - 128 / 2) \cdot 15 = 540 \, \mathrm{mm}^2$$

Thus the breaking surface of a lug is $A = A_1 + A_2$

$$A = 540 + 540 = 1080 \text{ mm}^2 .$$

Let us now calculate the breaking force to tension F_{Tu}, when the breaking stress to tension is $\sigma_u = 400$ N/mm^2 for two lugs if $F_{R1} = F_{R2}$, See Fig. 14.

$$F_{Tu} = 2 \cdot \sigma_u \cdot A = 2 \cdot 400 \cdot 1080 = 864000 N = 864 \text{ kN}.$$

Let us determine the calculational load carrying capacity of the two lugs F_{Tp}, if the allowable stresses were $\sigma_p = 120$ N/mm^2

$$F_{Tp} = 2 \cdot \sigma_p \cdot A = 2 \cdot 120 \cdot 1080 = 259200 N = 259.2 kN \approx 260 \text{ kN}.$$

The calculational load carrying capacity of the lugs F_{Tp}, when allowable stresses $\sigma_p = 85$ N/mm^2 (BV 1977 rules)

$$F_{Tp} = 2 \cdot \sigma \cdot A = 2 \cdot 85 \cdot 1080 = 183600 N = 183.6 kN \approx 184 \text{ kN}.$$

The calculational load carrying capacities F and the factor of underdimensioning K of the lugs are presented in Table 2.

Table 2

τ, N/mm^2	F, kN	K = 1000 / F
85	184	5.5
120	260	3.8
400	864	1.1

Conclusions

1. It must be observed that failure of the bottom lock took place in two stages. First, the failure of the weld joints and secondly, the failure of the lugs.
2. The bottom lock is underdimensioned by a factor of 1.1 ÷ 5.5 at calculational loads of F = 1000 kN.

4.3 The strength calculations of the side lock

There are two side locks which consist of two lugs, mounted to the aft bulkhead of the bow visor and extended, when the visor was closed, into two holes in the front bulkhead of the hull, one at each side of the ramp opening. In the closed position hydraulically operated bolts engaged holes in the visor lugs (Fig. 15). The hydraulic bolt installations were similar to that of the bottom lock.

The side locks failed at the welding of the lugs and the aft plating of the bow visor. The weld leg was 8 mm long. The lug is welded to the plating as a fillet joint (T-joint) along a closed contour.

Such fillet joints are calculated for shear stress τ and are calculated for a plane at a 45° angle at the leg k of the weld, hence the width of the weld is 0.7k (Fig. 16).

As the side locks operate at alternating cyclic loads the nature of which cannot be determined, the strength calculations have to be performed using the minimal allowable stresses in the range of $\tau = 42 \div 80$ N/mm^2.

The scheme of the calculations is shown in Fig. 16. When calculating shear stresses at sign-changing loads, a 180 degree change in the direction of the force does not affect the stress state of the weld.

The force F_1 affecting the lug causes bending and section stress and the force F_2 causes tension-pressure stresses. Let us determine the stresses in the weld joint:

Tension-pressure	$\tau_1 = F_2/A$
Area	$A = 2(B+H)\ 0.7k$
Shear	$\tau_2 = F_1/A$
Bending	$\tau_3 = M\,y/I$
Bending moment	$M = F_1 L$
Inertia moment	$I = I_1 - I_2$

$$I_1 = (B+1.4k)(H+1.4k)^3/12$$
$$I_2 = BH^3/12$$

The distance of the weld in question from the axis

$$y = H/2+d$$

The arm of force F_1 L = 210 mm.

The calculational width d of weld joint

$$d = 0.7k = 0.7 \cdot 8 = 5.6 \text{ mm}$$

The projections of force F

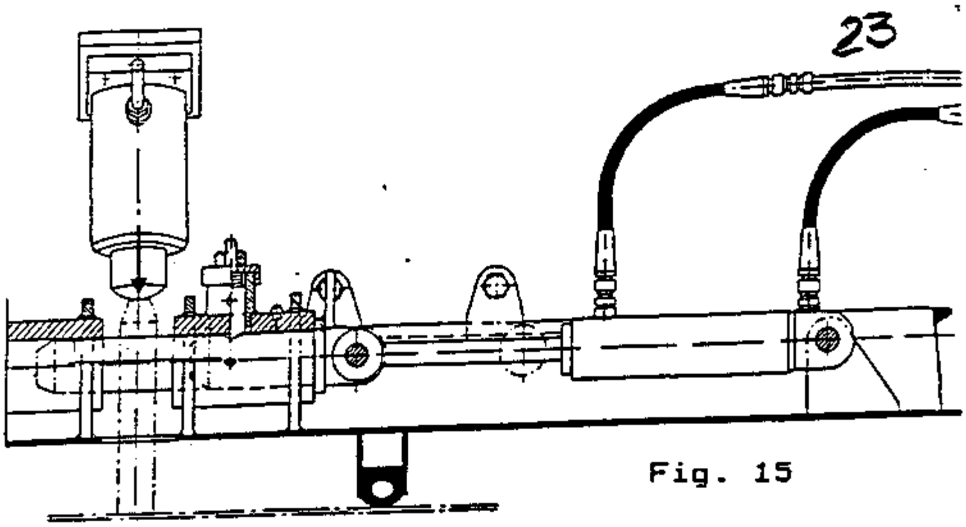


Fig. 15

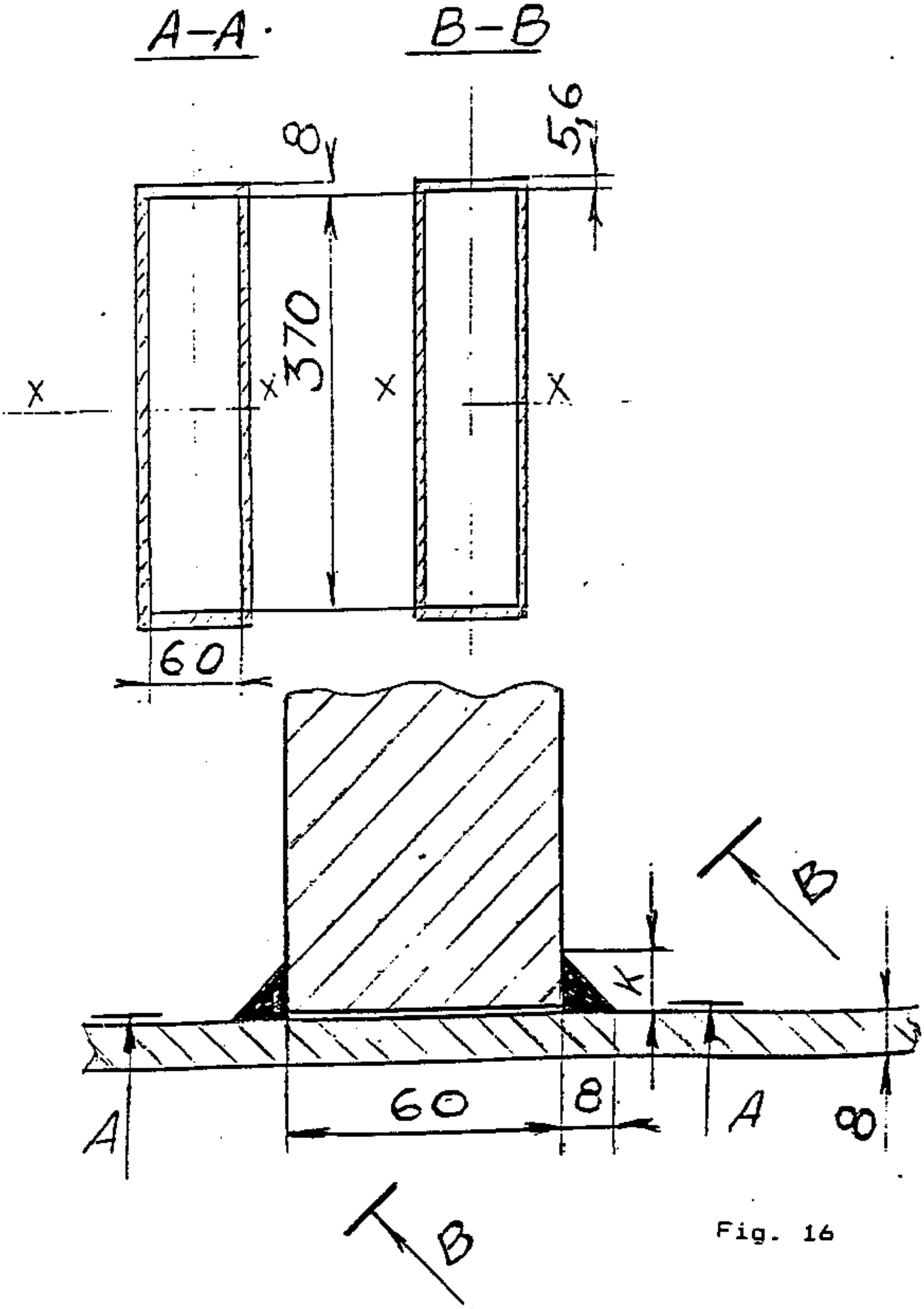
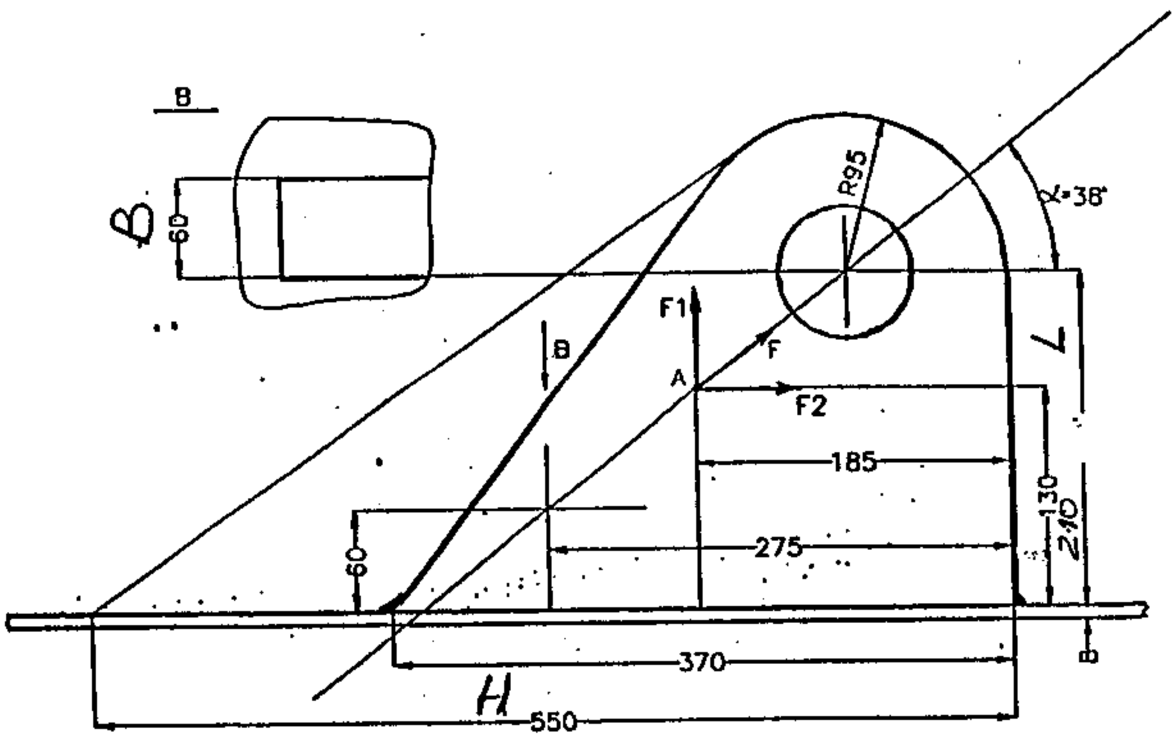


Fig. 16

SIDE LOCK



$$\tau_1 = F_1/A\ ;\quad \tau_2 = F_2/A\ ;\quad \tau_3 = F_2L/W\ ;\quad \tau = \sqrt{(\tau_1 + \tau_3)^2 + \tau_2^2} \le \tau_p$$

Fig. 16 A

```
REM     QBASIC A.INGERMA  20.01.97
    PI = 3.1416

 AL = 38 * PI / 180
          H = 370
k = 8
B = 60
LV = H / 2 - 95
L1 = LV * SIN(AL) / COS(AL)
LH = 210 - L1

     A = 2 * .7 * k * (B + H)
 I = ((B + 1.4 * k) * (H + 1.4 * k) ^ 3) / 12 - (B * H ^ 3) / 12
 W = I / (H / 2 + k)
 CONSTANT = ((SIN(AL) / A + COS(AL) * LH / W) ^ 2 + (COS(AL) / A) ^ 2)^
 FC = ps / CONSTANT
 PRINT "LH,L1,LV,="; LH; L1; LV
 PRINT "CONSTANT="; CONSTANT
 PRINT "FORCE CAPACITY TONS="; FC / 10000
 LPRINT "CONSTANT="; CONSTANT
   LPRINT "H="; H
   LPRINT "LH,L1,LV,k,ALFA="; LH; L1; LV; k; AL * 180 / PI
   LPRINT "PERMISSIBLE  STRESS MPa="; ps
 LPRINT "FORCE CAPACITY TONS="; FC / 10000
 LPRINT "STRESS  MPa "
 LPRINT "TENSION="; FC * SIN(AL) / A
 LPRINT "BEND="; FC * COS(AL) * LH / W
 LPRINT "SHEAR="; FC * COS(AL) / A
  LPRINT "WELD LEG-K ="; k
```

```
CONSTANT= 4.4105E-04
H= 370
LH,L1,LV,k,ALFA= 139.6841  70.31593  90  8  38
PERMISSIBLE  STRESS MPa= 42
FORCE CAPACITY TONS= 9.522729

CONSTANT= 4.4105E-04
H= 370
LH,L1,LV,k,ALFA= 139.6841  70.31593  90  8  38
PERMISSIBLE  STRESS MPa= 63.5
FORCE CAPACITY TONS= 14.39746

CONSTANT= 4.4105E-04
H= 370
LH,L1,LV,k,ALFA= 139.6841  70.31593  90  8  38
PERMISSIBLE  STRESS MPa= 80
FORCE CAPACITY TONS= 18.13853

CONSTANT= 4.4105E-04
H= 370
LH,L1,LV,k,ALFA= 139.6841  70.31593  90  8  38
PERMISSIBLE  STRESS MPa= 100
FORCE CAPACITY TONS= 22.67316

CONSTANT= 4.4105E-04
H= 370
LH,L1,LV,k,ALFA= 139.6841  70.31593  90  8  38
PERMISSIBLE  STRESS MPa= 240
FORCE CAPACITY TONS= 54.41559
```

Fig. 16 B

$$F_1 = F \cdot \cos\alpha = F \,.\, \cos 38^\circ ,$$
$$F_2 = F \cdot \sin\alpha = F \cdot \sin 38^\circ.$$

Total stresses τ equal

$$\tau = \sqrt{(\tau_1 + \tau_3)^2 + \tau_2^2} \;.$$

By solving this equation we obtain the relationship between τ and F. A computer program was drawn up and the results are presented below in Table 3.

The allowable stresses are:

1. $\tau = 42$ N/mm² - Weld joints operating under alternating loads in practice.
2. $\tau = 63.5$ N/mm² - As presented by Swedish experts.
3. $\tau = 80$ N/mm² -The requirements of the majority of the classification societies.
4. $\tau = 240$ N/mm² - The ultimate strength of the steel St 37 to shear ($\tau_u = 0.6 \cdot \sigma_u$).

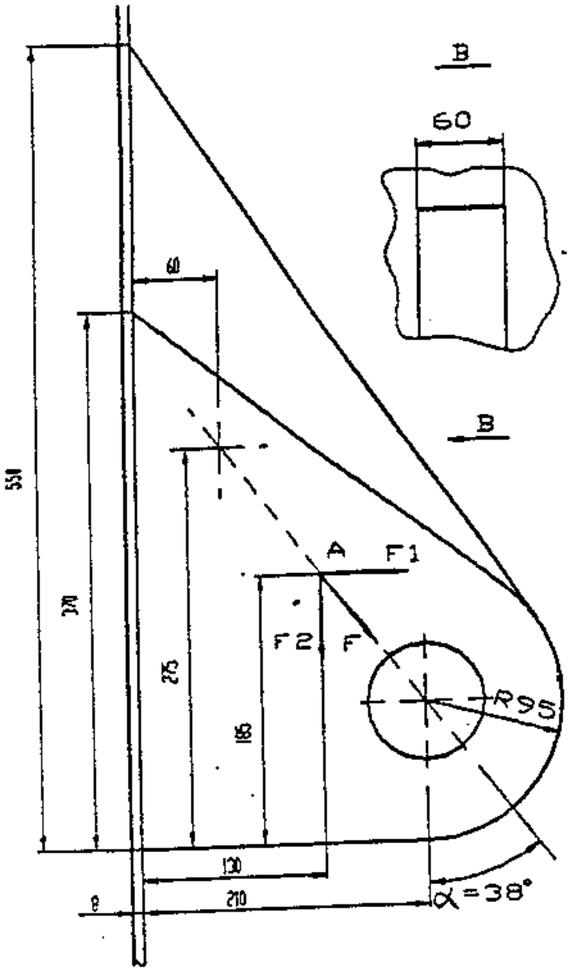
The load-carrying capacity of the side lock and the distribution of stresses in the weld joint is presented in Table 3.

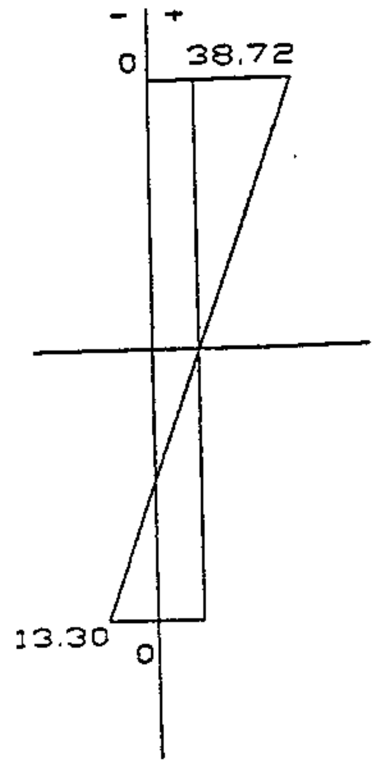
Table 3

		Calculational shear stresses N/mm²				
		42	63.5	80	100	240
Load-carrying capacity of the side lock kN		95.2	144.0	181	227	544
The stresses in the weld joints of the side lock N/mm²	Tension and Pressure	12.2	18.4	23.2	29.0	69.6
	Bending	26.8	40.6	51.1	63.9	153.3
	Shear	15.6	23.6	29.7	37.1	89.0
The relation between the calculational load and the load carrying capacity K		10.5	6.9	5.5	4.4	1.8

The factor K shows the underdimensioning of the side lock.

SIDE LOCK





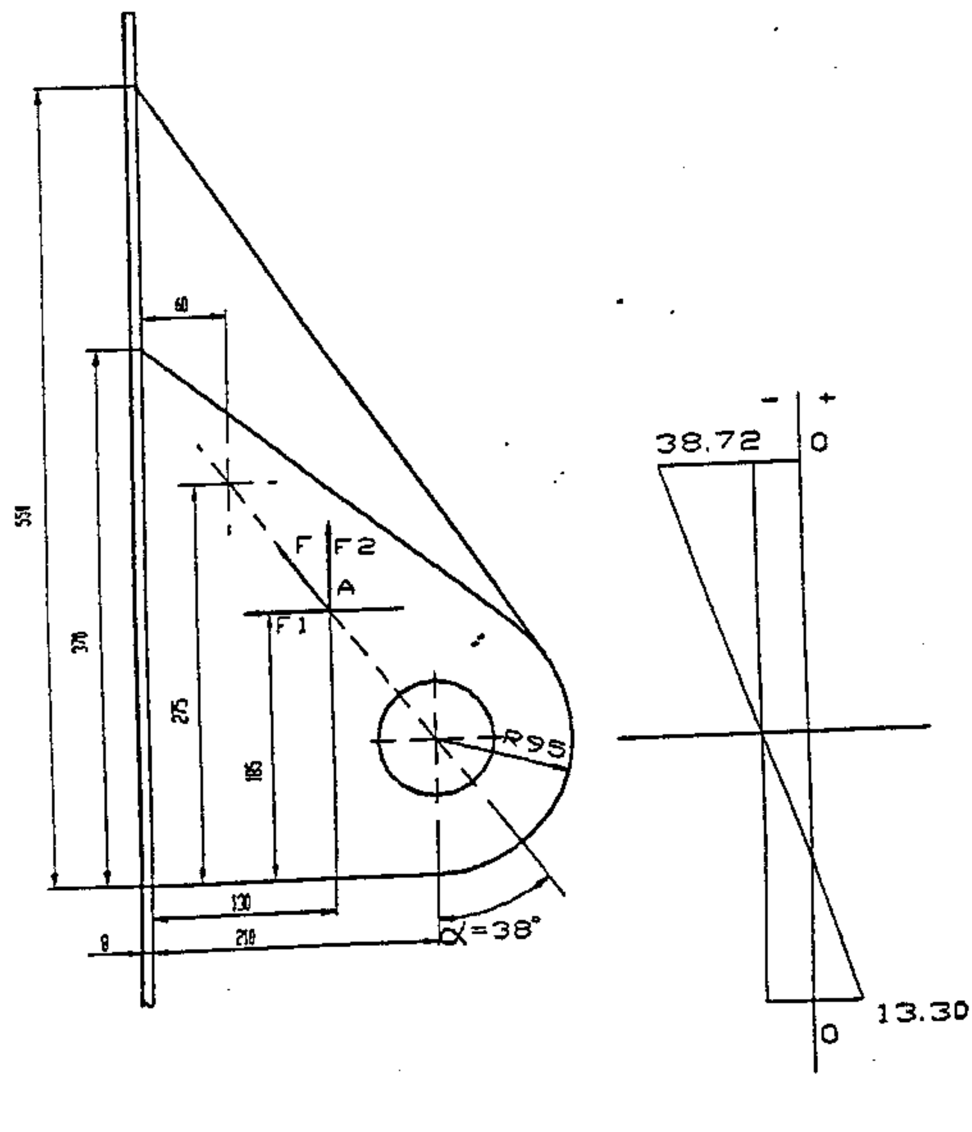
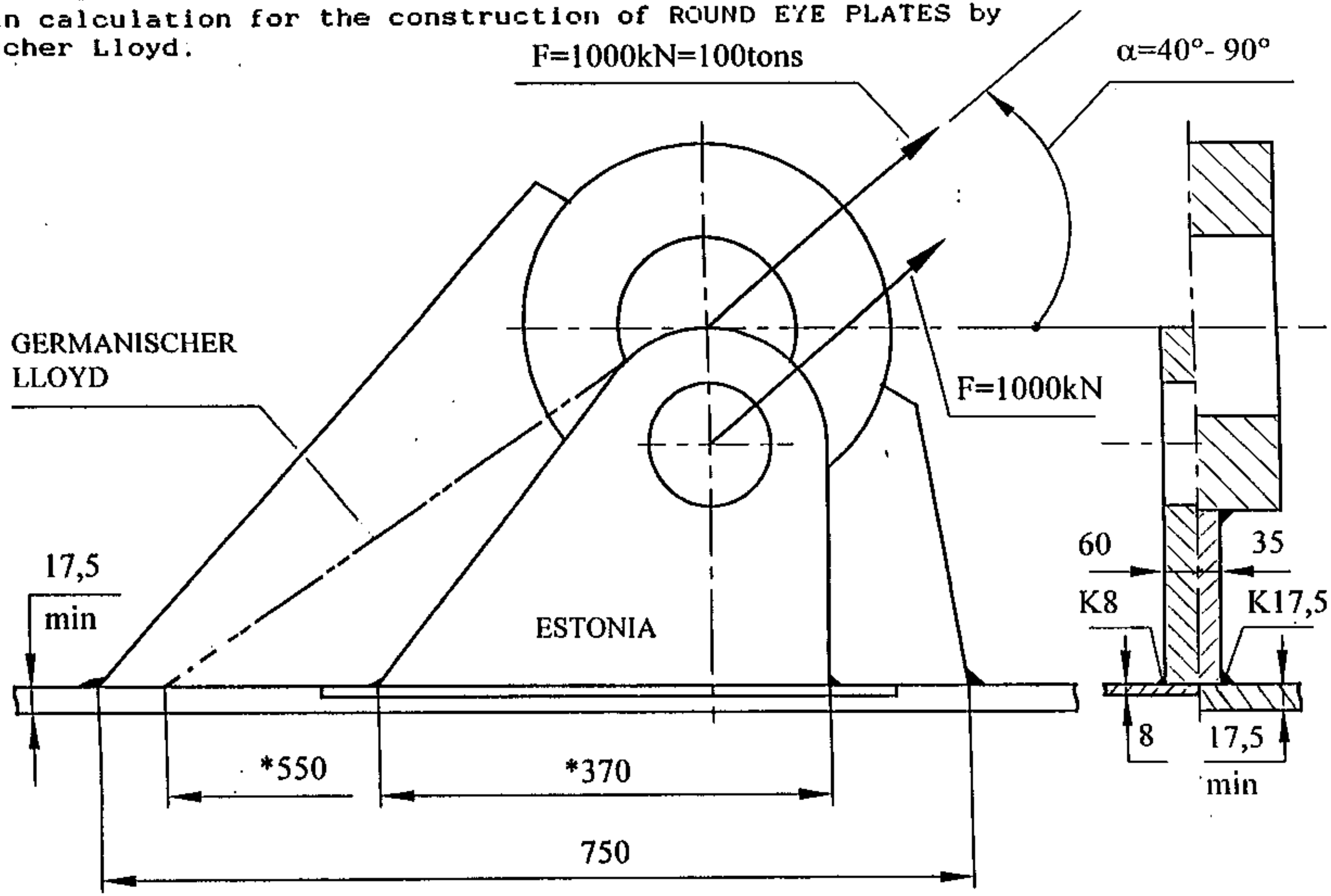


Fig.17

The larger construction is ROUND EYE PLATES from table 25 with nominal size 100 - permissible load 1000 kN.["Regulations for the Constructions and Survey of Lifting Appliances". 1992 Edition. Published by: Germanischer Lloyd.] The smaller construction is of a side lock of a visor of the MV ESTONIA. Both of the constructions are similar and have the same safety working (calculated) load capacity SWL=F=1000 kN (100 tons).
I have made strength calculations on both of the constructions. SWL for a side lock of a visor of the MV ESTONIA is 4.6 times less than calculation for the construction of ROUND EYE PLATES by Germanischer Lloyd.



* The actual dimension of a side lock of the M/V Estonia is 370, and the same dimension of the assembly drawing is 550

Fig. 18

```
REM     QBASIC A.INGERMA  20.06.1996
     pi = 3.1416


AL = 90 * pi / 180

k = 17.5
 H = 750

B = 35
LV = H / 2 - 237
L1 = LV * SIN(AL) / COS(AL)
LH = 300 - L1


ps = 80
A = 2 * .7 * k * (B + H)
 I = ((B + 1.4 * k) * (H + 1.4 * k) ^ 3) / 12 - (B * H ^ 3) / 12
 W = I / (H / 2 + k)
 CONSTANT = ((SIN(AL) / A + COS(AL) * LH / W) ^ 2 + (COS(AL) / A) ^ 2)^
 FC = ps / CONSTANT
 PRINT "LH,L1,LV,=";  LH; L1; LV
 PRINT "CONSTANT="; CONSTANT
 PRINT "FORCE CAPACITY TONS="; FC / 10000
 LPRINT "CONSTANT="; CONSTANT
   LPRINT "H="; H
   LPRINT "LH,L1,LV,k,ALFA="; LH; L1; LV; k; AL * 180 / pi
   LPRINT "PERMISSIBLE  STRESS MPa="; ps
 LPRINT "FORCE CAPACITY TONS="; FC / 10000
```

```
CONSTANT= 9.390068E-05
H= 750
LH,L1,LV,k,ALFA= 184.2039  115.7961  138  17.5  40
PERMISSIBLE  STRESS MPa= 80
FORCE CAPACITY TONS= 85.1964
```

```
CONSTANT= 9.390068E-05
H= 750
LH,L1,LV,k,ALFA= 184.2039  115.7961  138  17.5  40
PERMISSIBLE  STRESS MPa= 100
FORCE CAPACITY TONS= 106.4955
```

```
CONSTANT= 9.390068E-05
H= 750
LH,L1,LV,k,ALFA= 184.2039  115.7961  138  17.5  40
PERMISSIBLE  STRESS MPa= 95
FORCE CAPACITY TONS= 101.1707
```

Fig. 19

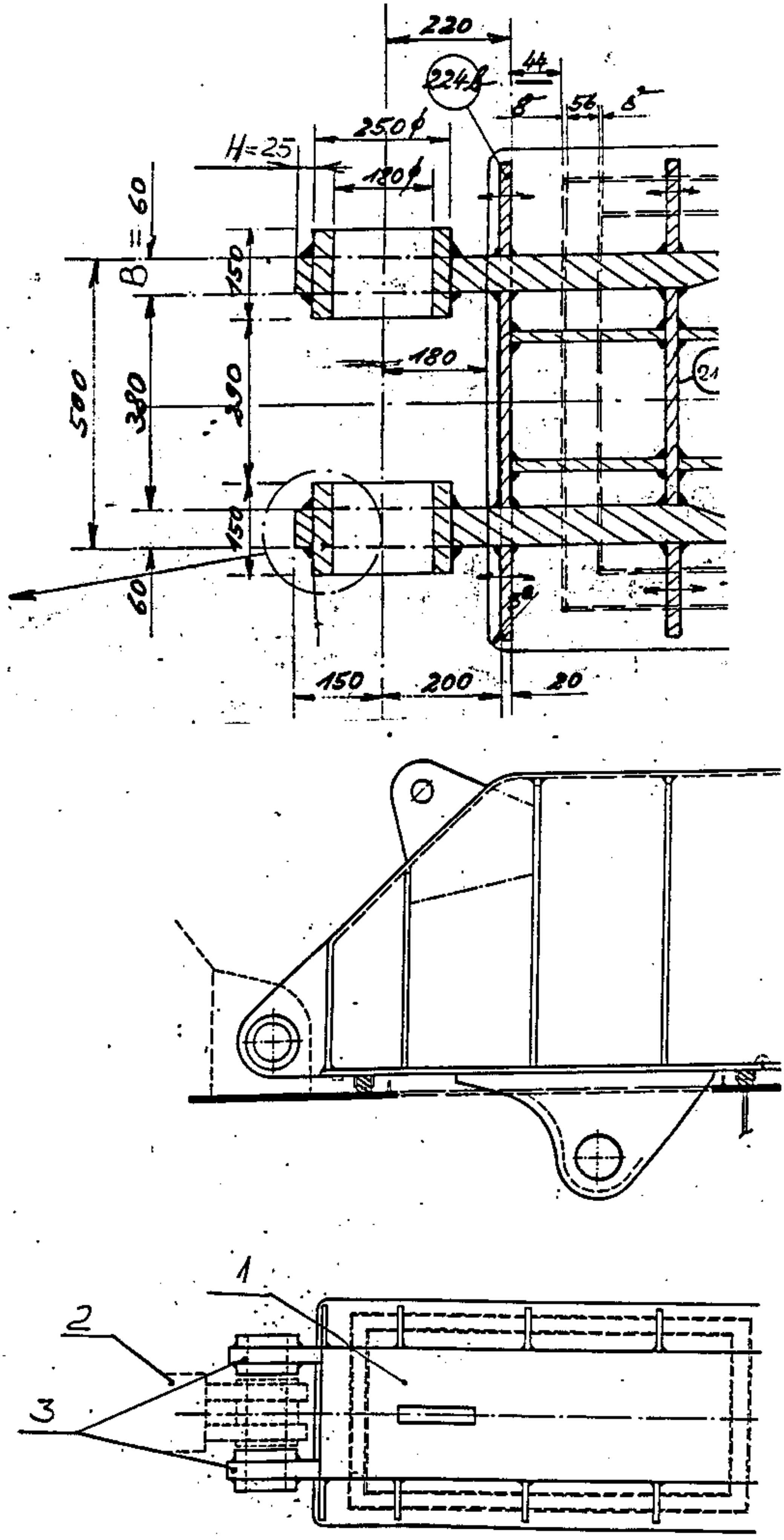


Fig. 20

Hence the side lock is underdimensioned by a factor of 10.5÷7.1 using calculational allowable stresses and by a factor of 1.8 using failure stresses. When conducting an analyses on the stress situation during tension (opening of the bow visor) and pressure (the closing direction of the bow visor) it becomes apparent that the upper edge of the lug is always under heavier load (Fig. 17). Hence the failure initiates at the upper edge. This is further confirmed by the damages to the side locks of DIANA II on Jan. 16th, 1993, Figs 5 and 6.
The underdimensioning of the side lock is also proven by the analogous construction presented in [11]. At the permissible load of 1000 kN (100 tons) of this round eye plate and at the similar calculational load carrying capacity of the side lock of MV ESTONIA the devices of MV ESTONIA have been underdimensioned by a factor of 4.6 at allowable shear stresses of τ = 95 N/mm^2 and by a factor of 10.4 at allowable shear stresses of τ = 42 N/mm^2. This proves again that the side lock was underdimensioned by a factor of 4÷10.
Both constructions are shown in Fig. 18. The strength calculations for the lug are presented, which made it possible to derive the allowable stresses used at design (Fig. 19).

Conclusion.

1. The calculational load carrying capacity of the side locks by allowable stresses is 4÷10 times too small.
2. The calculational load carrying capacity according to breaking stresses is underdimensioned by a factor of 2.

4.4 Strength calculations of the hinges

4.4.1. The design and failure of the hinges

The two beams 1 on the deck of the visor extended about 3 meters aft of the aft edge of the visor deck (Fig. 20).
A heavy steel bushing was welded into a hole in each of the two side plates 3 of each beam. The bushings had a bore, carrying a bronze bushing. The deck part of the hinge consisted of two lugs 2 welded to the deck, carrying between them a steel housing.
The hinges fail in two stages. The reason is that the holes in the side plating are not mechanically (accurately) elaborated but are made by flame cutting. Hence the bushings welded into these holes have significant

clearances and thus the weld joint fails in the first stage due to fatigue. Consequently the load is distributed over to the lugs of the side plating (the second stage) and these break due to overloads at a small quantity of cycles. This is the final failure of the hinges. As the failure takes place in two stages, the sum of breaking loads of the weld and the lugs cannot be considered to be the total load carrying capacity of the hinges.

4.4.2 The strength calculations of the welds

The bushing is welded into the side plates with a diameter of 250 mm and the weld has a leg of 10 mm. Weld joints are calculated to shear stress τ.

$$\tau = \frac{F}{A} \quad ; \quad F = \tau \cdot A \ .$$

A - the failure surface of the weld joint, consisting of four circular welds.

$$A = 4 \cdot \pi d \cdot 0{,}7K = 4 \cdot \pi \cdot 250 \cdot 0.7 \cdot 10 = 21991 mm^2 \approx 22000 \ \mathrm{mm}^2 \ .$$

Table 4 shows the calculational load carrying capacity and failure strengths (τ = 240 N/mm^2) of the weld joints of the hinges without any cracks. Actually the weld joints had cracks [2] and as the stress concentration factor of cracks is 3÷50 [12], the actual failure loads of the welds of the hinges could be 3÷50 times smaller. It can also be observed that weld joints without cracks are underdimensioned only at shear stresses of τ = 42 N/mm^2

Table 4

τ , N/mm^2	F , kN	K=1000/F
42	923.6	1.08
63.5	1396.4	0.71
80	1759.3	0.57
100	2199.1	0.45
240	5277.8	0.018

4.4.3 Strength calculations of the lugs of the side plates

The failure surface of the lugs of the two support beams is calculated using the width and height of the legs(B = 60 mm, H = 25 mm).
One lug has two failure surfaces, hence

$$A = 2 \cdot 2 \cdot B \cdot H = 4 \cdot 60 \cdot 25 = 6000 \text{ mm}^2 \ .$$

The failure takes place at low-numbered cyclic loads and based on the analysis of the failure surfaces the following calculations can be considered to be static failure due to tension stresses. It should be kept in mind, however, that the hole in the side plating was made by flame cutting and the cut surface had many stress concentrators, which are in this case not taken into consideration.

The strength calculation to tension stresses is

$$\sigma = \frac{F}{A} \quad ; \quad F = \sigma \cdot A \ .$$

The calculational load carrying capacity using allowable stresses of $\sigma = 123$ N/mm^2 is

$$F = 120 \cdot 6000 = 720000 \text{ N} = 720 \text{ kN} \ .$$

The load carrying capacity at failure stresses of $\sigma = 400$ N/mm^2 equals

$$F = 400 \cdot 6000 = 2400000 \text{ N} = 2400 \text{ kN} \ .$$

Conclusions

1. The failure of the hinges takes place in two phases. First, the failure of the welds and then the failure of the lugs of the side plates.
2. As cracks formed in the hinge during operation, the actual failure strength can not be estimated to be over 2400 kN, more likely even less due to the stress concentrators in the holes in the side plates made by flame cutting.

5. GENERAL ASSESSMENT OF THE DESIGN AND MANUFACTURING OF THE LOCKING DEVICES.

If the locking devices had been properly designed, manufactured and inspected in accordance with current requirements, would the accident of MV Estonia have been likely to take place?

First, let us define the term "properly designed and manufactured".

1. The calculational loads of 1000 kN per locking device should have been taken into consideration by using a safety factor of at least 2.5(5.0).
2. All the locking devices should have been under equal loads simultaneously (clearance free locks), i.e. forced locking (a power screw and a nut)

Let us determine the resistance moments of the Atlantic lock and the side locks relative to the axis of the hinges in the opening direction of the bow visor. The maximum opening moment on the bow visor caused by the sea load has been estimated at Mm=29000÷35400 kNm by calculations and experimnents.

Which of the moments is greater — the resistance momentum or the sea load momentum? The answers are presented in Table 5.

Table 5

Name of the part	Arm from hinge axis, m	Calculational force Fa, kN	Moment from hinge axis Ma, kNm	Failure force Fp, kN	Moment from hinge axis Mp, kNm
The Atlantic lock	6.87	1000	6870	2500	17175
The side lock (2 pcs.)	4.45	2·1000	8900	2·2500	22250
Gravity of the bow visor	4.9	600	2940	600	2940
	Total	-	18710	-	42395

The calculational resistance moment or Safety Working Load (SWL) is Ma=18710 kNm, which is less then the opening moment caused by the sea load Mm=35400 kNm, which is in turn smaller than the combined resistance/failure momentum Mp=42395 kNm.

$$Ma=17450 < Mm=35400 < Mp=42395 \text{ kNm}$$

When designing as critical a component as the bow visor it is necessary and reasonable to use a safety factor of 5. In that case we get

$$Ma=17450 < Mm=35400 < Mp=78850 \text{ kNm}$$

Thus it can be stated with great probability that the Safety Working Load of 1000 kN is sufficient for locking devices. Thus, once more, the mistake was made in the engineering/manufacturing of the devices. It is not acceptable to use the fact that the design was a traditional one as an excuse. Every engineer/manufacturer is responsible for his work.

6. FINAL CONCLUSIONS

1. As the sea forces are varying in direction and magnitude during operation, the only proper method of calculation is to fatigue strength and only locking devices that are simultaneously and equally loaded should be used i.e. clearance free forced locking.
2. The allowable stresses must take into consideration the imprecise methods of determining the loads and maximum safety factors should be used in determining the allowable stresses.
3. As welded structures are especially sensitive to alternating loads, the allowable shear stresses should not be greater than $40\div60$ N/mm^2 for steels with ultimate stresses of up to 450 N/ mm^2. This value is currently used world-wide.
4. Explaining the underdimensioning of the locking devices with the fact that there were no Bureau Veritas rules for calculations and allowable stresses is no justification for the engineer.
5. The Atlantic lock and the side locks were underdimensioned and were the first to fail. Then the weld joints of the hinges and finally the lugs of the hinge side plates failed. As the Atlantic lock and the side locks were underdimensioned and the holes in the side plates of the hinges were imprecisely manufactured, not securing the maximum load carrying capacity, the failure of the locking devices was inevitable already at the stage of engineering.

Tallinn, April 10th, 1997

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