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Rahka Klaus:

MV Estonia. Visor Damage and Visor Attachment Strenght Investigations at VTT.

Report VALB243.

VTT Manufacturing Technology.

Espoo 15.9.1997.

VTT MANUFACTURING TECHNOLOGY

MV Estonia
Visor Damage and
Visor Attachment Strength
Investigations at VTT

Report VALB243

Klaus Rahka

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VTT

VTT MANUFACTURING TECHNOLOGY

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Summary
Fractures of the MV ESTONIA visor attachments have been investigated and the strengths of individual attachments were estimated based upon testing full and sub scale models. A simplistic system using rules of statics was defined to estimate load levels at individual attachments as caused by wave loading according to model test results obtained elsewhere (the SSPA-Gothenburg tests).

Date Espoo 15.9.1997

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Foreword

The investigations reported in this paper have been performed as part of research carried out to clarify the causes of the sinking of MV ESTONIA in late September 1994. The hypothesis of the casualty to have been initiated primarily by failure of the bow visor attachments due to their structural weakness compared to wave loads to the bow has been investigated by observations and material identifications of authentic parts as well as modelling by calculations as supported by model testing.

The efforts reported have been defined in the course of work of the Joint Accident Investigation Commission of Estonia, Finland and Sweden to which the writer was assigned as an expert of material and structural integrity issues.

I want to thank all experts and Commission members for their contributions to quantify the level of load carrying capacity of the bow visor attachments of MV ESTONIA. It is hoped that some untraditional solutions presented in this report will meet interest and provide some contributions to discussion among experts and engineers in the trade and elsewhere.

Espoo, August 1997

Author

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1 Introduction

The MV Estonia sank during the early hours of 28 September 1994 in heavy weather in the northern Baltic Sea. All visor attachments were found to have broken and separation of the visor from the ship had occurred. Opening of the second water barrier of the bow, i.e. the bow loading ramp had followed as caused by interference with the visor leading to water entry to the automobile deck leading to considerable list. Capsizing of the vessel had resulted when upper decks were also flooded when doors and windows higher up had allowed water to enter.

Here we report on observations of damage, structural features and strength investigations of locking and lifting cylinder visor attachments. Some investigations of hinges have also been carried out.

The investigations include an approach to assessing visor attachment loading levels when the visor is loaded as measured during model experiments. Characterisation of fractures and deformations of authentic visor locking element related parts from the MV ESTONIA and model tests. Some relevant visor locking related structural elements from the near sister ship MV Diana II have also been ivestigated. Calculations were formulated and some fitted parameters used to analyse test results regarding attachment strength in order to obtain realistic estimates of actual attachments slightly deviating from the models. Side lock visor lug sites and the hinge bushing installation were also modelled by FEM (Finite Element Method) in other part-projects (VTT Technical Report VALC-246, Kaj Katajamäki and VALC-312/8.4.1997, Jukka Airaksinen), the results of which are also used in the estimation of the actual side locking strength. Bottom lock mock-up test results obtained by the Technical University of Hamburg were used together with hardness measurements of the authentic pieces for assessing the structural strength of that lock.

2 General information about performed investigations

Fracture morphologies of the MV ESTONIA visor attachments suggest that static type of ultimate strength is the practically relevant characteristic according to which release of the visor had occurred. Therefore - but not excluding e.g. the closely related low cycle fatigue requiring loads exceeding cross section yield limits - the investigations reported here have focused on determination of a realistic upper bound strength of the visor attachment system. Individual strength estimates are all based upon testing either complete full or sub size models (lockings) or testing parts of authentic components (hinge weldments and platings). All strength experiments performed have been done under monotonically increasing load up to failure of respective test piece and calculation parameters have been introduced to "calibrate" the calculation models accordingly. Features of the fractures as found have been examined on a minimum principle not striving for an exhaustive investigation. Specimens have been stored for availability to further interests. Wave induced loads as measured at a single support of a model visor have been recalculated to identify load levels at five primary attachment locations. It is believed that the extent of the investigation is sufficient for the conclusions that it has been fully possible for the visor to have broken loose while the ship's bow has submerged into a wave to a depth less than needed to immerse its weather deck into green water.

3 Geometrical features, visor attachment loading and visor detachment scenarios

To help assessing a realistic way that the visor could have been detached from the bow of MV ESTONIA, a summary of the geometry of the visor and its attachment locations is presented. Load distribution onto the various attachments was estimated using simplistic representations of moment and force equilibrium and experimental information on wave induced lifting, pushing and sideways translating loads positioned to cause the resulting and measured opening, yawing and twisting moments that act to rotate the visor around its horizontal transverse, vertical and horizontal longitudinal axes. The shape of the visor and its attachment configuration were not accurately accounted for. Instead this shape effect is included in the parametric study that an assumed variation of the wave load centre of action implies.

3.1 Visor geometry

With reference to Figure 1 - using coordinates adopted by e.g. SSPA for reporting the wave load model tests with the origin about 8.5 m above still water level in the centre between the visor hinges -

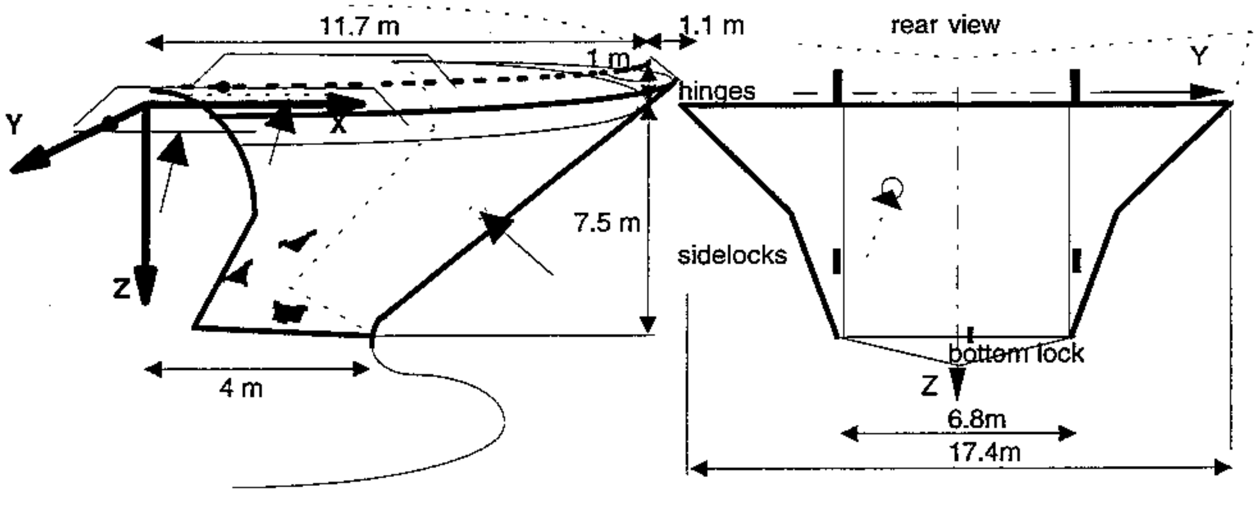


Figure 1. Bow of MV ESTONIA with visor and associated lockings.

the stem of the visor stretches approximately from X = 4 metres at Z=7.5 below the hinge axis or 1 metre above the still water level to X = 11 metres at -Z=1 metre above the hinge axis level. The largest width of the visor was Y = ±9 metres at X = 2.85 metres forward of the hinge axis at its upper aft edge at weather deck (Z = 0 metres at the hinge axis) and at the low end of the aft plating Y = ±3.4 metres at X = 1 metre and Z = 6.5 metres or 2 m from the still water level. At Z = -2 or 2 meters above the weather deck, a bulwark about +1,5 m high in its aft end and 1 m high further forward rose above the hinge level. The bulwark is more vertical than the visor shell plating in its aft parts. In the plane normal to the stem and containing the hinge axis, the visor shell plating curvature is approximately a circle with a radius of 7 metres and leaning 45 degrees forwards. In this plane through the hinge axis the aft edges of the visor shell plating are at X = Z = 2.85 metres at their extreme at about Y = ±6 metres but the aft edges of the visor are drawn further back both below and above this location. The stem leans about 45° forwards. The flare of the visor shell plating is about 45°, except on the lower aft sides, where the side shell plating is somewhat steeper due to the passage to the car deck. Higher up the flare is 45° also here. The flare in the lower extreme is wider than 45 degrees Attachment coordinates are given in Table 1.

Table 1. Locations of visor attachments. Ref to Fig 1.

	X	Y	Z
Hinges	0	±3.4	0
Side locks	1.15	±3.4	4.3
Bottom lock	1.5	0.4	6.7
Lifting cylinder to hinge beam	1.3	±3.4	0

3.2 Forces and moments acting on the bow visor and their theoretical relationship

Forces and moments acting on the visor have been measured in straight head and oblique bow seas at the SSPA model test basins. Simulation for estimation of the lifting load for head seas has also been performed (work performed at VTT-Manufacturing technology). Measurement results were given as a set of force and moment components acting in a point centred between the hinges.

The force components are the longitudinal Fx (head load, positive forwards), transverse Fy (positive to starboard) and vertical Fz (positive down) of the resultant of the acting bow load. The components of the acting moment are the opening, yawing and twisting moments (My, Mz and Mx) about the horizontal transverse hinge axis (My about Y coordinate-axis) and vertical and horizontal longitudinal "axes" through the hinge centre (yawing Mz about Z-axis and twisting Mx about X-axis). These had been obtained in basin tests using a model of MV ESTONIA. The model visor had been supported on the model vessel by one single attachment composed by force and moment transducers.

The position of the line of the resultant of wave load action - giving one condition of the hydrodynamic centre of the visor - (X;Y;Z) is algebraically defined as follows:

Mx = Fz*Y - Fy*Z

My = Fz*X - Fx*Z

Mz = Fy*X - Fx*Y

From Mx; Z=Y*(Fz/Fy)-Mx/Fy,
From Mz; X=Y*(Fx/Fy)+Mz/Fy , which yield
My = (Fz/Fy)*Mz+Y*(Fx*Fz/Fy)-Y*(Fx*Fz/Fy)+(Fx/Fy)*Mx, that is

My*Fy = Mx*Fx + Mz*Fz

The three equations combine into the single condition as given in the box tying force and moment components together.

Measurements reported indicate that Fx is about as large as Fz. Fy is up to about 1/3 of Fx. Thus (Mz+Mx) is ≈ My/3. In (straight) head sea the load is symmetric at port and starboard causing Fy and Y to become 0. The twisting and yawing moments Mx and Mz and Fy thus occur only in oblique bow sea. With the above information the position of the load action line is unambiguously defined. However, the actual position of the wave centre of action on this line - the hydrodynamic centre of the visor or the value of (X;Y;Z) containing the information of visor shape - lacks accurate definition. Therefore, a parametric presentation of attachment reactions has been developed where the varied parameter is the X-coordinate of the wave centre of action, the origin (X = Y = Z = 0) being on the hinge axis midway between the hinges. Y and Z are defined by the measured ratios of force components and moments, i.e. the Fy/Fres-ratio and the ratio of Fx/Fz and also Mz/My. Mx follows from the others as accounted for above.

3.3 Measured values of force and moment components

Force and moment components obtained from model tests as performed by SSPA (Gothenburg, Sweden) are shown in Figure 2 together with a best estimate equation describing their relationship. Here the force is Fzx=Fx*1.414=Fz*1.414. The mean trend and its standard deviation are given in the graph. The fitted equation is by the present author for use only within the range of given data.

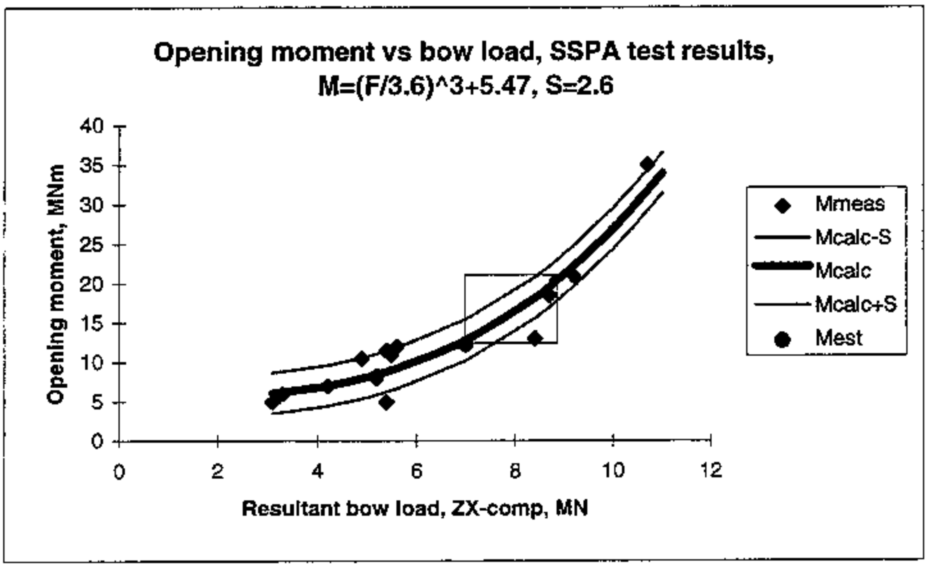


*Figure 2. The interrelation of the measured bow force and opening moment. Estimated critical range 7 to 9 MN indicated. Force is component Fzx in ship's longitudinal plane, Fzx=Fx*1.414=Fz*1.414.*

The experimental and sufficient (ch 3.2) independent relations regarding the basic force and moment components as used in the following are

Fz/Fx = 1 and Mz/My = 0.1 and Fy/Fres = 0.23.

3.4 Distribution of bow load onto visor lockings

There were five "engaged" attachments for the visor and additionally two lifting hydraulic cylinder systems connecting the visor to the hull. Two hinges supported the upper aft, two side locks the lower sides and one bottom lock was installed to secure the visor bottom in a location close to the aft centre of the visor lower extreme.

In calculating the reaction loads at the individual five actual attachment sites basic principles of statics have been employed. Thus the positions of the attachments and an assumed bow load action centre are input parameters in the analysis. A distance related part of structural stiffness of the visor is included in the reaction estimation. Due to the high number of attachment points the visor attachment system is statically undetermined and some load or moment sharing need to be taken as parameters. Here it turns out that only the opening moment needs to be shared by an assumed portion by the side locks and the bottom lock. The twisting flexibility may influence the relative reactions at the two side locks on the one hand and the hinges in carrying the net effect of the transverse component Fy of the bow load. Based upon visor and seal stiffness measurement results the stiffnesses of the individual attachment sites are taken to be much higher than the global stiffness of the visor, such that no evening of reactions (actually shift of load from side locks to hinges in order to maintain global balance) actually occurs between the two visor sides or upper and lower parts.

Load reactions were assigned to each attachment according to their relative distances from each other and the load action centre. This gave a system directly in balance both as to their direct sum as well as the reactive moments. The reactions are therefore determined only by the relative positions of the action centre with respect to the positions of the attachments. The calculations have been performed for a positive opening moment only, idealising the visor as a pivoting beam structure as illustrated in Figure 3.

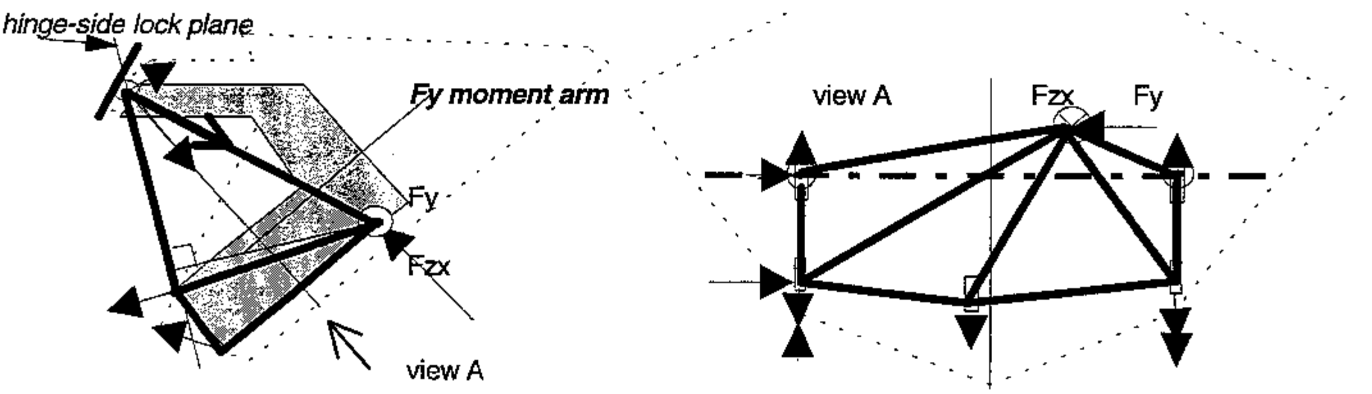


Figure 3. Side view of visor idealization as a rocking beam structure for attachment load calculation. Acting load components Fzx and Fy indicated by lettered arrows, some of the reactions by arrows only.

The position of the moment arm for the transverse force component Fy should be noted as this will bear upon how the attachment reactions depend on the position of resultant wave action. As this moment arm does not point to the hinge axis a load action centre independence is not obtained. No part of load is assumed to be taken to deform the visor.

In the force distribution scenario under an opening moment, direct load reaction is taken by the hinges only and opening moment is resisted by pull in the lockings. Push in a lock may be induced by sufficient effect of the sideways transverse load Fy. The total hinge reactions will include locking reactions also as the hinges are the centre of support of the visor structure - effectively a simply supported swing. Visor lock take no direct side reactions as locating horns are provided for this purpose. Reactions at locating horns at the side locking sites will also respond to the twisting and yawing moments and help the hinges to hold the visor from moving sideways. Here we assume that the lockings have some play and are flexible in the vertical direction so as to minimise vertical reaction, passing that onto the hinges. The solution may be an extreme value solution in so far as visor stiffness is not assumed to even out any reactions. The flexibility of the visor is high and suggests no effect of visor stiffness.

Sharing of opening moment reaction between the bottom and side locks is unknown. A base-line calculation has been done for 50/50 sharing of the opening moment by side locks vs bottom lock. The slight starboard location of the bottom lock may influence, and can – as one value shown in the equations – be allocated with 44% of the total according to its sideways location on starboard of the ship's keel ((3.4-0.4)/6.8 = 0.44) leaving 56% (0.56) to the side locks.

Sharing of the side lock reaction to the opening moment between port and starboard is obtained to be proportional to the sideways distance of the bow force action centre or the Y-coordinates of the lockings and the point of bow load action (X,Y,Z). For any oblique bow waves the action centre is obtained to be located Y metres from the ship's centre line to the wave encounter side, leading to sharing in proportions of (3.4-Y)/6.8 and (3.4+Y)/6.8 between starboard and port of the share of the total opening moment left over by reaction at the bottom lock. For port wave encounter pull will intensify on port. Y = Fy/Fx*(X-Xo) is obtained, where Xo = Mz/Fy.

In addition to pulling hold against the opening moment, any position of the action point forward of the hinge axis causes a yawing moment by the transverse Fy-load. The related reaction is a force couple pair acting in ensemble at the side locks and the hinges. The moment arm of Fy is the distance (length of normal) from the assumed action point to the hinge-sidelock plane. This induces an added pulling reaction at the port side lock and a tension relieving reaction at the starboard side lock (if the flexing of the visor and play in the lock admits, this could create even a pushing load at starboard) causing no change in the holding against the opening moment. The port side lock will experience severe pull compared to its strength and will break asking for least load to the bow (as the reader can deduce having noted the strength of the p-side lock reported later). This failure will cause relief of holding against the opening moment at the side locks and thus addition of some extra tensile reaction to the bottom lock will occur as demanded by hold requirement against the opening moment.

If the starboard side lock were not forced into a pushing reaction from the ship due to play in the lock, the opening moment reaction occurring at the bottom lock could also change, but this has not been assessed here. The total reaction in the port side lock may simply become as shown below, Figure 4. Transverse side support is provided by locating horns close to the side locks and by hinges in proportions of (a/(b-a)).

It may be found that wave action centre locations taken to be realistic seem to cause the port side lock to break at a load level lower than that needed to break the next attachment.

Pull in side locks (for stb replace +:s by - in F_{psl} eq):

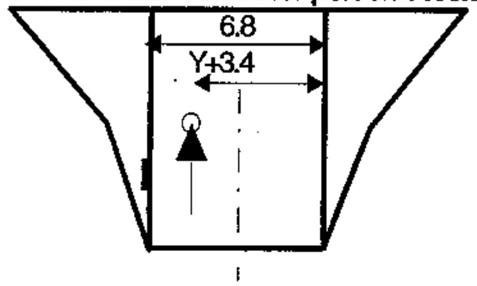
$$M_y = ((Fzx/3.6)^3 + 5.47 + s*2.6);$$

$$F_{psl} = My*0.56*((Y+3.4)/6.8)/4.45$$

$$+ 0.23Fzx* S\ (a/b)/6.8$$

$$-0.15$$

opening moment reaction share for port side lock

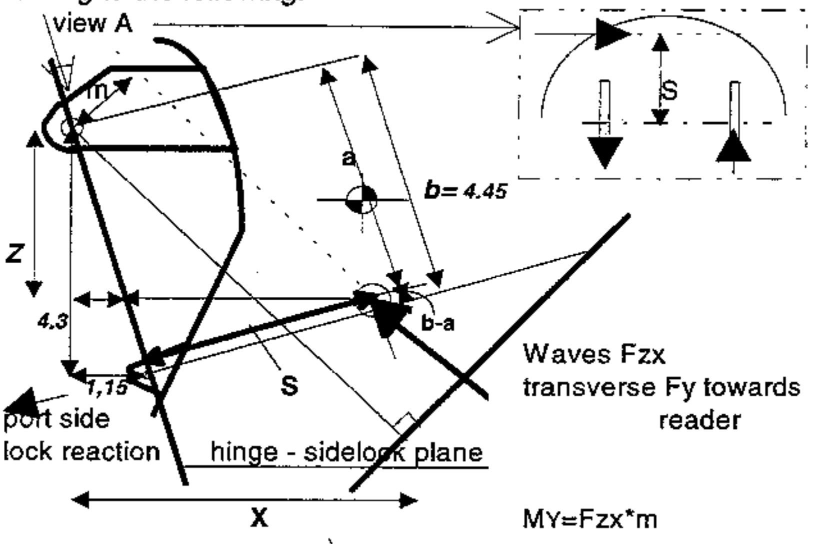


where -0.15 MN is approximately the side lock reaction due to the weight of the visor and the side lock force couple responding to the transverse force Fy = 0.23 Fzx is obtained through calculating:
The moment arm S for Fy is obtained according to the following:

$a = Z*b/4.3 +$
$+(X-Z*1.15/4.3)*1.15/b$

$b = 4.45$

$S = (X-Z*1.15/4.3)*4.3/b$



Lock reactions directed due to local play and local vertical flexibility

Figure 4. P-side lock load and visor dimensions in calculation of port side lock reaction to opening and yawing moments (transverse load component).

The following analysis given in Figure 5 deals with the load in the bottom lock also including this situation.

***The tensile load F_{bl} in the atlantic (bottom) lock** is estimated as:*

$M_y = (Fzx/3.6)^3 + 5.47 + s*2.6$

$F_{bl} = 0.44My/(6.82\sin 60)+$

-0.3

0.3 = visor weight support reaction

If port side lock breaks add: (0.23*Fzx*S*(a/b)/6.8)*4.45/6.82

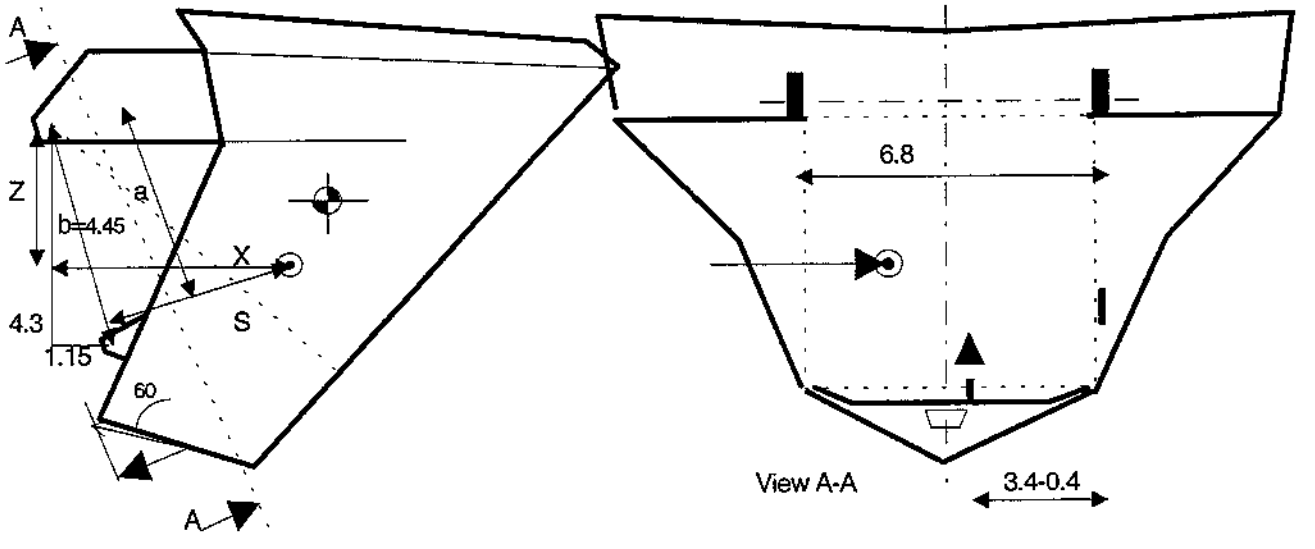


Figure 5. Bottom lock load and visor dimensions in calculation of bottom lock reaction to opening and yawing moments (transverse load component) after p-side lock failure.

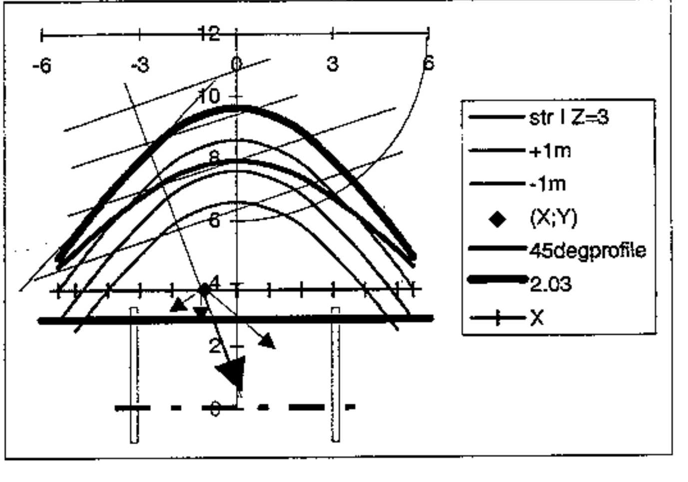


Figure 6. Horizontal sections of the visor in the vicinity of the wave load action centre (X;Y) indicating visor shapes through Z = 0.93 m below hinge axis and at stringer I (Z=3m) and str I ± 1m. Approximate actual visor shape in hinge plane perpendicular to stem also indicated showing also projection of line of wave load resultant for 30° oblique wave.

A schematic presentation of the visor profiles in the region of one assumed point of wave action centre for X = 3.8 metres is shown in Figure 6.

3.5 Loads on hinges

As the loads on the lockings react as aftwards tension to resist opening of the visor, the loads on the hinges will act as forwards and down directed support loads. The external and the lock reactions will thus be supported at the hinges. Thus the load components of the external load and the reactive tension in the bottom lock may be divided between the hinges in proportion to their transverse distances from the respective points of load action. The side lock reactions – as they occur at the same Y-coordinates as the respective hinges will add directly to the hinge on the same side as the respective side lock. In the asymmetric case of the port side lock having broken, some of the twisting and yawing moment will have to be balanced by small additional force couples at the hinges. These will be in the magnitude of 0.03 to 0.2 MN and will affect the hinge load levels only marginally.

3.6 Result examples

Results from visor attachment load calculations performed using the above methodology are presented in the following Figure 7a..

The bow load levels covered are 5, 7 and 9 MN and the mean value of opening moment was used. Calculation results are given for five positions of wave load action identified as "calc points" 1 to 5:

1. a point behind the visor encompassed volume or the only possible location of wave action on the ship's centreplane (Xo,0,Zo).

2. a point on the visor's aft plating (X = 2.85 m)

3. a point forwards of the visor's aft plating which yields the load on the sb side lock to become 0 from having been tension for the previous location of wave action centre

4. a point on a transverse line through the visor that bissects the horizontal cross section area of the visor

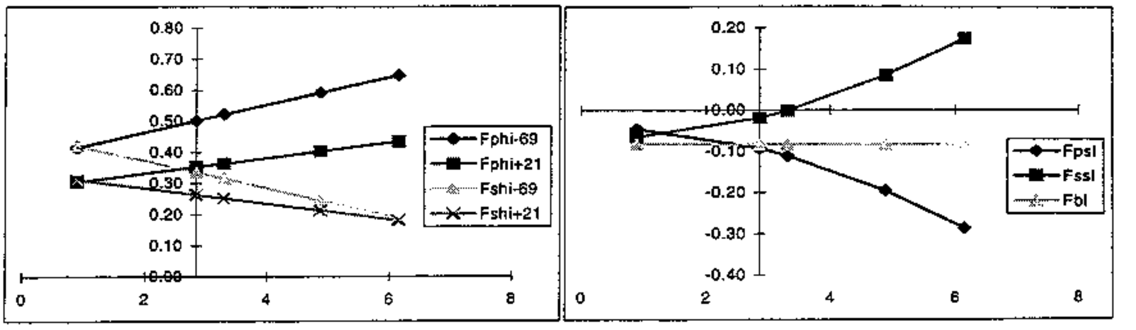
5. a point on the visor shell plating marking the forwardmost point where the line of load action enters the visor volume

The situation after p-side lock failure is displayed in the subsequent Figure 7b.

The calculation results indicate that the reactions at the visor side attachments obtain an increasing value as the point of wave action is assumed to lie more forwards. This sensitivity of the reactions to the position of the bow load action centre is essentially a result of the property of the geometrical layout of the attachment system which sets the condition for the choice of moment arm for the transverse Fy-load component. Had the visor been locked by a centrally positioned bottom lock only, sensitivity of reaction to bow load action centre would not have occurred. This is also evidenced by the b-lock load for the "all intact"-case. The side lock load is most sensitive to the assumed position when all attachments are holding. In port wave encounter the weakest of the locks – the p-side lock – is expected to break first leading to redistribution of reaction mainly to the bottom lock with some relief of load at the hinges.

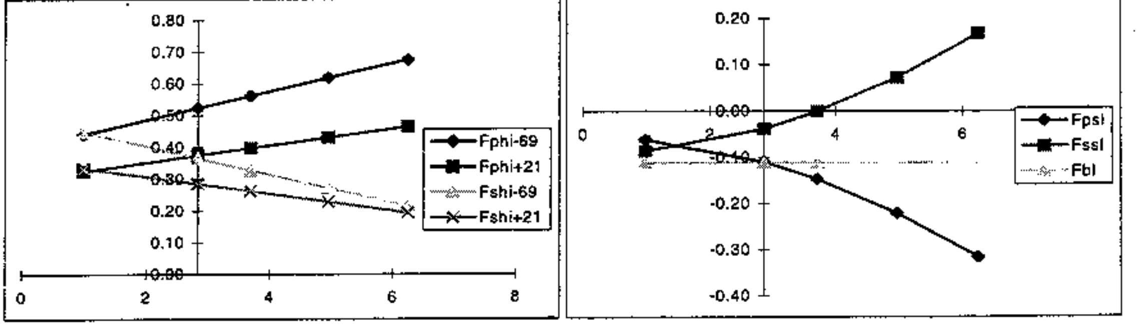
All intact Fres=5.00 My=7.94 Xo=0.898 Fz=3.44 Fractions of Fres

calc point	X	Fphi-69	Fphi+21	Fshi-69	Fshi+21	X	Fpsl	Fssl	Fbl
5	6.150	0.65	0.43	0.19	0.18	6.150	-0.29	0.18	-0.08
4	4.890	0.59	0.40	0.24	0.21	4.890	-0.20	0.08	-0.08
3	3.300	0.52	0.36	0.32	0.25	3.300	-0.11	0.00	-0.08
2	2.850	0.50	0.35	0.33	0.26	2.850	-0.09	-0.02	-0.08
1	0.898	0.42	0.30	0.42	0.31	0.898	-0.05	-0.07	-0.08



All intact Fres=7.00 My=12.25 Xo=0.989 Fz=4.82 Fractions of Fres

calc point	X	Fphi-69	Fphi+21	Fshi-69	Fshi+21	X	Fpsl	Fssl	Fbl
5	6.250	0.68	0.47	0.22	0.20	6.250	-0.32	0.17	-0.11
4	4.970	0.62	0.43	0.27	0.23	4.970	-0.22	0.07	-0.11
3	3.700	0.56	0.40	0.33	0.26	3.700	-0.15	0.00	-0.11
2	2.850	0.52	0.38	0.37	0.29	2.850	-0.11	-0.04	-0.11
1	0.989	0.44	0.33	0.45	0.34	0.989	-0.06	-0.09	-0.11



All intact Fres=9.00 My=19.87 Xo=1.248 Fz=6.19 Fractions of Fres

calc point	X	Fphi-69	Fphi+21	Fshi-69	Fshi+21	X	Fpsl	Fssl	Fbl
5	6.550	0.70	0.53	0.24	0.23	6.550	-0.37	0.16	-0.16
4	5.190	0.64	0.49	0.30	0.27	5.190	-0.26	0.05	-0.16
3	4.430	0.61	0.46	0.34	0.29	4.430	-0.21	0.00	-0.16
2	2.850	0.54	0.42	0.41	0.34	2.850	-0.13	-0.08	-0.16
1	1.248	0.47	0.37	0.48	0.38	1.248	-0.09	-0.12	-0.16

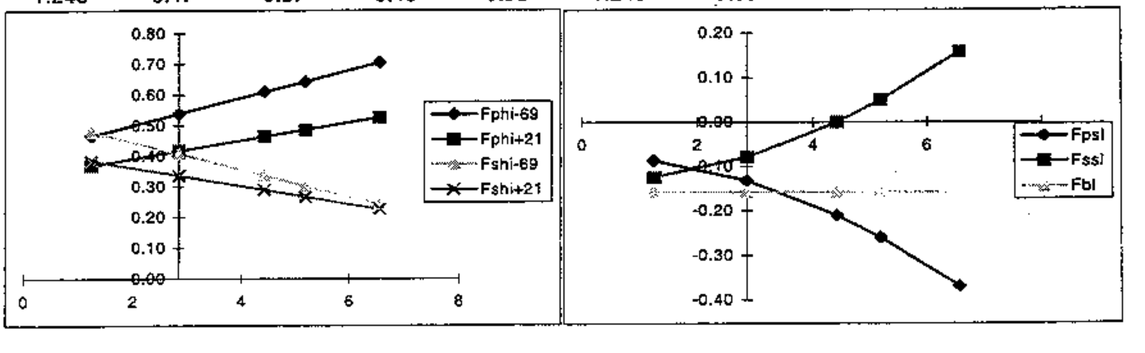
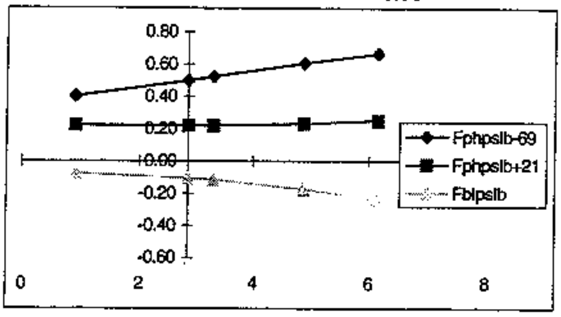
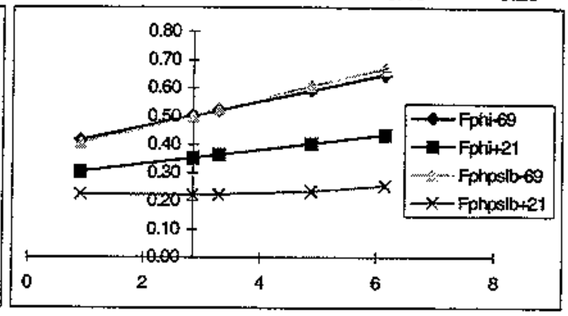


Figure 7a. Load reactions at visor attachments (left for hinges, right for locks) when all are holding for Fres 5, 7 and 9MN from top to bottom row. Pull from ship is negative, push positive. Fphi-69=load component in weak direction at port hinge, Fphi+21=load component in strong direction at port hinge asf (see also Ch 6.1), Fpsl=load at port side lock asf, Fbl=load at bottom lock. All values fractions of Fres. Horizontal axis is X for wave load centre, vertical axis is load fraction of Fres.

Port side lock broken Fres=5.00 My=7.94 Xo=0.898 Fz=3.44 Fractions of Fres

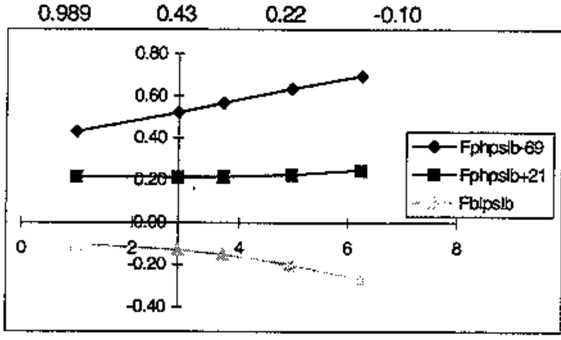
calc point	X	Fphpslb-69	Fphpslb+21	Fblpslb	criticality	X	Fphi-69	Fphi+21	Fphpslb-69	Fphpslb+21
5	6.150	0.67	0.25	-0.24	Hinge	6.150	0.65	0.43	0.67	0.25
4	4.890	0.61	0.23	-0.17	0.86	4.890	0.59	0.40	0.61	0.23
3	3.300	0.52	0.22	-0.11	Bl	3.300	0.52	0.36	0.52	0.22
2	2.850	0.50	0.22	-0.10	0.30	2.850	0.50	0.35	0.50	0.22
1	0.898	0.40	0.23	-0.08		0.898	0.42	0.30	0.40	0.23

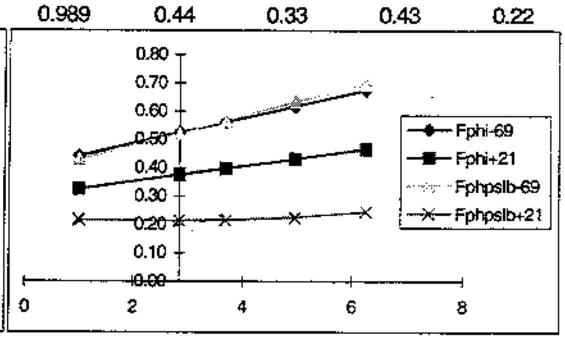




Port side lock broken Fres=7.00 My=12.25 Xo=0.989 Fz=4.82 Fractions of Fres

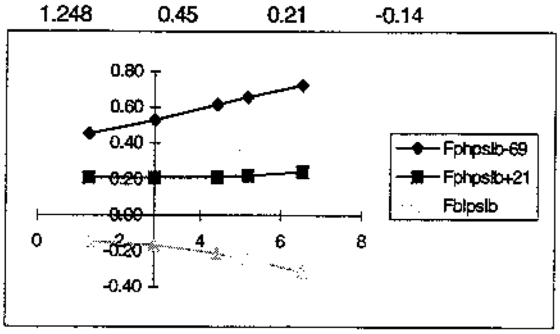
calc point	X	Fphpslb-69	Fphpslb+21	Fblpslb	criticality	X	Fphi-69	Fphi+21	Fphpslb-69	Fphpslb+21
5	6.250	0.70	0.25	-0.27	Hinge	6.250	0.68	0.47	0.70	0.25
4	4.970	0.64	0.23	-0.20	0.61	4.970	0.62	0.43	0.64	0.23
3	3.700	0.57	0.22	-0.15	Bl	3.700	0.56	0.40	0.57	0.22
2	2.850	0.52	0.21	-0.13	0.21	2.850	0.52	0.38	0.52	0.21
1	0.989	0.43	0.22	-0.10		0.989	0.44	0.33	0.43	0.22





Port side lock broken Fres=9.00 My=19.87 Xo=1.248 Fz=6.19 Fractions of Fres

calc point	X	Fphpslb-69	Fphpslb+21	Fblpslb	criticality	X	Fphi-69	Fphi+21	Fphpslb-69	Fphpslb+21
5	6.550	0.73	0.24	-0.32	Hinge	6.550	0.70	0.53	0.73	0.24
4	5.190	0.66	0.22	-0.24	0.48	5.190	0.64	0.49	0.66	0.22
3	4.430	0.62	0.21	-0.21	Bl	4.430	0.61	0.46	0.62	0.21
2	2.850	0.53	0.21	-0.17	0.17	2.850	0.54	0.42	0.53	0.21
1	1.248	0.45	0.21	-0.14		1.248	0.47	0.37	0.45	0.21



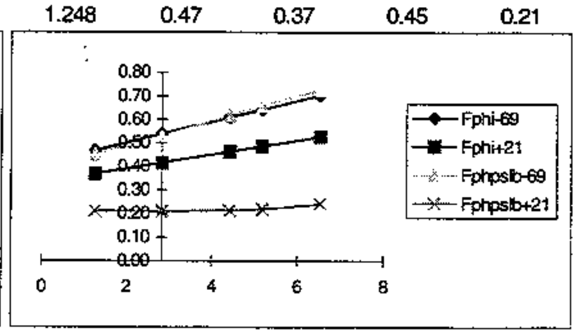


Figure 7b. Load reactions at visor attachments after port side lock failure. Note induced sensitivity of bottom lock reaction to choice of action location.

The bottom lock – according to the chosen logic of assigning reaction loads – was insensitive to the position of wave action centre as long as the wave encounter side lock was holding and the locking arrangement essentially symmetrical. The bottom lock reaction will become sensitive to wave action centre assumption if the side lock breaks because the starboard side lock continues to be "unloaded" by the transverse Fy load being still sensitive to the position of the bow load action centre and holding lost by the p-side lock against the opening moment now has shifted to the bottom lock.

3.7 General conclusion

The above calculation indicates that breaking the side lock at 1.2 MN local reaction (equalling its strength as arrived at below) occurs at a wave load level which may be insufficient to break the next attachment. The side lock may thus break without another attachment failing. This would support the damage pattern that occurred to MV ESTONIA's sister ship DIANA II in January 1993 in the form of partial attachment failure. This involved side lock fracture and hinge damage. The shape of the bottom locking fore peak deck lugs of DIANA II was more robust indicating a stronger design (up to the limit of the visor lug of about 1.8 MN) than that of MV ESTONIA. A recurring judgement about an attachment breaking sequence of the MV ESTONIA accident is, however, not possible. For a direct head wave the bottom lock could reach its breakpoint of 1.5 MN at an estimated bow force of somewhat higher than the values given above before the side lockings became critically loaded.

The outcome of simplistic locking system load sharing analysis yields the possible effect of moving the bow load centre more forwards (an increasingly more forwards protruding bow design) implying an increasing effect of the transverse load to increase attachment reactions and thus to weaken the system strength. The load needed to overcome the strengths of the bow visor attachments is thus sensitive to the shape of the visor, which has not been investigated. This sensitivity intimately follows from the way the moment arm of the transverse load Fy is in position along the normal through the bow load centre to the attachment plane. This seems to be the effect of the visor's shape and attachment configuration - particularly the aft positioned hinges in relation to the lockings.

3.8 Visor detachment scenarios

Visor detachment depends on the individual reactions in relations to the strength levels of the attachment sites. The above presented estimation has suggested that for 30° bow waves a minimum resultant bow force of 7 MN (lifting component around 5MN corresponding to design load level) may be sufficient to raise the pulling load on the port side lock up to 1.2 MN with some cautiously chosen wave action centre. According to work presented below this would be enough to break the side lock in the local load direction found to apply. The loads at the port hinge and the Atlantic lock are still below their breaking capacities. The weakness of the port side lock compared to the starboard side lock has been recognised. The least bow load seems to be needed to cause the port side lock to break first, followed by break of the next attachment – the hinge or the bottom lock – at a somewhat increased level of bow load. It has not been possible to define in great accuracy the strength of the hinges, but approximate evaluation indicates that a hinge may be at risk if the local transverse shearing load reaction component directed down and forwards reaches up to about 4.6 MN. Hinge failure may thus happen second if a total bow load higher than the values given above were combined with a lower (than average) opening moment, which would be insufficient to break the bottom lock next. A combination of a higher bow load and a lower opening moment at a higher water pressure and less deep ingress of the ship (causing less opening moment) could raise the local load at the wave side hinge to the critical level before the lockings. In direct head sea a higher load is needed and then the bottom lock could be at risk first. A low bow load combined with a high opening moment would result in the port side lock becoming critical first, followed by the bottom lock.

4 Structural features and damage to the visor

Some structural features on the visor were found to deviate from the drafted structures on drawings. Such features – which, however, were estimated not to affect the breaking of the visor lockings – were observed as follows:

- The vertical-longitudinal stiffener to which the bow end of the Atlantic lock visor lug was welded was lower than the drafted version. Since the weldments remained intact, this deviation had no effect on the casualty. (Visor bottom and stiffener edge plate failures on port side of lug stem appear to have come about at a later stage.)
- A couple of other stiffeners of the visor bottom plate were missing altogether. No effect on the events of the accident have been conceived.
- A structural reinforcement plate below the port side-locking lugs had been exchanged inside the visor as judged on the basis of weldment remnants from previous strengthening plate and the unpainted replacement plate's surface.
- Stiffener knee plates have been removed from the port side horizontal upper corner of the visor tying the visor deck to the aft bulkhead.
- About 100 mm of the 1000 mm long fractures on both sides of the visor bottom plate occurred in their fillet weldments, indicating some welding weakness, whereas the rest of the fractures were in the bottom plate. No effect has been conceived for the casualty of these weaknesses.
- An apparent crack indication in the lower port side outer shell plating, continuing into the stringer was observed to be a crack in the paint only.

It may be summarised that some minor features of damage and deviations from the as drafted structures have been observed. They may regarded to be unrelated to the visor or ramp attachment and none of these have been in a location or of the magnitude as having had any effect on the loss of the bow-visor. Other features related to the attachments are discussed in conjunction to the respective attachment.

5 Stiffness of the visor, visor seal and lifting cylinder reaction

Measurements of the stiffness of the visor were taken for purposes of judgement of the load distribution onto the various lockings under the action of the sea onto the visor shell plating. For this purpose one of the visor hinge arms (the portside) of the upside down stored visor was lifted and the resulting changes of the diagonals of the visor central opening were measured. The visor was supported on four "feet", each approximately under the ends of the hinge beams. The result was obtained with a 37 ton lifting load - which needed a 20 ton counterweight at the other arm to hold the visor down. The diagonals changed by +6 and -5 mm's. This translates to +8.5 and -7 mm to the vertical deflections of the sides to the opening in the directions of load action with reference to the other side. The "shear compliance" of the visor is thus of the order of 25 mm/ 100 tons, which may be considered to be lower than the stiffnesses of the locking device regions. The conclusion is therefore that the visor is flexible enough that it may itself distort under load such that the loading distribution on the various lockings is not smoothed such as the design assumption of an even distribution of the sea loads onto the visor attachments would have meant. A more detailed study may be needed to clarify this issue however.

The flexibility of the seal was measured in order to facilitate comparison and quantification of reaction of visor support elements to wave-induced forces. The visor seal was found to have progressive stiffness, such that a 10 mm compression was reached with a 10 kN load per metre of seal length and the following 5 mm needed 15 kN more, see Figure 8.

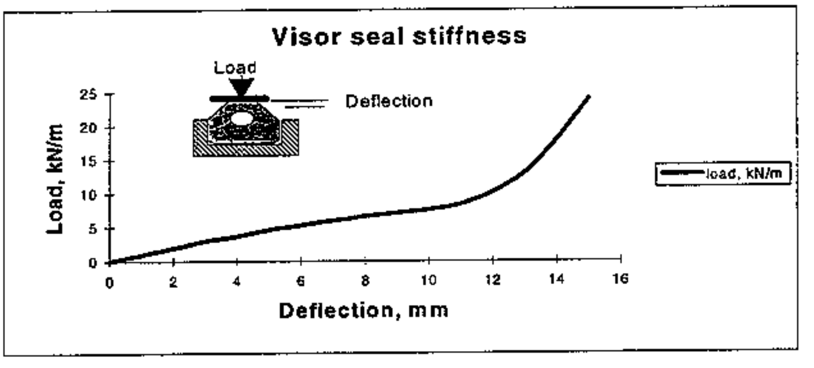


Figure 8. Visor seal stiffness as measured for 1 m length of seal.

Lifting cylinder reaction has been contemplated for whether they may have had an effect on the visor attachment reactions. Without considering oil flow inertia, the spring constant of the oil contained in the cylinder may be assessed based on the bulk modulus of oil. This is given as 1500 MPa in handbooks, translating to about 70 bars/% volume change. One percent volume change in one lifting cylinder will occur by about 13 mm compression and this would mean about 0.36 MN of load in the cylinder of id 280 mm and piston rod od 180 mm. 1 MN reaction at a cylinder would need then about 40 mm of piston displacement, translating to 4 times more at the side locks and up to six times more at the bottom lock. The lifting cylinders would therefore not cause a force reaction due to their spring reaction before the lockings had broken. Inertia effects of oil flow have not been considered as the wave load action is judged to develop more slowly than that causing oil flow inertial loads.

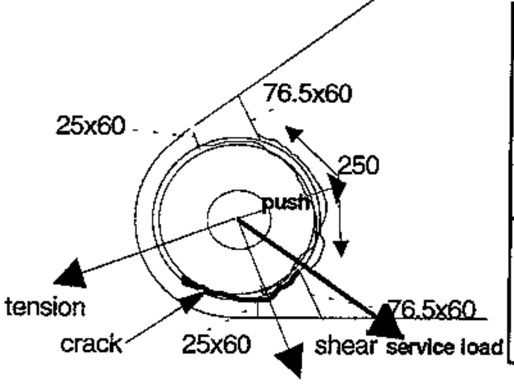
6 Strength of individual visor attachments

6.1 Hinges

The strength of the hinges is estimated in analogy with test results obtained at the Royal Institute of Technology using a slice of authentic hinge plate rim-bushings as recovered from the wreck.

Hinge strength assessed according to result by Royal Institute of Technology from testing authentic weldment

120000/(2x14x6.4) =717 N/mm^2



		area	strength	ultimate load	
Tension					
fillet weld	tension	6340	717	4.55	MN
	shear	1806.9	717	1.30	MN
weld total				**5.84**	MN
rims	tension	6000	250	1.50	MN
	TENSION TOTAL			**7.34**	MN
Shear					
weld tot	shear	6400	717	**4.59**	MN
rims	shear	18360	125	2.30	MN
	SHEAR TOTAL			**6.88**	MN

Figure 9. Hinge details and strength estimates for tension and shear. Effect of noted fatigue crack in the lower sector has been accounted for. Loads expected in service would cause push and shear components.

Accordingly, and referring to Figure 9, in loading vertically (called shear) and in aft directed tension (an irrelevant loading mode if locks are holding), the hinge is estimated to be able to carry load from 4.6 MN up to about 7 MN depending on hinge bushing to lug clearance, the weaker result applying to a larger clearance. Fatigue as noted in the lower sector of the bushing to lug weldment has been accounted for in the ultimate strength estimate but its effect is small. Progression of the fatigue damage, may, however, continue in cyclic load of lesser load amplitude. A large clearance between the bushing and the lug plate would increase the effect of fatigue and could lower the strength numbers from 4.6 MN.

6.2 Atlantic lock

Breaking of the Atlantic lock of MV Estonia occurred in the forepeak deck structure by fracture of weldments between locking bolt guiding and mating bushings and their holding lugs and bracket that held the bushings to the forepeak deck. Those fractures and the associated lug materials have been analysed and identified elsewhere. The mating lug on the visor had stretched and bent. The total lock is shown in Figure 10.

The strength of the fore peak assembly has been estimated (basic assessment also encompassed adapted EuroCode principles and methods for collapse strength communicated by Prof. E. Niemi, Lappeenranta University of Technology) to be twice the strength of the weaker or starboard side and is approximately then given by an equation as follows:

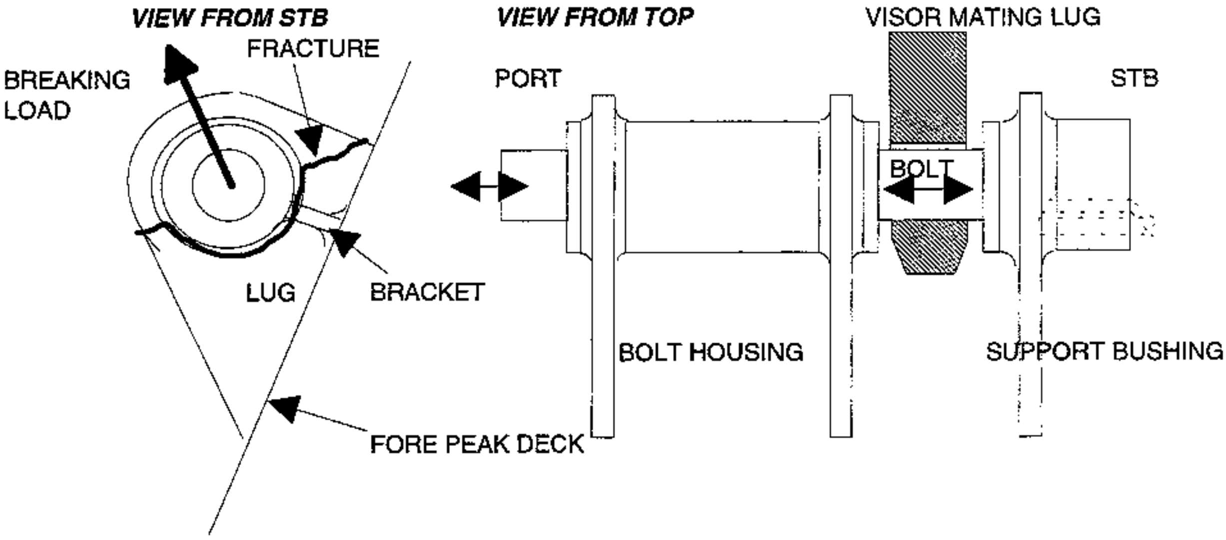


Figure 10. Atlantic lock fore peak deck structural components.

Strength of b-lock = 2 *	two sides
[f*UTS$_w$*a*(b(65+65)+128+128)/√2	weldment projections of bracket and lug at ultimate tension
+ f*0.57*UTS$_w$*a*(π/2*128-128)	lug weldment vertical sectors at ultimate shear
+2*g*YS$_{pl}$*15*36]	lug rims in tension at yield

Here the lug sections are loaded to a fraction g of the material's yield point (250 N/mm^2) and the weldments to a fraction f of their tensile strength (845 N/mm^2) on their projected cross section area, and corresponding fractions of their shear strength (0.57 of the tensile strength) on the parts of the weldments that are subjected to direct shear. To find values for f and g experimental results are used from welded and unwelded test pieces. An unwelded full scale model lock failing at 0.98MN yielded that g=0.89 applies to YS and a fully welded full scale test piece with nominally 3.9mm welds made of high strength steel St52 and tested at the Technical University of Hamburg (failure load 2.04 MN) that a factor f = 0.72 applies to the UTS. Such a value would not be far removed from a typical value frequently found for flow stress leading to fracture.

In the equation for the b-lock strength b is the bracket factor originating from the eccentricity of the loading mode of the starboard tie bracket. B = 0.125 = 0.5 * 16/64 from the assumption of bending of the bracket having a 16 mm internal moment arm and the local force couple being determined by tension on the projected area of a bracket fillet weld.

The narrowest range estimate of the strength of the bottom lock of 1.5 MN is based on analysing the deformation of the mating lug as presented below. For a loading capacity of 1.5 MN and a weld hardness of HV = 270 commensurate with UTS = 845 MPa as found on the actual weldment, a weld throat a = 3.5 mm is obtained. This compares well with spot checks of remnants of the authentic lock.

6.2.1 Authentic Atlantic lock visor-lug of MV Estonia

The mating lug on the visor had deformed both by stretching and bending. This study has found out how much the lug had stretched by tensile load alone (as part of the total stretch may have come about during excentric pulling after the maximum strength of the bottom assembly had been "consumed") and what tensile load has been necessary to cause that deformation. The material of the visor lug was identified to be mild steel by hardness testing.

A drawing of the visor lug is shown in Figure 11. A photograph of the Atlantic lock visor lug is shown in Figure 12 in its "in situ" upright position in the visor. The view is from aft port slightly from above the lowest horizontal stringer.

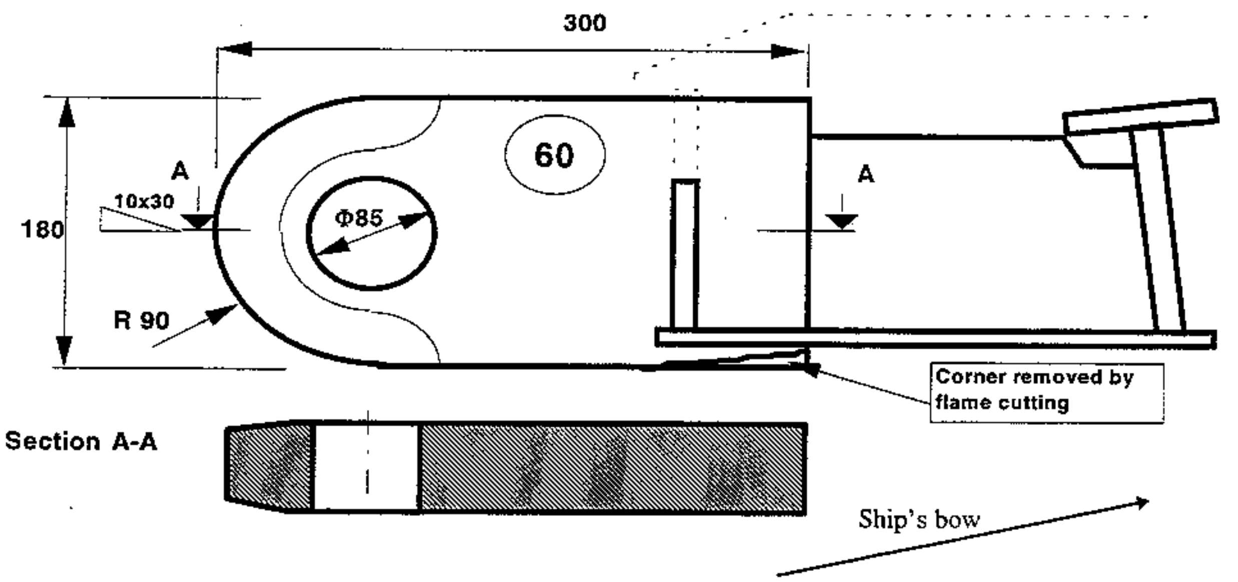


Figure 11. Atlantic lock visor lug as designed. The dashed part of the vertically positioned longitudinal stiffener to the visor bottom plate was missing in the actual visor above and to ship's bow from the lug.

The aft end of the lug is bent to starboard and the surrounding base plating of the visor has fractured on port and has buckled at starboard. This suggests that a fairly high starboard facing load in the bottom part of the visor has acted sometimes and apparently during the accident. Such a situation could have developed if a clockwise twisting rotation around the ship's longitudinal axis of the visor occurred due to a lifting action at the port side. Contact marks in the bottom locating horn recess on starboard suggest that the lower portion of the visor has been forced to port at some stage of the sequence of events. The resulting and remaining total sideways displacement of the aft end of the lug is on the order of 10 cm.

Figure 12. Atlantic lock lug site in the visor after visor recovery.

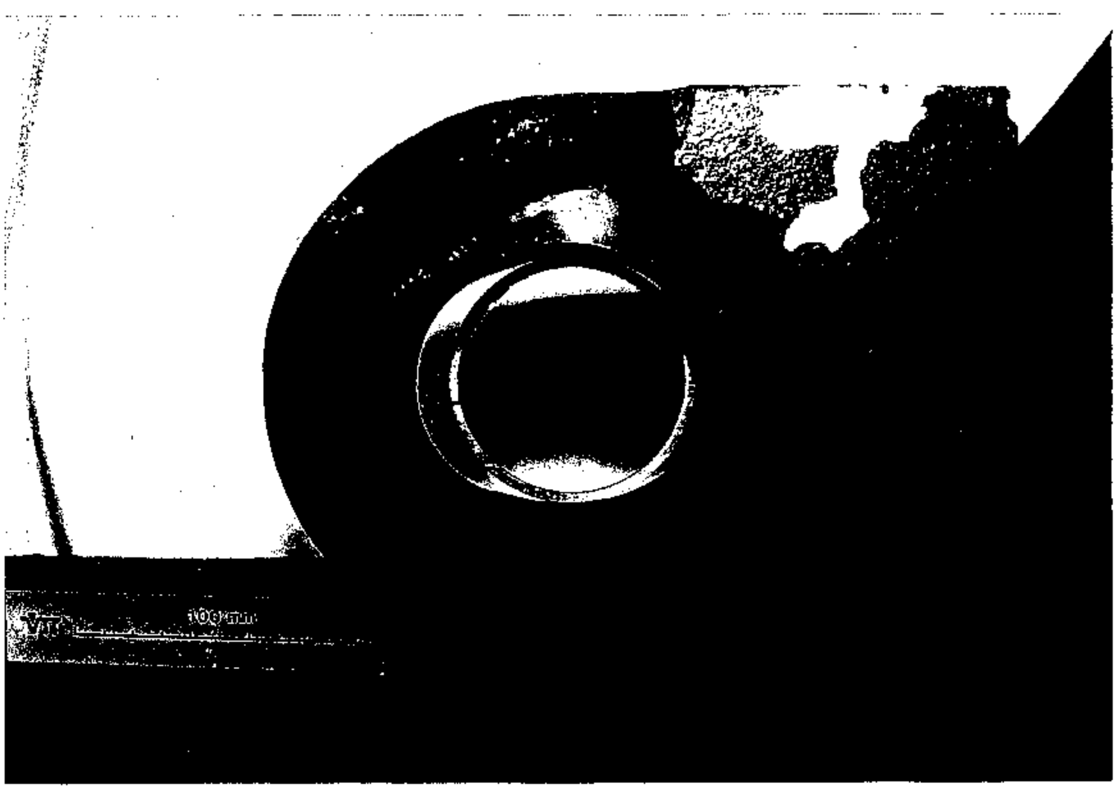


Figure 13. Side view from starboard of Atlantic lock visor lug after removal from visor. Ring insert in eye denotes the approximate apparent position of the eye in its original shape.

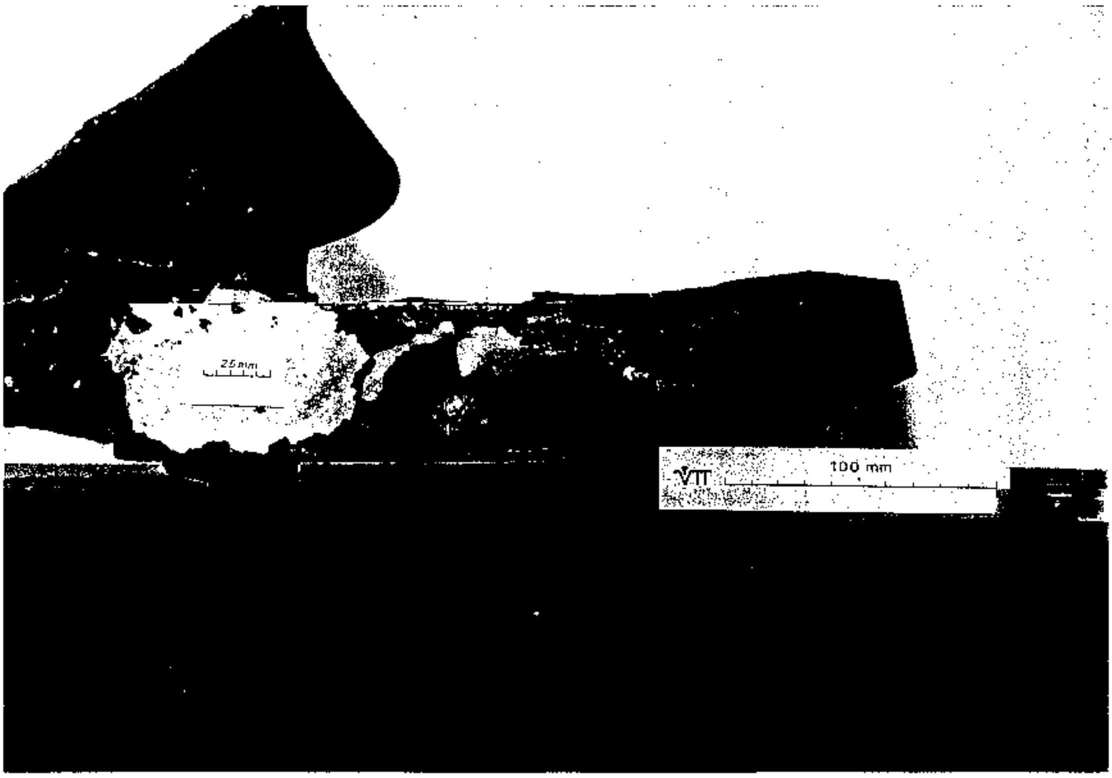


Figure 14. Top view of Atlantic lock visor lug after removal from visor. Lug aft end bent to starboard, bending of stem is very small and practically undetectable.

Photographs of the visor lug – as removed from the visor – are shown in Figures 13 and 14 from the side and from "top" respectively. Also here we may note deviation of the visor bottom plate stiffener dimensions from those marked on the drawing.

6.2.1.1 Visor lug material

The steel grade of the authentic visor lug was investigated by Brinell hardness number measurement using a 30kN (3000kp) indenting load and a 10 mm steel ball indenter. Three measurements gave HBS 10/3000 = 131, 136 and 137. This transforms into an ultimate tensile strength of around 450 MPa. It may be concluded that the steel grade is thus ordinary ship-building mild steel.

6.2.1.2 Visor lug deformations and maximum estimated load

In the following, the eye end is called the aft or stern end and the stem of the lug is located toward the bow end as the lug was in the ship. In the side view, a ring insert at about lug mid thickness is shown in the lug eye illustrating the apparent original position of the eye. At the upper pole of the eye the ring insert leans upon undamaged surface of the hole. The indication for this is that here we may discern machining marks from hole cutting potentially dating back to lug manufacture. In the long direction of the lug the original nominal diameter of the eye of 85 mm has elongated to the present deformed state of the lug such that at midthickness the hole is practically 95 mm "long". The transverse vertical diameter of the eye has reduced to about 83 mm. An analog transverse (vertical) contraction of the eye could be observed in tensile testing of mock-up and model lugs, elaborated in later sections of this report. An additional deformation of the eye is observed at its end towards the ship's bow as indicated by the ring insert. Compression and/or wear of the stem - or bow - end of the lug appears to have occurred and stretch of the aft end of the lug is apparent. The aft ligament of the eye along the lug is 47.5 mm, which is as designed and indicates no wear of the aft end. The "pear" shape of the deformed eye with the narrow end facing aft, is noted. The narrow aft facing eye sector occurs through the thickness of the lug. At the (vertical)"waist" of the "pear" the eye is about 80 mm wide and it is possible to fit a Φ 78 mm disc into the "bottom" (or aft end) of this end of the elongated eye. Φ 78 mm corresponds approximately with the diameter of the locking bolt. Hence it is judged that the lug has stretched and that the tension action has been caused by the locking bolt (or the reacting load held by its attachments).

The full length of the eye is about 95 mm at midthickness and 93 mm at ¼ thickness from the concave (starboard) side. At the aft end of the hole the side bend is 10 mm or slightly less.

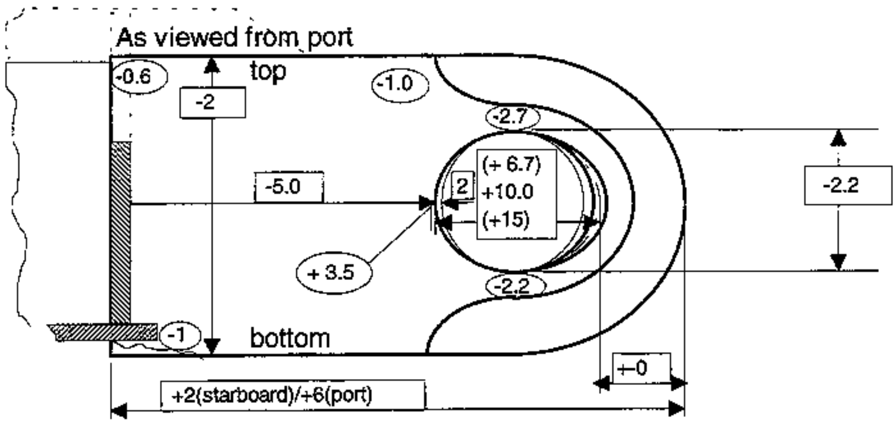


Figure 15. Lug dimensions as differences to nominal as drawn dimensions. Note also reduced thickness of eye ligaments above and below eye, which supports tensile stretch of lug.

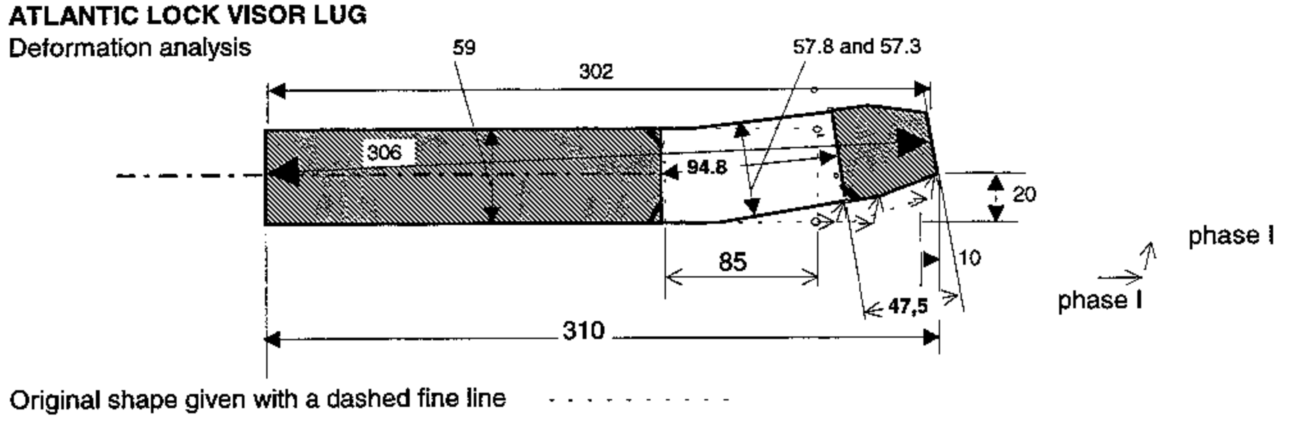


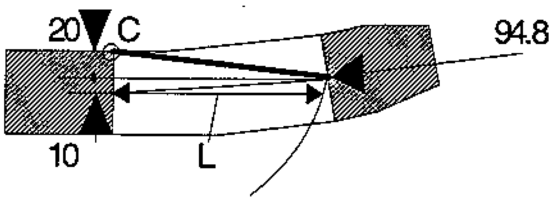
Figure 16. Longitudinal section of the deformed lug indicating also the path of assumed deformation. i.e. Phase I and Phase II. Crushed eye corners indicated.

Taken lug dimensions are given in Figure 15 as deviations from the nominal as drawn dimensions and in Figure 14 as actual millimetric dimensions of the lengthwise ("horizontal") through-thickness cross section of the lug.

An appraisal of the deformation of the lug end has been made. Here we observe that three of the eye hole corners (aft port and both bow corners) have been crushed leaving the stb aft corner unaffected. Apparently this crushing has occurred during later stages of the accident – i.e. the bending Phase II included – when the bending action by the locking bolt occurred after the forepeak deck side of the lock had broken. The bending has apparently occurred so that the locking bolt has been supported by the starboard bow end corner of the eye hole and the stern end port side corner has been pushed aft and to starboard by the bolt, the port end of the bolt having been held back by the hydraulic piston rod. The port bow corner crushing may have occurred during back and forth motions of the visor after all lockings and fastening sites had broken but with the locking bolt still in the lug eye and hanging on to the piston rod. This deformation can be produced with a comparatively low force and has been named Phase II. Analog crushing of eye corners at port aft and starboard bow corners of the eye were observed in all comparative mock-up tests.

Prior to this bending, stretching has occurred (Phase I) as evidenced most critically by the movement aft of the starboard aft corner of the lug. The shape of the eye prior to this bending, which also contributes some stretch, may be found drawing circles through the chosen stern midheight points of the eye ligament section using the supporting starboard bow corner of the eye hole as centre. The intersection of these circles with the corresponding profile lines of the unbent lug mark the locations of the eye hole ligament after phase I, i.e. at the end of the pulling phase, during which the critical forepeak deck lugs had broken. This "circular arc" deformation model was checked by back-calculating a 1/3-scale lug deformed by eccentric pulling at the Royal Institute of Technology". It was found that the elongation of the hole occurring due to this side bending is overestimated by the "circular arc"-model, and that a correction term equalling one tenth of the amount of side bending may be subtracted from the estimate of additional elongation for the side bending effect obtained by the circular-arc model to find the corrected elongated shape of the lug after phase I.

We may thus find the length of the hole prior to phase II bending by simple geometrical calculation. As illustrated below, by knowing the length of the eye at lug midthickness (l = 94.8 mm), thus having its measurement origin 30 mm sideways from the deformation circle centre C at the stb bow eye corner, and that the bending caused 10 mm movement of the stern end (the other end of the 94.8 measurement) to starboard we obtain for eye hole length after phase I



$$L(phase\ \mathrm{I}) \ = \ \sqrt{\left[94.8^2 - 10^2 + 20^2 - (20+10)^2\right]} \ + \ 10/10 = \ 92.8\ mm$$

The last term 10/10 is the empirical correction for the "error" of the circular arc model. This correction was deduced from a test using excentric tension performed at the Royal Institute of Technology, Stockholm.

For finding how much the eye had actually stretched during phase I, we need to identify the effect of wear on the shape of the eye. Here we get help from finding the original position of the original eye hole, that is we need to find out whether the eye could have elongated due to other reasons than stretching, e.g. wear or crushing. For this purpose we note that the upper sector of the eye still bears machining marks and has therefore remained unharmed through the life of the lug. Also the aft end ligament was noted to be 47.5 mm which is as designed and therefore taken to be in its original shape. The only location bearing signs of a different lengthening effect than stretching could thus have occurred at the stem (ship's.bow) end of the eye. For the purpose of identifying whether this eye sector has changed during use of the visor information on the behaviour of the position of the transverse ("pole to pole") midplane of the deformed hole during stretching is of help. We may find it so that the deformation of the hole may be divided up into deformations of the aft and bow halves of the eye. If the bow end deformation is subtracted from the total, the rest or the aft end deformation is taken to be due to loading.

To position the transverse ("pole to pole") midplane of the deformed eye the "radius" of the bow end of the hole was measured optically taking x-z coordinate readings of the hole edge at the aft and upper ("equatorial" and "polar") eye curvatures for both sides (port and starboard) of the lug (Note: By comparison to the original eye radius after finding of the "neutral" surface within the lug this measurement would also reveal the amount of wear on the bow side of the eye.). A second order curve fit was done to the measurement points. The position of its "lowest" or "polar" point was found by mathematical analysis and this point was taken as the position of the hole midplane as deformed. The position of the equatorial extreme bow end of the eye hole was read when the vertical hairline of the optical system was tangent to the eye hole profile. A second set of readings of the polar position was taken by fitting the tangent minimum to the hole or taking the midpoint between first "off-centre" grid intersections with the eye upper profile and taking the polar position as the midpoint of the former. Both methods were used for both sides of the authentic lug as well as both sides of the stretched only and the stretched and bent end of the full size mock-up lug reported below. Comparison measurement was used to test that method for one case using randomly chosen ten people to take measurements directly of the radius, i.e. everyone participating was asked to identify directly the position of the lowest point of the eye. Either method was found to give the position of the midplane to within 0.2 mm. This may be considered to be the objective accuracy of the determination of the original eye position (its "transverse" midplane) in the lug. Using the centric tensile only tested end of the full size lug mock-up it was found that the position of the vertical midplane of the eye stays at the distance of the original radius of the eye from the equatorial "bow" end through the tension loading. During eccentric pull the midplane rotates around an axle that is about 6 mm from the material midthickness towards starboard (the concave side), implying that some stretch of the stem end of the lug eye hole position occurs for phase II also at the lug midthickness.

From the full size mock-up tensile test explained in detail below the position of the midplane was found to remain unchanged during stretching. During the subsequent eccentric one sided tension - which caused mostly sideways bending and also some stretching - the mid transverse plane of the eye was observed to turn slightly such that the turning axis was around 6 mm displaced from the lug midthickness plane toward the concave side. At this position the radius of the deformed hole is 44 to 44.5 mm, which is 1.5 to 2 mm more than designed. It is logical to conclude that this difference is a result of wear or crushing.

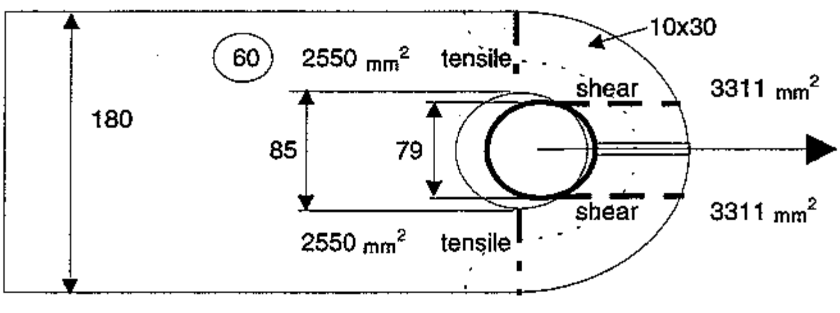
We may summarise these measurements further as follows:

- The length of the bow end stem of the lug is up to 5 mm less than nominal. This has been assessed more closely by observing the probable position of the transverse midplane of the eye using measurements of the undeformed upper sector of the eye hole. This way wear or deformation on the bow face of the eye has apparently caused up to about 2 mm elongation of the hole leaving 3 mm of the missing length of the stem to be due to manufacturing.
- The outer length of the lug is 2 mm more than nominal on the concave side and 6 mm more than nominal on the convex side. The elongation on the concave side indicates also that the lug has elongated due to pulling. Both numbers match with the overall situation supporting the conclusion that the lug has been 3 mm shorter than designed and that the net elongation of the lug during phase I has been on the order of 92.8 - 85 - 2 ≈ 6 mm.
- The aft end eye ligament is 47.5 mm, which is exactly the nominal indicating no wear or observable thinning of the aft end of the lug. The elongation of the aft end of the eye is thus concluded to have resulted from tensile stretching.

The elongation due to tension of the authentic visor lug of the MV Estonia Atlantic lock has accordingly been found to be about 6 mm. Supported by results from centric and eccentric tensile testing of model lugs and the full size twin lug made of highstrength St52 steel (HBN 165) as explained below this elongation has been estimated for high strength steel St52 to have required a load of approximately 1.76 MN (180 tons). The authentic lug was found to be mild strength A-grade steel, which reduces the estimate for the authentic lug by a factor of about 1-((450+250)/2))/((520+350)/2)) = 0.195, that is by almost 20%. The Atlantic lock of MV Estonia is thus concluded to have carried a load of at most **1.50 MN (147 tons)**.

6.2.2 Ultimate strength of the visor lug

A cross check was performed of the lug fracture strength and design calculation. The result is as shown in Figure 17 and as follows.



yieldonset MN	steel	GY/MN	**tensile** strength/MN	stress@1MN	**shear** strength/MN	stress@1MN
0.65	Fe 52	0.89	5100*520= 2.65	164 x 1.20	6622*286= 1.89	90 x 1.68
0.46	**A 42**	0.64	5100*450= **2.30**	125 x 1.57	6622*247= **1.64**	70 x 2.16
	STRESS AT 1 MN		196 MPa		151 MPa	

Figure 17. Ultimate strength and stresses at 1MN of visor lug.

The cross sections subjected to tensile stress are together $95 \times 60 - 10 \times 30 = 5100\ mm^2$. Taking that these could determine the ultimate strength a breaking loading capacity of 2650 kN or nearly 300 tons would be obtained. At design load the tensile stress would be 1 MN/ $5100\ mm^2$ = 195 MPa, which is more than the used permitted design stress for high strength steel. The shear stress would be 1 MN/6622 mm^2 = 151 MPa, which would be more than the shear yield strength for high strength steel and exceed a moderate permitted design stresses for both high strength (90MPa) and mild (70MPa) steel. The calculated lower bound shear fracture load would be 1.65 MN for mild steel grade A 42 and 1.89 MN for high strength Fe 52, the latter also being slightly lower than the test result of 2.04 MN obtained at the Technical University of Hamburg.

6.2.3 Authentic Atlantic lock visor lug of MV Diana II

The visor lug and locking bolt of the former MV Diana II were recovered from the ship for purposes of comparing measurements with the ESTONIA-lug. This was necessitated because of earlier observations of this locking mechanism to have significant play (reported to be on the order of 30 mm by Turbotechnic after the January 1993 accident) after the MV Diana II had experienced heavy weather and damage to its visor locking devices in early 1993. It became of interest to find if the locking bolt was unbent or worn and if the visor lug had stretched and/or bent. Photographs of the lug and the bolt are shown in Figures 18 and 19.

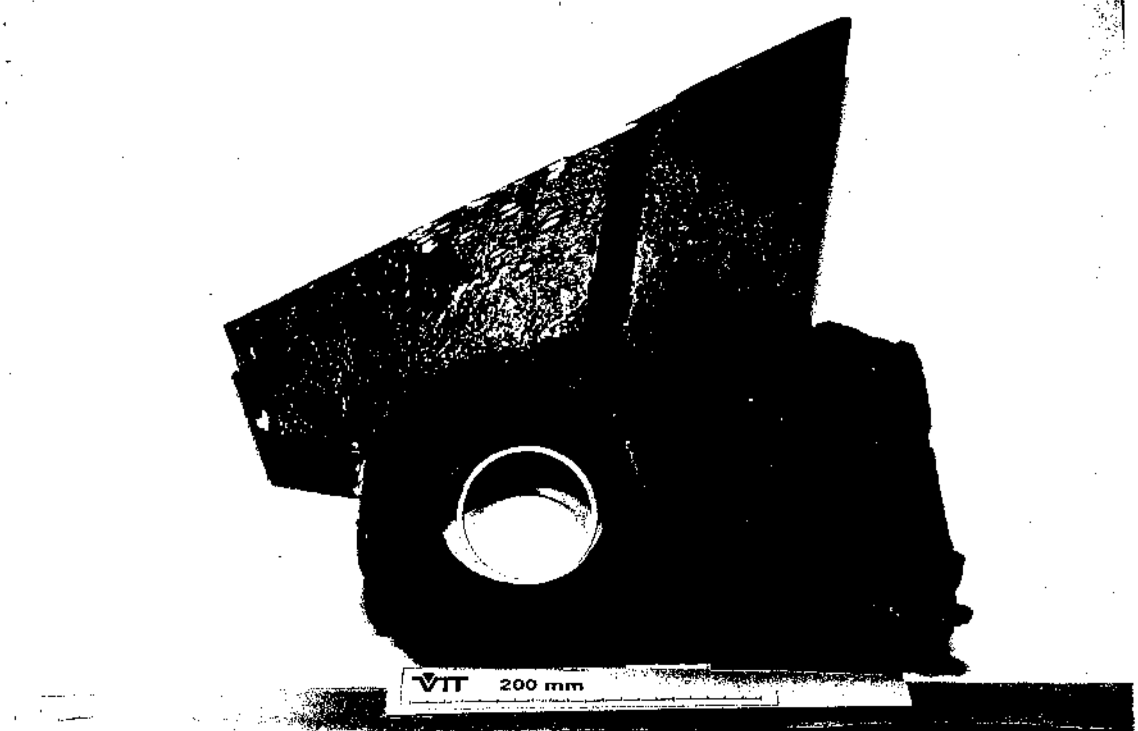


Figure 18. The Atlantic lock visor lug from the MV DIANA II. Part of the added plating surrounding the lug tip has been removed for measurement purposes.

The material of the visor lug of MV DIANA II was identified to be most probably mild steel. Brinell hardness was measured using 30kN (3000kp) load and a 10 mm steel ball indenter producing a 5.20/5.25 mm dent yielding HBS 10/3000 = 131/128, which corresponds with UTS = 430 MPa. The material is of similar strength as the MV ESTONIA lug.

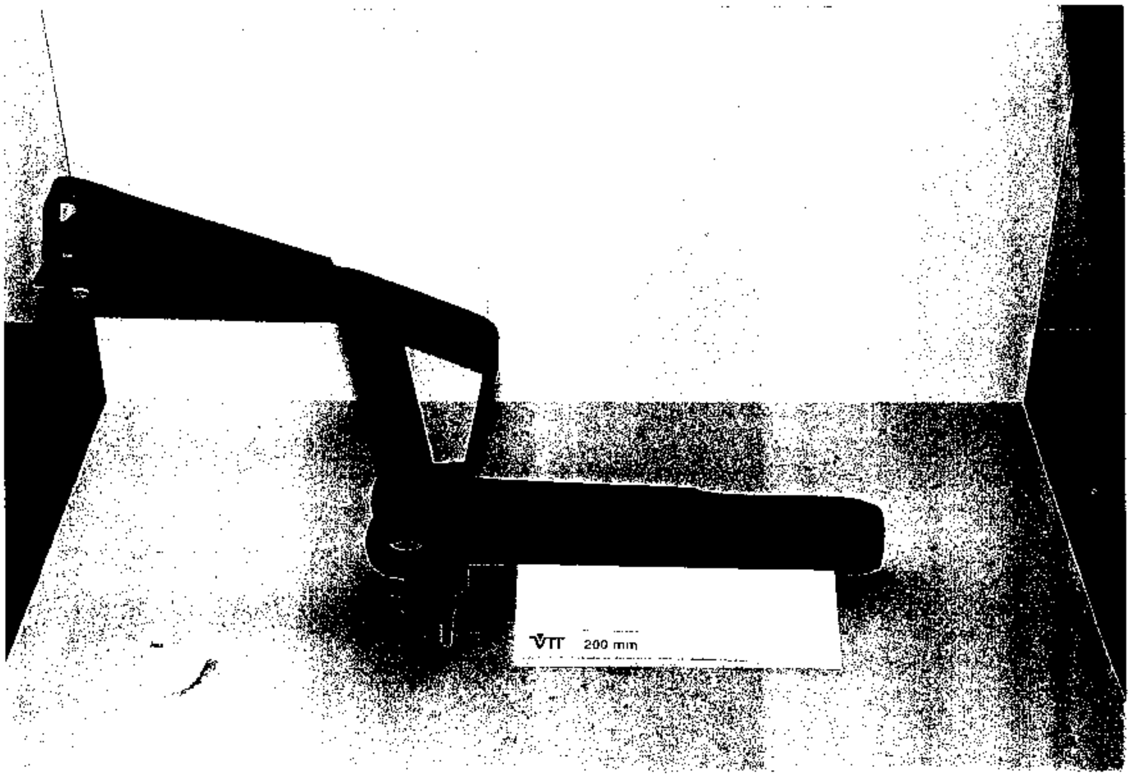


Figure 19. The Atlantic locking bolt from MV DIANA II. View from the upper aft sector.

The outer length of the lug is 305 mm, which is 5 mm more than the designed length. A ring insert measuring the designed eye diameter of 85 mm has been positioned into the lug eye. The ring demonstrates that the 5 mm increase in length appears to be due to stretching the aft or stern end of the lug. This is supported by noting the pear-shape of the eye, the stern end being close to the bolt diameter in close similarity with the eye of the MV ESTONIA lug. Significant wear is evident at the upper bow sector of the eye. Corresponding wear is evident on the bolt.

It is meaningful to note that the fore peak lugs of the MV DIANA II Atlantic lock have been more robust than those of MV ESTONIA, why it can be understood that the two ships can have met the same magnitude of bow forces as evidenced by equal magnitudes of lug lengthening, without the Atlantic lock having failed on DIANA II, however.

6.2.4 Full size mock-up tests performed by the Technical University of Hamburg

Of some interest is also to note the fullsize Atlantic lock mock-up test series performed by the Technical University of Hamburg on specimens manufactured by the Meyer Werft. The mock-ups had been manufactured of high strength steel (St 52) deviating from actual construction materials of the authentic lock found onboard of MV ESTONIA. The visor lug material of MV ESTONIA was identified only after the mock-up tests to be mild steel. Varying lug to bushing weldment lengths in the forepeak lug structure were applied. Because of unexpected breaking of the authentic sized visor lug in the initial test with "full" length weldments in the fore peak mock-up (3, numbers in order of strength of the mock-ups) leaving the forepeak mock-up intact, an extra test (4) was performed using an "oversized" visor lug. Two tests (1 and 2) were performed later, one with intermittent weldments (2) and one with no weldments (1) in the forepeak lug assembly.

The test mock-ups that failed in the forepeak lugs and numbered here as 1, 2 and 4 had either no weldments as test 1, intermittent – nominally 3 mm – weldments, as test 2, and test 4 that had continuous, nominally 3mm weldments between the forepeak deck lugs and the boltholding bushings. Test 3 also had continuous, nominally 3mm weldments, but the failure occurred in the mating visor lug.

The mock-ups failed at 100 tons (1), 142 tons (2), and about 200 tons (3) and (4) by fracture of the forepeak structures in the tests numbered 1, 2 and 4 as listed whereas test no 3 failed in the pulling "visor" lug. Test 4 had an oversized "visor" lug. The conclusion is that about 200 tons seems to be the absolute theoretical maximum value of strength of the fore peak part of the Atlantic lock configuration as designed, if weldments were sufficient by "a" and continuous and had the lock been manufactured of high strength steel. Failure of the pulling/mating lug occurred by combined shear/tensile fracture of the tip of the lug and the failure load corresponds with simple shear failure criterion of 0.55 of the ultimate tensile strength being reached on both sides of the pulling bolt. 0.55 is slightly less than the theoretical for plane strain shear of 0.57 (or $\sqrt{3}/3$) indicating only slight deviation from simple theoretical and highest possible.

100 tons is the breaking capacity of the forepeak lug assembly if no weldments at all are made. While the minimum cross section of each lug was $36 \times 15 \times 2$ mm^2 = 1080 mm^2, giving about 520×1080 mm^2 = 57 tons for the capacity of each lug. The combined capacity of the system of three lugs is on the order of 1.8× the individual capacity. Information from spot checks of the weldment size was obtained indicating a = 3.9 mm. The hardness was HV10 = 330. Taking these values for the fillet weldment dimension and hardness and the weldments are obtained to break in direct tension at about 0.72 of their ultimate tensile strength on the normal projected cross section, which may be taken to suggest a simple ordinary flow stress criterion of a moderately tough weldment. The lug rims were assumed to be at a fraction of 1/1.8 of yield when the weldments break at the maximum capacity of the assembly. The weaker starboard side of the Atlantic locking may be taken to provide half of the loading capacity of the lock. This yields that the lug rims provide about 0.5 MN to the total loading capacity of the bottom lock and the rest will be provided by the weldments up to about 1.8 MN, which in turn is the maximum estimated capacity of the recovered authentic visor lug. Taking that the bottom assembly had broken at about 1.5 MN, the weldments can have had an average thickness of 3.5 mm had they been continuous. This is reasonable as the observed weldment dimensions have been on this order.

6.2.5 Model tests and load estimates of Atlantic lock visor lug

Five model test pieces were manufactured of the visor lug. All model lugs were double ended, meaning that they consisted of two lugs each lying "stem to stem". Two were in Fe 52 steel like the design intent of the authentic and three were made of extruded aluminium (AlSi1Mg). The steel model lugs were one in full scale and one in 1/3-scale (linear) and the aluminium lugs were in 1/6 scale. The two larger model lugs are shown with the authentic lugs in Figure 20.

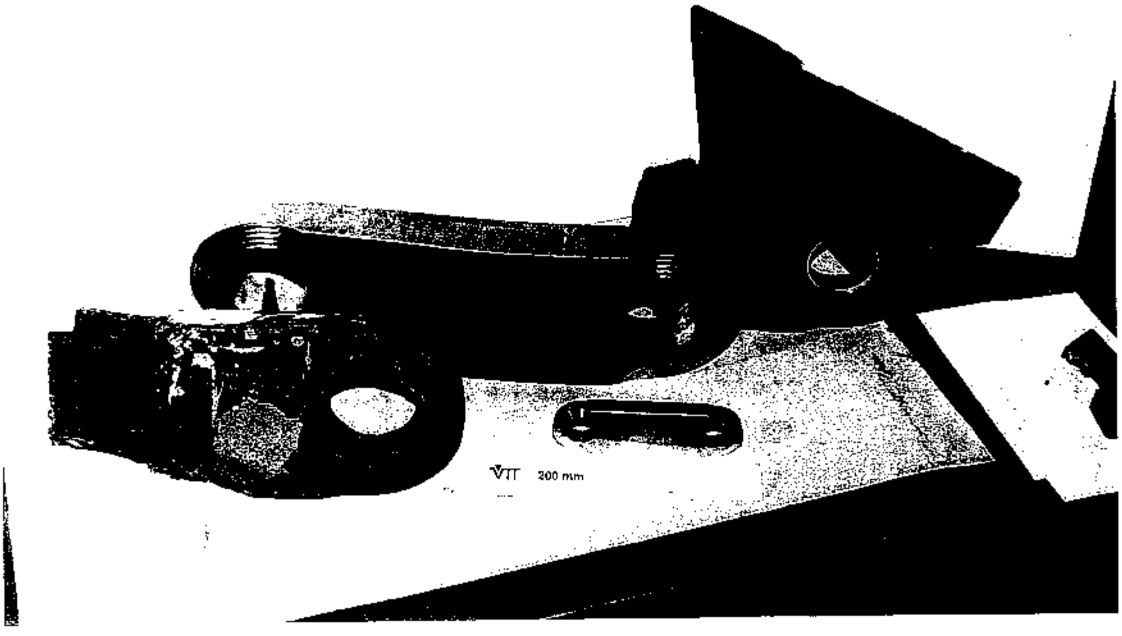


Figure 20. Full size and 1/3 scale model lugs together with authentic MV ESTONIA and MV DIANA II Atlantic lock visor lugs.

The tests were:

- **Centric full size Fe-52 tensile test**: The full size mock-up lug (both ends) was pulled to a total of 7.5 mm permanent elongation of each eye. The tensile eye elongation - load curve is shown in Figure 21. At final elongation of 7.8 mm for each hole the load was 1.92 MN. **In the range of deformation of interest for interpreting the authentic lug load, i.e. around 6mm eye elongation the curve is nearly linear according to load/tons = 172 + 8.5*(X-5) for Fe 52-grade steel, X being the permanent elongation in mm. A correcting term approximated by (50/350)*172 = 24 tons may be subtracted to obtain the number for A-grade steel (St42).**

- **Eccentric full size Fe-52 tensile test**: One end of the full size lug was subjected to loading by a bolt acting as a lever. The load was applied at one end of the bolt pulling along the length of the lug. The pulling force was applied in two phases, 350 kN and 115 kN at 0,2m and 0,6 m distance respectively to the lug, both excerting a maximum bending moment of 77 kNm plus a pulling stress of around 200 N/mm^2 on the side ligaments of the eye. The side bending thus achieved was 10 mm within the eye of the lug. Significant bending of the lug stem also occurred in addition to this. The position of the eye midplane transverse to the lug was subsequently found to have been unaffected by the bending at a distance of around 6 mm from the lug midthickness.

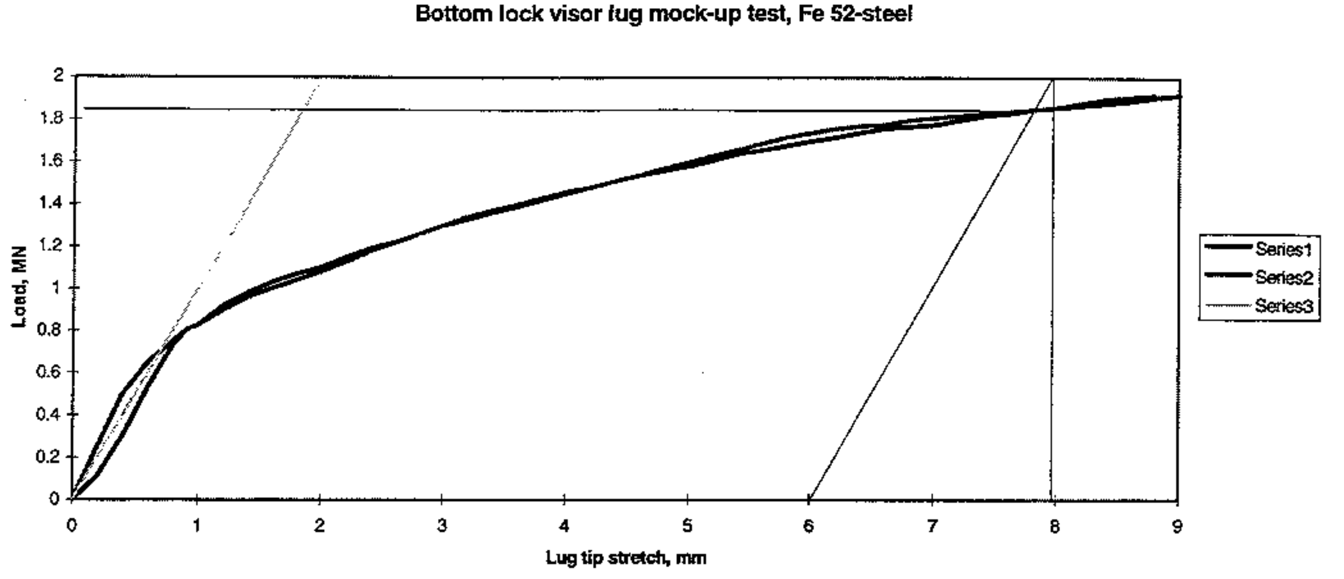


Figure 21. Load- eye elongation curve of full size Atlantic lock visor lug model of St 52 high strength steel.

- **Eccentric 1/3-scale Fe-52 tensile test with bolt end support:** This test was undertaken such that the bolt ends on one side of the lug were pulled with a supporting bracket inserted between bolt ends on the opposite side of the test lug to prevent them from taking support from the lug eye corners at the stem. The test was done because the deformation analysis of the authentic lock lug had indicated that some bending had evidently been combined with phase I pulling and it was unclear, whether such a combination of bending could reduce the pulling force needed to cause a certain nominal extension of the lug. It was confirmed that the lug deforms to closely resemble but slightly exceed the bending effect estimated for end of phase I of the authentic lug. The load required to stretch the lug midthickness location is not affected by this mode of applying the load.

- **Two eccentric 1/6-scale Aluminium tensile tests using pivoting C-clevises for bolt attachment:** Two tests were made. The purposes were to find the effect of the magnitude of eccentricity of pulling on the type of deformation that could be obtained. The magnitudes of eccentricity were e > 5 mm and 5 > e > 0, i.e. the line of force being either outside of the lug dimensions or inside. In the first case the pulling started with the centre-line of the lug being 30 mm from the line of pulling. In the second test this distance was around 2 mm. The lug ends twisted in both cases, but only in the second case did the concave side of the lug stretch to a length larger than the original length of the lug. Bending was much less than for the case of the larger eccentricity and full contact between the bolt and the lug eye was obtained as in the authentic visor lug. Only eye hole corner contact with the bolt was obtained in the test with large eccentricity. A pear shape of the full thickness of the eye was only obtained in the test with small eccentricity. This test also supports the conclusion that the authentic lug was at some time pulled in a phase I-fashion with a force keeping the bolt in contact with the full thickness of the aft end of the lug.

6.3 Side locks

6.3.1 Fractures

Detailed observations of fracture morphologies were made on the side lock remnants on the visor on both sides. The corresponding locations of the visor are shown in Figures 22 and 23.

Figure 22. Side lock region on port side as found on visor after the casualty.

The general breaking mechanism had been by fracture of the horizontal stringer under the lug, vertical stiffener edge corner and tear (along the lug weld edge) of aft plating onto which the locking lug eye plate had been attached by welding. Apparently the lug to bulkhead-weldment had been stronger than the rest of the structure, judging from the fact that nothing of the lug related material had been left on the visor. The holes left in the visor aft bulkhead measured 390 mm * 85 mm, the foot print of the lug only being 380 mm * 60 mm. The width of the weldment along the plating outside of the lug plate footprint has thus been on the order of 12 to 13 mm.

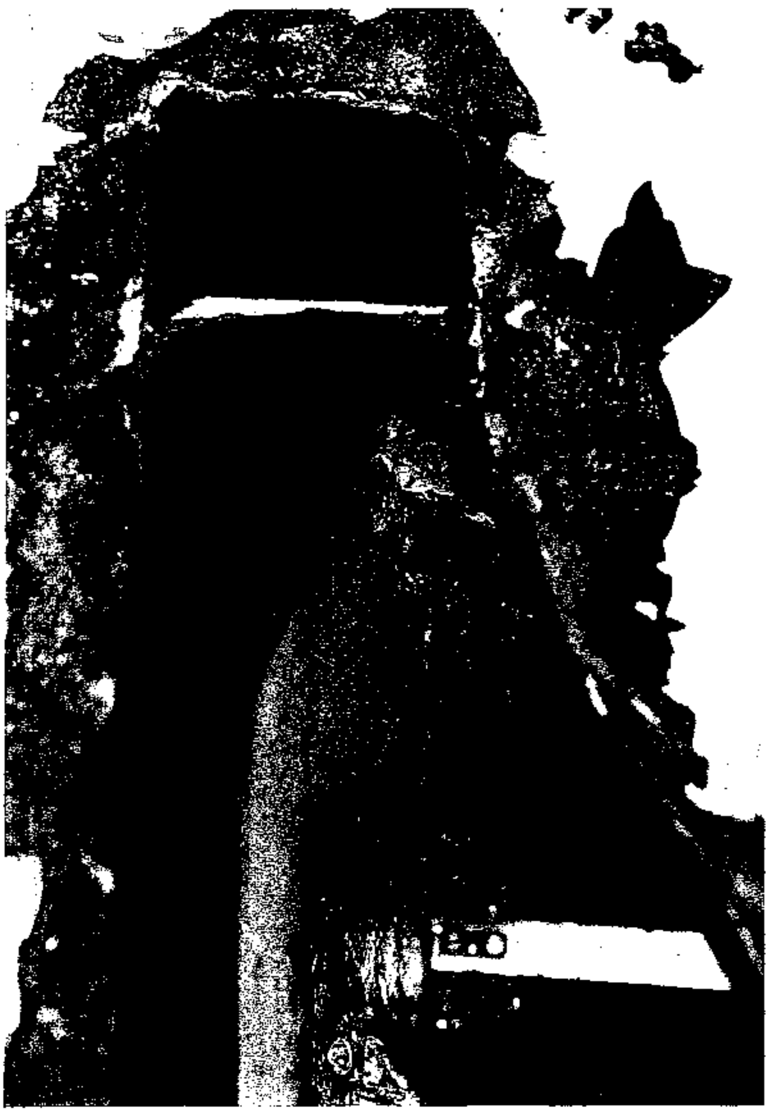


Figure 23. Starboard side lock site of the MV ESTONIA visor.

The strength estimation of the side locks presented a particular problem for numeric or analytic methods, because of their complex structure consisting of plate, stringer and stiffeners. Numerical modelling was done in a parallel project using finite shell elements and elastoplastic material constitutive laws for stress-strain interdependence, analytic calculations were done using "primitive" linear stress distribution assumptions. The static indeterminacy of the side-lock assembly called for an adjustment factor in the analytic calculation for structure member compliance differences. This factor was found by analysis of experimental data. Numerical modelling results were interpreted as such.

The testing program evolved into a series of tests because of new needs that uncovered consecutively after each test. Subsequent tests incorporating improved similarities with the authentic visor structures were then done. Four side lock mock-up tests were made altogether, the last of them corresponding best with the actual visor structure.

6.3.2 Material investigations related to visor side locks of MV Estonia

Photographs of the holes left in the visor at the sites of the side locks are shown in figures 22 and 23. The holes in the aft bulkhead measure 390 mm by 85 mm. Fracture had occurred by shear and tear through the bulkhead plate on both sides. The horizontal stringer fractures are different on port and starboard but the vertical stiffeners seem to have failed by shear and tear through the stiffener edge corners.

Basic mechanical material properties were determined by standard types of tensile tests for the bulkhead and horizontal stringer plates. Tests were done in the direction of the plating and one through thickness tensile test was performed to find out if the aft visor plate is weakened by lamellar tearing. The tests show that the plates are standard A-type high quality ship-building steel (Bulkhead: YS 306/311 MPa, UTS 454 MPa, A5 39%, RA 68%; Horizontal stringer YS 332/336 MPa, UTS 476 MPa, A5 33%, RA 58%). Through thickness strength is not reduced by bulkhead plate lamellarity (YS 335MPa, UTS 474 MPa). Chemical analyses were taken from the vertical stiffener, showing it to be good quality killed weldable structural steel plate (C 0.13, Mn 1.00, Si 0.19, P 0.025, S 0.027, Cu 0.01, Al 0.035 /%). The materials are thus of high quality as also evidenced by the ductile fracture morphologies.

From material investigations and fracture morphologies it may be summarised that side locking strength is not reduced by material or welding related factors on starboard, as fractures occur through the plates as shear and mixed shear and tear. On port side on the other hand attention is drawn by the horizontal stringer fracture, half of which is shear through the plate itself but the other half being a weldment related failure since the stringer plate edge can be seen. The lower corner of this edge has been sheared off but the upper surface of the plate seems almost untouched by welding and the corner of the edge is preserved.

An apparent small size of the fillet weld bead on the lower side has been well fused to the plate but failure occurred by the corner of the edge having torn off. The total width of the sheared fracture is on the order of the thickness of the plate. Having fractured due to almost simple shear, the weak half of the horizontal stringer to bulkhead joint has contributed to the strength of this joint by one half of its tensile strength. The other half being of full strength, a total of about 75% of the horizontal stringer strength has been assumed to contribute to the port side lock holding strength

The rest has apparently withheld as much force as was possible geometrically and materialwise, noting that the materials have good ductility and are therefore not sensitive to the sharp notches of the weldments in monotonically increasing load. Based on mock-up tests and strength calculations, the strength of the side locks has been estimated to be up to 1.59 MN for starboard and up to 1.19 MN for port using an assumption of load action at 38° to the aft plating plane of the visor and 3 mm width for the vertical stiffener weldment respectively. The height of the mating lug has then been taken as 214 mm from the eye centre to the visor aft plate midthickness. If the load had acted parallel to the aft plating, the readings would have been up to 1.35 and 1.02 MN for failure at the stringer at stb and port respectively. Only 1.2 MN would be relevant for starboard as the failure point would actually move to the bottom end of the lug as the shear limit strength of the aft plating would be reached first. The defected stringer would remain the critical location for port.

In order to check if lack of fusion can be suspected, a transverse cut sample was prepared by polishing and etching. It is shown in Figure 24, and displays apparent weakness of fusion on the upper side. The weld on the lower side was apparently too slim. Together they were unable to move the failure into the stringer plate.

Figure 24. Etched section through the horizontal stringer aft edge on port at the side lock displaying about 4 mm shear width on the low side (shear through edge corner) and 4mm shear width along plate surface on the top side.

The vertical stiffener was assumed to be welded using a 3mm weldment projected width for the tensile cross section. In the yard-designed condition, i.e. without vertical stiffener, the load readings were estimated to be 1.10 MN for port and 1.45 MN for starboard at 38 degrees to the bulkhead and 0.96 MN and 1.29MN for port and starboard respectively with load parallel with the aft plating. In all calculations above the spring constant ratio of the aft plating and the stringer-stiffener structural elements has been taken to be equal to 0.315 as found by test 4/2 and being slightly less than the theoretical 0.36 for plane strain shear/plane strain tension (1-v^2)/(2(1+v)), v=0.285. For other ratios the load capacity estimates would be different and up to geometry/judgement. The theoretical shear/tension spring ratio would be a good guess slightly raising the capacity estimates and a categoric neglect of a ratio - taking the value as 1, would act to lower the calculated capacity a little because of the aft plating getting to be critical. Shear failure would be contemplated for aft plating failure and tensile failure would prevail for the stringer stiffener elements. If tensile failure were assumed for the aft plating then a close to double capacity would be obtained - obviously seriously in error compared to the strength against the relevant transverse loading mode. Of some interest is to note that a calculated capacity of 312 tons with load in the lug plane and at 38 degrees to the aft plating is obtained for the two added vertical stiffeners to work at their full capacity, i.e. that they should have been welded through and positioned both under the mating locking lugs. The mating lug joints should then have been welded to the same nominal weldment dimensions as the stiffeners, i.e. with at least 20 mm penetrating weldments. The mock-up tests and estimation methods are elaborated below.

6.3.3 Side lock mock-up tests

The implemented mock-up test program for the side locks consisted of four tests altogether. A box measuring about 2000 * 600 * 200 mm was built such that side locking lug models were attached asymmetrically to the large faces of the box - simulating the visor aft plating. The lugs were positioned so that the load line of the assembly allowed pulling at 38° to the large face simulating tearing of the lug due either to visor opening or force couple reaction due to sideways loading causing the yawing moment. Tilting forward around the stempost would also cause the same local loading. A general view of the assumed load system of the visor and adopted asymmetric test arrangement is shown in Figure 25 (Other load directions were included in the calculations as explained below). A drawing of the last test mock-up is shown in Figure 26 and a picture of the test piece 4/2 in Figure 27.

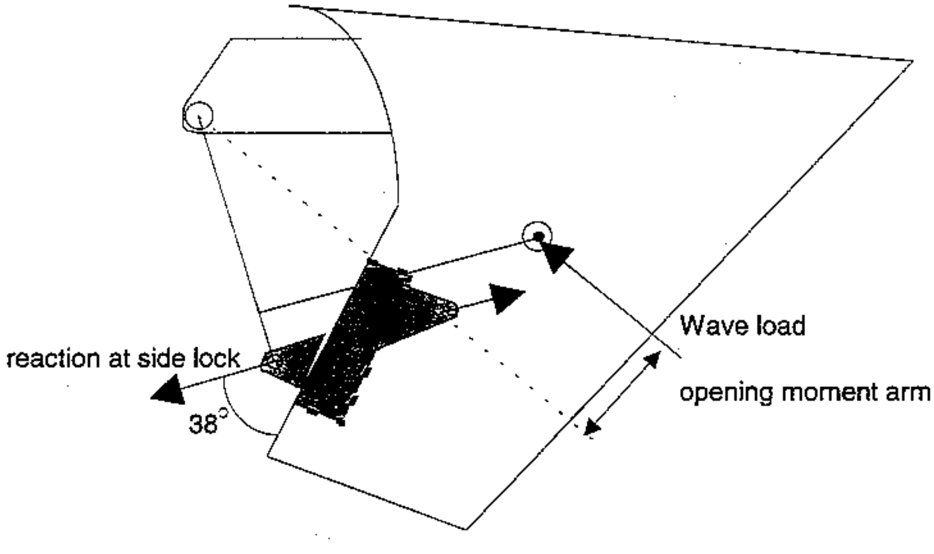


Figure 25. Antisymmetric structural arrangement and load system assumed for the side locking mock-up tests.

The four different tests differed from each other in the details of the stringer and stiffener arrangements. The fourth test mock-up box was further modified from the three first boxes by building the long side of the box to resemble the inner edge of the visor plating. The changes in structural arrangements from one test to the next had a very marked effect on the levels of failure loads, and it is summarised that only the fourth test result is relevant from the point of view of actual failure load reading.

The structural arrangements and failure details of the fourth test resembled the starboard side lock of the visor best of all tests. The second mock-up is also relevant in displaying the failure load level and stress value for only the aft plating of the visor after failure of the horizontal stringer but without vertical stiffeners, i.e. in yard "designed" condition. This failure occurred when the theoretical ultimate shear strength of the plate was reached at the point of maximum tensile resistance of the mock-up after holding by the horizontal stringer had been lost. This

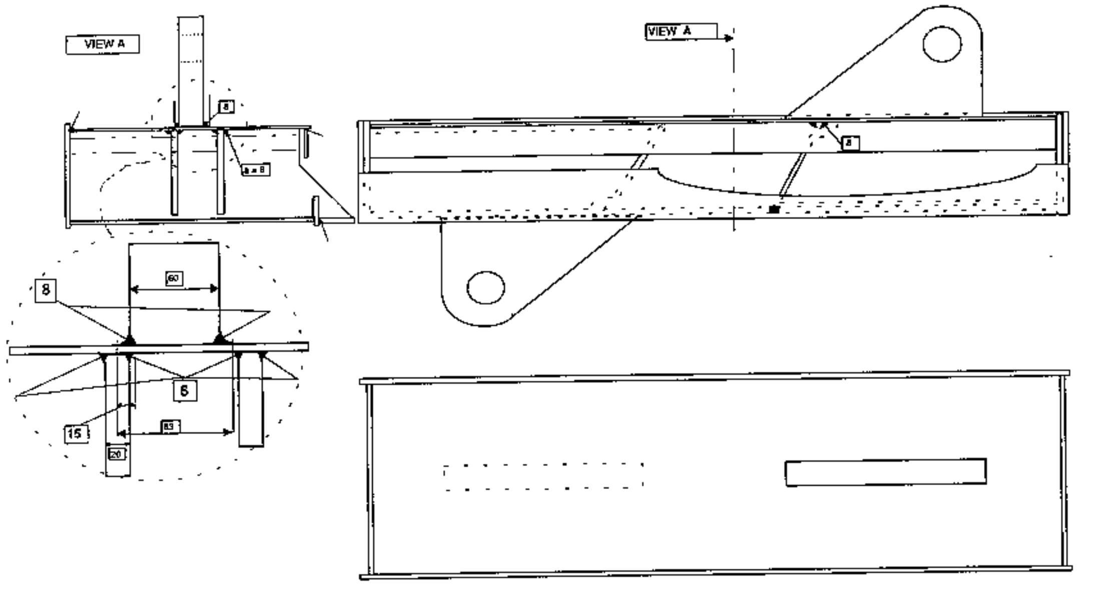


Figure 26. Drawing of side lock test mock-up No. 4.

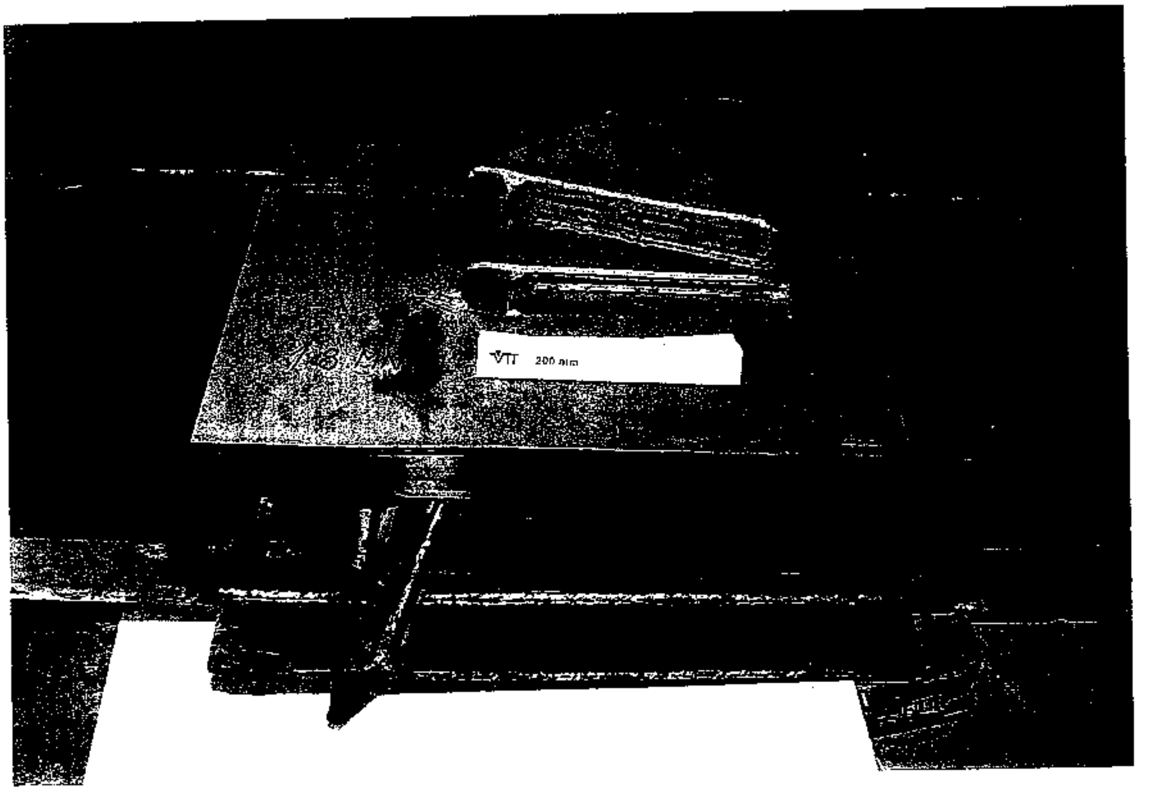


Figure 27. Side lock test mock-up 4/2 after testing.

test also suggested that stiffness differences between the aft plating and the stiffeners may be assessed assuming plane strain - or out of plane shear - to dominate the stiffness of the bulkhead and plane strain tension to dominate for the stiffeners. The side lock site is statically indeterminate and analytical strength modelling should therefore include such a stiffness ratio or spring constant effect for obtaining a realistic failure condition estimate and loading capacity number.

6.3.4 Strength calculations of side locks

For purposes of sensitivity analysis for structural and material deviations between the mock-ups and the actual structures a strength calculation macro of the side locks was programmed into EXCEL-worksheet assuming the linearized stress distributions (mechanically equivalent to real or the primary stress) around the bending neutral axis and linear stress-displacement interdependence. The geometry of the assembly and equations used are shown in Figure 28.

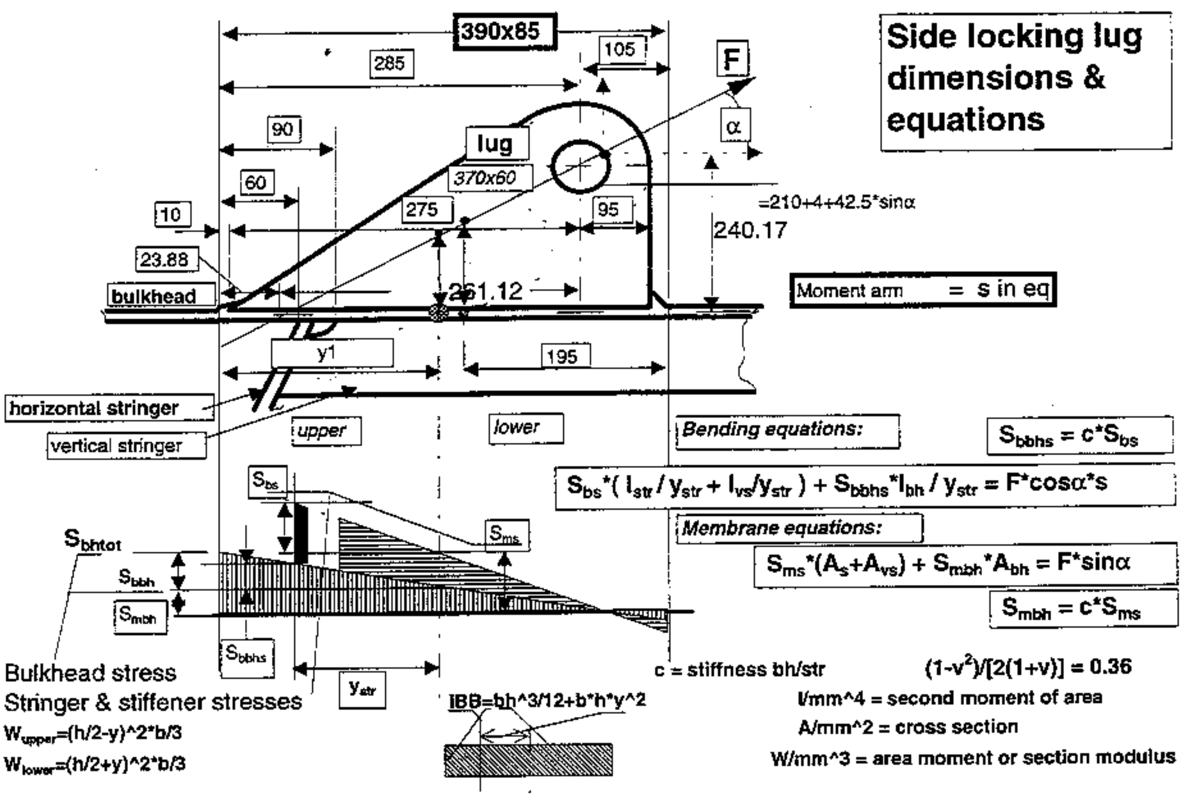


Figure 28. Configuration of side locks, structural details and equations used for strength sensitivity assessment.

The load capacities for starboard and port were thus possible to estimate. The procedure is the traditional, i.e. by seeking the neutral axis (NA) of bending (by setting the section moduli [mm^3] equal on both sides of NA) after which the moment arm for inclined pull is found asf.

Values for the breaking stresses of the various elements were as follows according to tensile testing of aft plating material:

Horizontal stringer and vertical stringer weldment in tension:	1.15*476 MPa = 547 MPa
Aft plating in shear:	0.57*476 MPa = 274 MPa

The effect of various directions of load action can be and was estimated. A load parallel to the visor aft plating was calculated, which would comply with a scenario of visor being raised without an opening movement as well as a load working through NA, i.e. a load with no bending effect. The former would correspond with e.g. an elastically "soft" direction of loading of the Atlantic lock and would include tension in the hinges. The pull only mode perpendicularly to the visor aft plating does not seem to be feasible but shear only parallel to the aft plating may be realistic. Supported by these test results, primarily test 4/2 (second loading of fourth mock-up after repair of its first irrelevant failure at 1.7MN) failing at 1.8MN, the estimated strength levels for the lock at starboard and port are given in Table 2. Plate dimension and material differences between mock-up and actual visor have been accounted for. The numbers represent those obtained for a 3 mm thickness of the broken vertical stiffener weldment.

Table 2. Side lock strength estimates for load acting in the plane of the lug.

	Port	Starboard
Load acting at 38°	1.19 MN	1.59 MN
Load acting at 0°	1.02 MN	~~1.35~~/1.22 MN
Load acting through NA(angle)	1.86 MN (62°)	2.43 MN (62°)

6.3.5 Side-lock mountings from MV Diana II

A side lock base left over from the bow modification for MV Mare Balticum was recovered for metallurgical survey of the remaining damages on the side locks of the January 1993 incident. Also the repair was observed, Figure 29.

The observation yielded remaining cracking in the original plating in several locations, one of them can be seen in the bulkhead plate starting at the toe of the fillet weldment of the vertical backing stiffener, Figure 29. Strengthening plate pieces had been added as part of the repair action, but the repair piece fit and welding quality of the repair appears poor. A strength estimate cannot be given.

6.4 Lifting cylinder lower platforms

Lifting cylinder lower platforms were torn from deck B, part of it accompanying the torn units. The principal strength element in this assembly is plate B and its circumference of the twin lug. It was found that on port slightly more than half of this circumference had cracked in a ductile manner displaying evidence also of fatigue. Part of older fatigue cracking was clearly evidenced by semielliptic black fracture surfaces, Figure 30.

The originally sharp-edged black colouring smoothed out as storage time elapsed, as shown in Figure 31. The black stain was quite evidently hydraulic oil and dirt.

Some of the rest of the remaining ligament was also due to fatigue and ductile tearing.

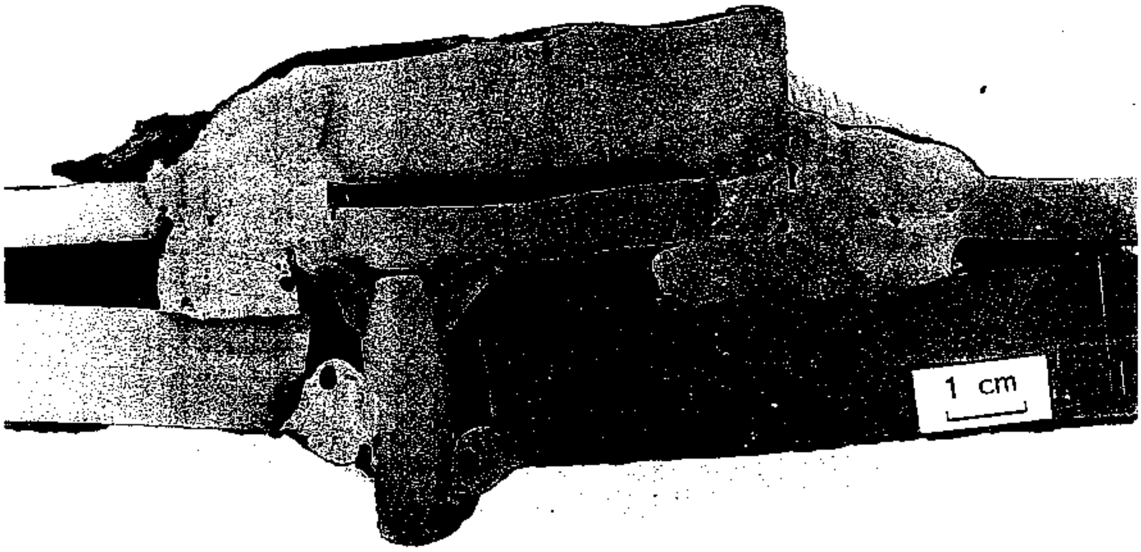


Figure 29. Metallurgical horizontal section through side locking lug, aft bulkhead, vertical stiffener and repair pieces of side lock site in MV DIANA II.

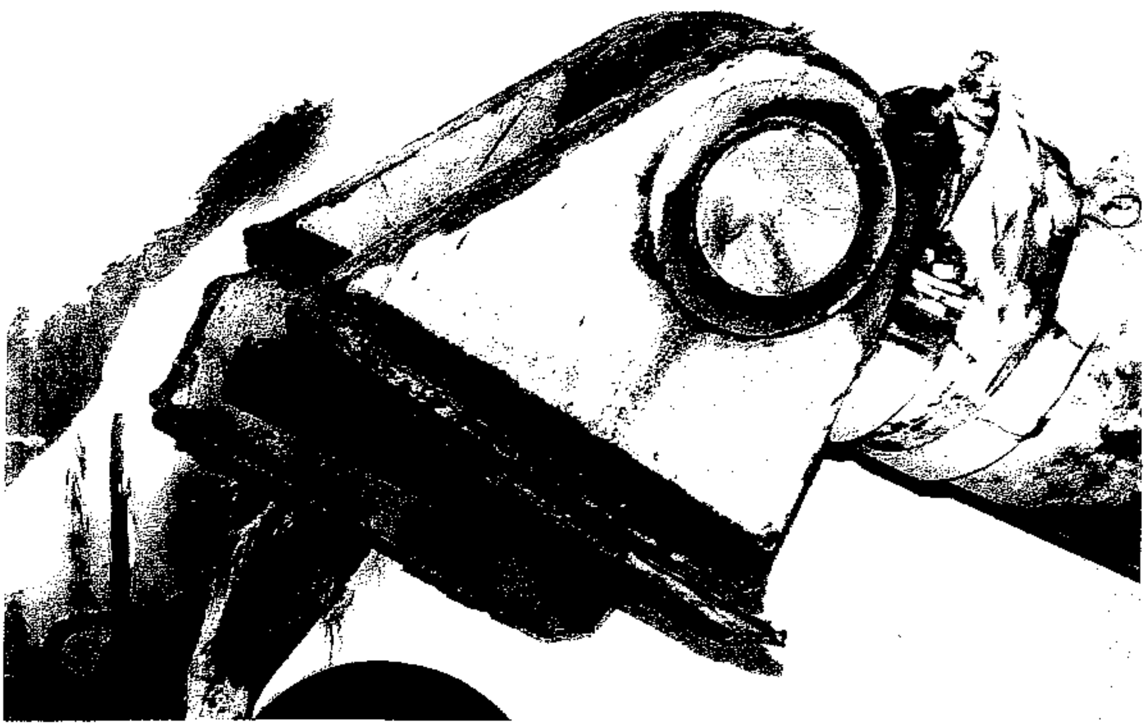


Figure 30. Port side lifting cylinder lower lug as recovered with visor.

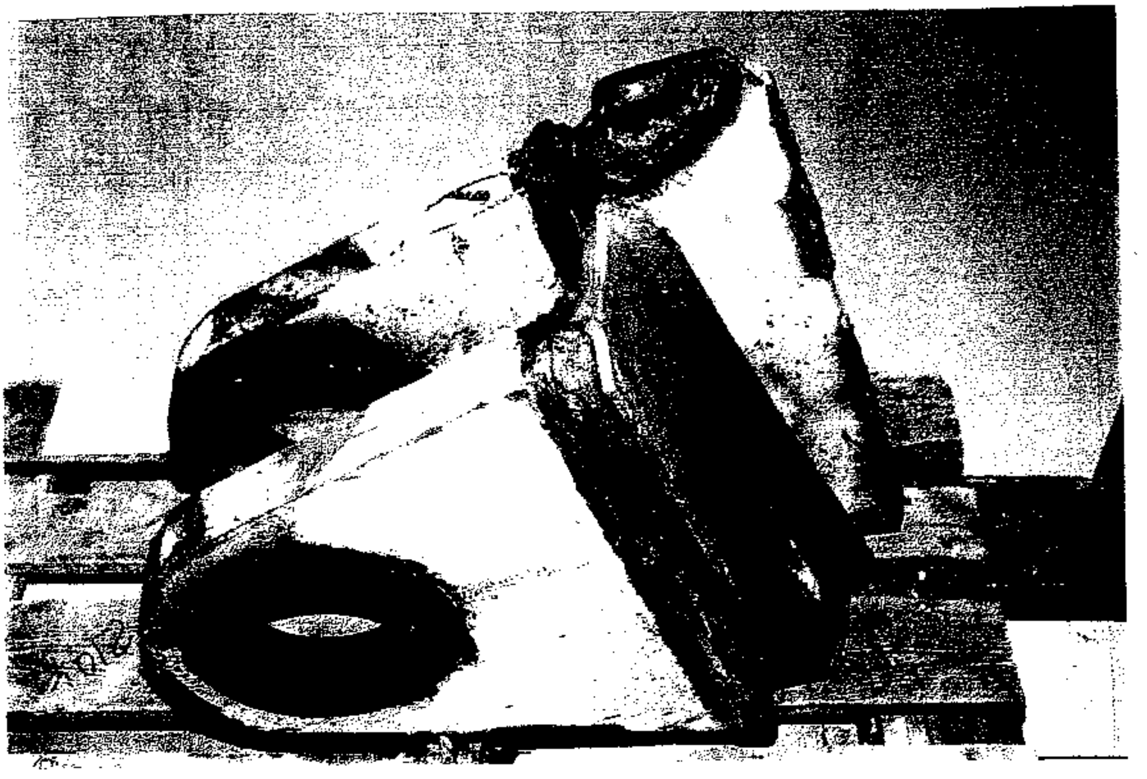


Figure 31 . Port lifting cylinder lower platform as seen from bow centre. Position tilted 120 degrees bow up. Bow end of lower port extreme stringer weld repaired. Centre stringer bent with broken fillet weld, nearest (centremost in ship) fillet weld missing.

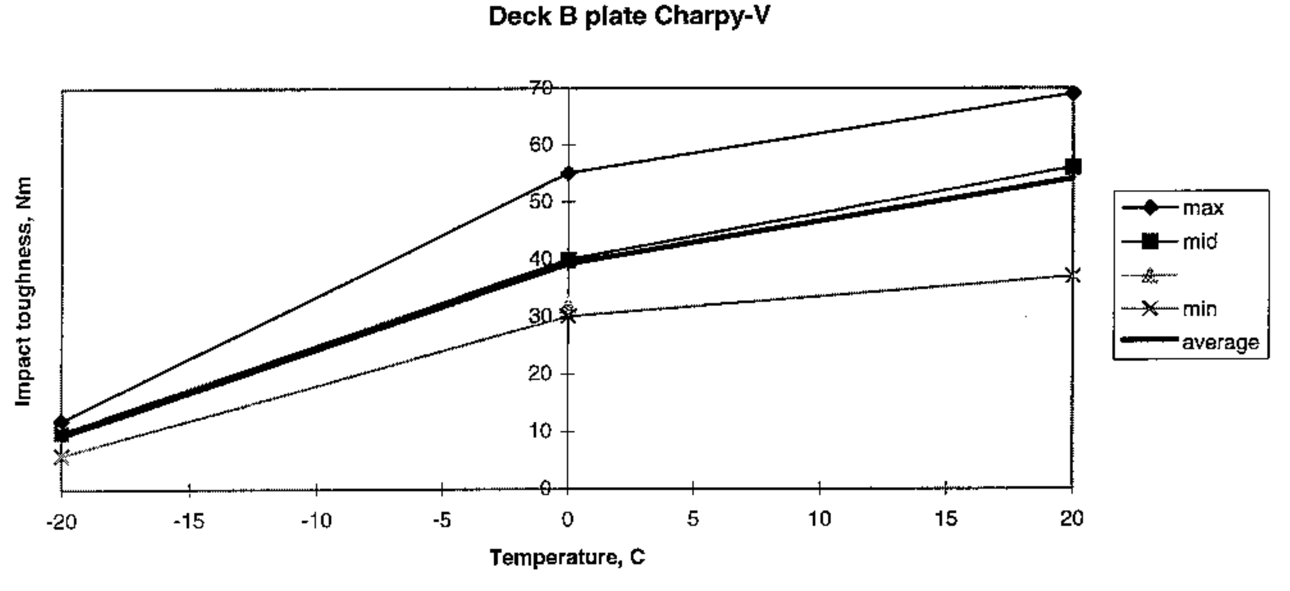


Figure 32. Deck B Charpy-V impact toughness values.

The extreme port side stringer under deck B had been weld repaired as evidenced in Figure 31. The centremost stringer had been welded with only one fillet weld - the other one missing. Twin fillets had been applied to all other stringers under port and starboard lifting cylinder lower lugs.

The remaining section had fractured by cleavage - a mechanism typical of cold brittleness. Deck B plate was also found by Charpy-V impact testing to have a tendency for cold brittleness, even at room temperature as indicated by impact toughness shown in Figure 32. The reduction of strength due to this possible source of weakness has not been assessed accurately, however.

The load to tear the lug platform from deck B may be estimated based on ultimate shear strength and the fraction of section left ahead of the fatigue cracked part of the plate. Taking that the platform perifery measures about 560+240+560+240 mm and the plate thickness is 20 mm, the total uncracked cross section = 0.032×10^6 mm^2. Assuming an ultimate shear strength of 250 N/mm^2, the tearing strength without fatigue could have been 8 MN, excluding a supporting effect of the underlying stringers. With half of the section cracked, only half of this capacity was left on port, and somewhat more on starboard. Because of possible brittleness in deck B plate it could have been somewhat weaker, leaving a probable range of 4 to 2 MN.

7 Summary and Conclusions

Bow load division onto visor attachments and the strength of individual visor attachments of MV ESTONIA have been assessed.

In the assessment of local load levels at the individual visor attachments a three-dimensional "space frame" idealization was used to simplify the geometrical configuration of bow load action centre and its location with respect to the five primary attachments of the visor, these being the two hinges and three locks. Acting load information was taken from basin model test results obtained elsewhere (SSPA-Gothenburg) as theoretically further elucidated in this work concerning interrelationships of moment and force components. Due to the actual three-dimensional arrangement of visor attachments - in particular the far aft removed hinge points in relation to the longitudinal positions of the visor lockings, severe bow load accumulation onto the lockings was observed. In oblique head waves from port of the bow the wave encountering side lock was deduced to take most of the load of all locks - possibly twice the load taken by the bottom lock.

All strength assessments of individual lockings are upper bound numbers and the marginal possibility of fatigue induced further weakness has not been assessed except for the hinges, where cracking reported to the Commission has been accounted for. From the appearance of the fractures of authentic attachments the upper bound solutions are relevant.

Using full size model testing and direct observations on authentic fractures the port side lock was found to be only a little stronger than the design load assigned to each lock. It was apparently the weakest of all - partly weakened by welding related deficiency at its primary structural weldment within the visor and seems to have possibly broken or lost its holding rigidity first in the sea conditions prevailing during the accident night.

The strength assessment of the bottom lock was based upon analysis of the deformed visor lug. Using the result obtained the apparent weldment size of the broken fore peak deck assembly was calculated. The result - an average fillet weldment size of about 3.5 mm - conforms with direct observations on the authentic lock recovered from the wreck, which has been investigated elsewhere.

Materials used to build the authentic visor lockings have been identified complementing identification performed elsewhere. No indications of high strength steel of the St52 - type has been found in the structural elements checked either by hardness testing or by tensile tests (hardness tests of the b-lock bolt from the near-sister ship MV DIANA II indicate the bolt material to be of higher strength). All plate materials investigated as related to the primary five visor attachments have displayed high quality and ductility. Only the lifting cylinder bottom attachments -deck B - were found to display signs of cold brittleness, but the effect of it has not been quantified. Being a secondary attachment from the point of view of possibly retarding the sequence of the accident, further work was not performed on this element.

8 References

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