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## IMPACT ON A STEEL SONAR DOME

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Author(s)  
dr.ir. R.H.B. Fey *RF*  
ing. J.J. Uwland

**Centre for Mechanical Engineering**  
Schoemakerstraat 97  
P.O. Box 49  
2600 AA Delft  
The Netherlands

Phone +31 15 269 69 00

Sponsor : Ministry of Defence  
Royal Netherlands Navy  
P.O. Box 20702  
2500 ES Den Haag

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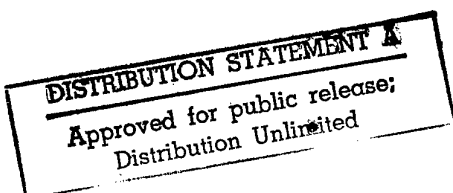
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## MANAGEMENT UITTREKSEL

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### *Aanleiding voor het onderzoek*

In het verleden zijn de sonar domes van de S-fregatten uitgevoerd in staal. Over een periode van meer dan 15 jaren zijn er schadegevallen geweest aan de stalen domes, maar die hebben nooit geleid tot disfunctioneren van de sonar. Vanwege betere transmissie-eigenschappen zijn de domes van de M-fregatten gemaakt van composiet. Tot op heden is één van de composieten sonar domes totaal verloren gegaan, terwijl er ook één door impact zware schade heeft opgelopen.

### *Doel van het onderzoek*

Voor een representatief schadegeval aan een stalen sonar dome zal worden geanalyseerd wat de contactkracht tussen het object en de dome is geweest tijdens de impact. Bovendien zal worden berekend welke hoeveelheid energie door de stalen dome is geabsorbeerd. Met deze gegevens zal in een later stadium getracht worden specificaties te formuleren voor een verbeterde composieten dome.

### *Korte omschrijving van het onderzoek*

Ten einde de contactkracht en de geabsorbeerde energie tijdens de impact te berekenen is een eindige-elementenanalyse uitgevoerd. De moeilijkheid bij de analyse was met name het feit dat gegevens over de impactor, die de schade aan de stalen dome veroorzaakt had, ontbraken. Dientengevolge zijn aannames over vorm, grootte en impactrichting van de impactor noodzakelijk geweest.

### *Conclusies*

Het berekende deformatiepatroon komt goed overeen met het deformatiepatroon zichtbaar op de beschikbare foto's. Bij een penetratiediepte van 0.2 m is de berekende contactkracht 230 kN en de bijbehorende geabsorbeerde deformatie-energie is 32.5 kJ.

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## 1. Introduction

In the 1980's sonar domes were constructed out of steel. Nowadays they are made of composite material because composite material disturbs the sonar signal less than steel. Recently, one composite dome was heavily damaged and one composite dome was even completely lost because of impact with unknown objects. The steel domes were also regularly damaged but the damage was always relatively local and the sonar dome did never fail.

The goal of this research is primarily to derive a criterion for impact resistance which will be used in the design of new composite domes. In order to achieve this, we will study a case of impact on steel dome no. 3, which was damaged by an unknown object, see Figure 1-Figure 8 (photos dd. July 26, 1984). This damage is thought to be representative for impacts under operating conditions. More specifically, the amount of deformation energy which is absorbed during the impact by the steel dome will be estimated.

Because the object, which hit the dome, is unknown, it is necessary to make assumptions about the size and the shape of the object, the stiffness and mass of the object, its speed during impact and the direction of the impact.

The calculations presented in this report were carried out with the explicit finite element package LS-DYNA3D [1,2].

## 2. Description of the model

### 2.1 Geometry

Photos of the undamaged dome are given in Figure 1-Figure 3. The thickness of the hull of the dome is 4 mm. There are primary frames (160 mm x 3 mm) and secondary frames (60 mm x 3 mm) on the inside of the dome in horizontal and vertical direction. On top of the primary frames are tubes with a radius of 7.5 mm. The horizontal frame distance between the vertical frames is maximal 375 mm. The vertical frame distance between the horizontal frames is 120 mm.

The geometry of the mesh of the dome, which will be described in more detail in the next section, is based on drawing Z 01.10 (August 1976) entitled 'Spantenlijst (vorm van de boeg-dome)', drawing G 44.6 (January 1975) entitled 'Ensemble du bulbe' and the telefax dd. September 9<sup>th</sup> 1997 from Rijkswerf Den Helder to MARTECH-MMT and the visit of TNO Centre for Mechanical Engineering to Rijkswerf Den Helder dd. November 19<sup>th</sup> 1997.

As stated before, the object, which hit the dome, is unknown. In order to get an impression of the geometry of the object which hit the dome, the photos of the damage to dome no. 3 have been examined (Figure 4-Figure 8). On the outside the damaged area has approximately a circular shape with a radius of 0.4 m. In this circle a right-angled dent is visible, see Figure 4 and Figure 5. Several hypotheses can be given for explaining the right-angled dent:

1. On the inside of the dome two large stiffeners are visible in the damaged area: a longitudinal stiffener and a transversal stiffener, see Figure 6.
2. The direction of impact is of influence.
3. The object which hit the dome had sharp edges.
4. A combination of the three hypotheses given above.

On the inside of the dome (Figure 6) we see tilting of the top horizontal primary stiffener over three vertical secondary stiffener distances. Also local buckling of secondary stiffeners is visible.

We will assume that the object which hits the dome is a sphere (because of the circular shape of the damaged area) with a radius of 0.2 m. The radius of the sphere is chosen smaller than the radius of the damaged area (0.4 m) because it is expected that a plastic hinge with a circular shape will move outward.

## 2.2 Meshing

Because the damage to the dome is local, located on one side and relatively high, only the port side is modelled. Actually, the dome visible on the photos was hit on starboard side. This is not relevant for the analysis, however, because starboard and port side are symmetric. Figure 9 shows the place on the dome, which was damaged (marked by dotted curve).

Figure 10-Figure 14 show the element mesh from different view points. The model consists of 11186 shell elements and 321 beam elements and has 10735 nodes. Near the location of impact the mesh has been refined. In this region 3 shell elements are used in the direction of their height for the secondary frames and 7 shell elements for the primary frames. The tube on top of the primary frames is modelled with beam elements. The minimal element length is 20 mm.

The positive x-axis points from the aft to the front of the ship. The positive y-axis points from port side to starboard side. The positive z-axis thus points from the top to the bottom of the ship.

## 2.3 Material Properties

The sonar dome is manufactured of AISI 316 L steel (Stoff-Nr. 1.4404), see Table 1. The material model used in the calculations is an elasto-plastic model with linear hardening for the shell elements and an elasto-plastic model without hardening for the beam elements.

Young's modulus [N/m <sup>2</sup> ]	2.0 10 <sup>11</sup>
Poisson ratio [-]	0.3
mass density [kg/m <sup>3</sup> ]	7980
0.2 % yield stress [N/m <sup>2</sup> ]	190.0 10 <sup>6</sup>
tangential stiffness [N/m <sup>2</sup> ]*	754.0 10 <sup>6</sup>

*Table 1 Material properties*

\*The tangential stiffness is calculated here as the difference between the tensile strength (490.0 10<sup>6</sup> N/m<sup>2</sup>) and the 0.2% yield stress divided by the failure strain (0.4). The actual tangential stiffness could be higher because it is very well possible that the maximum stress occurs at a lower strain, say 0.2. Unfortunately, the stress-strain curve is not available to verify this. This implies that we use a conservative approach here.

The impacting sphere is assumed to be rigid.

## 2.4 Boundary Conditions and Initial Conditions

The top the dome is mounted to the ship with bolts. Here, displacements and rotations of the dome are suppressed. Because the impact takes place near frame 170 (of the ship) it is assumed to be sufficient to model the dome as far as frame 166 (see drawing Z 01.10, August 1976, entitled 'Spantenlijst'). Behind frame 166 the dome is assumed to be rigid (the back of the dome lies against a wooden frame). The dome has a very stiff double bottom which is assumed to be rigid too.

On the vertical plane which cuts the dome in half in longitudinal direction, the displacements in y-direction and the rotations around the x-axis and the z-axis are suppressed.

The impacting rigid sphere moves in the horizontal xy-plane under an angle of 20 degrees with the projection in the horizontal xy-plane of the normal of the dome surface at the location of impact. This projected normal makes an angle of 50 degrees with the positive x-axis. The rigid sphere thus moves in the horizontal xy-plane under an angle of 70 degrees with the positive x-axis.

The sphere has a constant velocity of 10 m/s. In fact the velocity of the sphere is not of much importance in this analysis, because we aim for a quasi-static analysis. This means that the velocity of the sphere should be small enough to keep the amount of kinetic energy in the dome during impact low. At the same time the velocity should be as high as possible to ensure that the amount of cpu-time needed for the analysis remains acceptable.

## 2.5 Contact Algorithm

The contact algorithm which is used in the calculations is a penalty stiffness formulation as in surface to surface contact, see [2]. In the contact model friction was not taken into account.

### 3. Results

Based on Figure 4, the real permanent penetration depth after unloading is estimated to be 0.2 m. Some of the calculated results are presented in time history plots. The time can easily be converted into the penetration depth by multiplying the time with the penetration speed of 10 m/s. This means that in the calculations carried out below, the penetration depth increases linearly with respect to the time and there will be a time dependent contact force between the impactor and the dome: unloading is not carried out. This means that the permanent penetration depth (plastic deformation) will be somewhat lower than the calculated penetration depth (elastic and plastic deformation).

#### 3.1 Deformations

Figure 15 and Figure 16 show the deformation of the inside of the dome at times  $t = 0.02$  s and  $t = 0.03$  s respectively. The deformation patterns at these two times are not significantly different. The upper horizontal primary stiffener is tilted over three vertical secondary stiffener distances. Secondary stiffeners buckle locally (especially the horizontal ones near the fourth vertical secondary stiffener). This deformation pattern corresponds remarkably well with the deformation pattern visible in Figure 6. The vertical primary stiffener is buckled near the lower horizontal primary stiffener.

Figure 17-Figure 20 show the contour plots of the displacement in x, y and z-direction and the total displacement for  $t = 0.02$  s respectively. The damaged area is approximately circular with a radius of 0.4 m, which is the same radius as calculated from Figure 4. Therefore the situation at  $t = 0.02$  s will be investigated in detail. The centre of the damaged area has a maximum displacement of 0.2 m.

Figure 21 shows the effective plastic strain at  $t = 0.02$  s. The maximum effective plastic strain is 0.17 and occurs in the horizontal secondary stiffener just below the region of impact.

In Figure 7 and Figure 8 cracks are visible which are not predicted by the model. The real structure is chain welded. In the model continuous welding is assumed.

Figure 22 gives the total displacement for  $t = 0.03$  s.

#### 3.2 Stresses

Figure 23-Figure 25 show the stresses in x, y and z-direction respectively for  $t = 0.02$  s. In the region where there is direct contact between the sphere and the dome we see tensile stresses up to 314 MPa. Just outside the region of contact we see compressive stresses up to -242 MPa.

Figure 26 shows the Von Mises stress for  $t = 0.02$  s. The maximum Von Mises stress is 321 MPa and is found in the secondary stiffeners in the region of contact.

### 3.3 Forces

In order to check the order of magnitude of the calculated forces two analytical formulas were applied: Roark's formula and Lupker's formula.

#### 3.3.1 Roark's formula

In [4] an analytical expression is given for a concentrated transversal force  $F_R$  acting on the centre of a flat circular elastic plate (radius  $r$ ) with a simply supported edge as function of the transversal displacement  $y$  of the centre:

$$F_R = \frac{16\pi D y}{r^2} \frac{1+\nu}{3+\nu}$$

where:

$$D = \frac{Et^3}{12(1-\nu^2)}$$

and with  $\nu = 0.3$ ,  $E = 2.0 \cdot 10^{11}$  N/m<sup>2</sup>,  $t = 0.006$  m (equivalent thickness due to correction for secondary stiffeners) and  $r = 0.4$  m (radius of damaged area).

For a deflection of  $y = 0.2$  m the transversal force  $F_R$  becomes 100 kN.

#### 3.3.2 Lupker's formula

In [5] a similar force-deflection relationship is derived where a localised zone of plastic deformation and a spherical profile is taken into account:

$$F_L = \frac{8}{3} \pi M_0 \left[ \sqrt{\{(1-6P_0)ky/t\}} + P_0 ky/t \right]$$

where:

$$M_0 = \sigma_0 t^2 / 4$$

and:

$$P_0 = pr / (2\sigma_0 t)$$

Using the values given in the previous section and  $\sigma_0 = 190 \cdot 10^6 \text{ N/m}^2$  (maximum stress elastic perfectly plastic material model),  $k = 3$  (spherical profile),  $p = 0 \text{ N/m}^2$  (internal overpressure), a deflection of  $y = 0.2 \text{ m}$  requires a force  $F_L = 140 \text{ kN}$ .

For an internal overpressure  $p$  of 1 atmosphere ( $10^5 \text{ N/m}^2$ )  $F_L$  increases to 200 kN.

### 3.3.3 LS-DYNA3D calculation

Figure 27 and Figure 28 show the force magnitude and the force in global y-direction as a function of time. From these figures it becomes clear that the force in global y-direction is dominant for the force magnitude. At  $t = 0.02 \text{ s}$  the penetration depth is about 0.2 m and the contact force is 230 kN, which is somewhat higher than but in the same order of magnitude as the forces following from Roark's formula and Lupker's formula. Probably it is somewhat higher because the bending stiffness of the stiffeners is underestimated in the formulas; moreover the primary stiffeners have been replaced by secondary stiffeners in the calculation of the equivalent thickness.

The force does not seem to increase much more at  $t = 0.03 \text{ s}$ .

## 3.4 Energies

Figure 29-Figure 33 respectively show the external work, the internal energy, the kinetic energy, the hourglass energy and the total energy of the dome as a function of time. The total energy is equal to the sum of the internal energy, the kinetic energy and the hourglass energy. Note that the external work should be equal to and in fact equals the total energy. Note further that at  $t=0.03\text{s}$  the kinetic energy is about 5% of the total energy and the hourglass energy is about 2% of the total energy. This means that 93% of the external work is transformed in deformation (internal) energy (=summation of elastic and plastic energy). The deformation energy is 32.5 kJ for  $t = 0.02 \text{ s}$  and 51 kJ for  $t = 0.03 \text{ s}$ . At this point we remind the reader that the approach we used to calculate the tangential stiffness is conservative: the actual tangential stiffness could be considerably higher (see page 6). In the latter case the deformation energy will also be higher, maybe even up to 25%.

At  $t = 0.03 \text{ s}$  the internal energy seems to increase almost linearly.

Figure 34 shows a contour plot of the internal energy density, which makes clear to which extent the several parts in the structure contribute to the internal energy.

## 4. Conclusions and Recommendations

The main conclusions are:

1. The calculated deformation pattern corresponds very well with the deformation pattern found in reality. Both show global tilting of the top horizontal primary frame over three vertical secondary frame distances. Also the places at which local buckling of secondary stiffeners occurs correspond well. The radius of the damaged area is in reality equal to 0.4 m. In the calculation we have this situation for  $t = 0.02$  s (penetration depth 0.2 m). The only (minor) difference between the calculation and reality is that in the last situation a right-angled dent is visible, which is absent in the calculation.
2. Based on the reported assumptions, we find for a penetration depth of 0.2 m a contact force of 230 kN and a corresponding deformation (internal) energy of 32.5 kJ. This is 20% of the energy (165 kJ) of an anchor (2100 kg) falling from a height of 8 m [3].

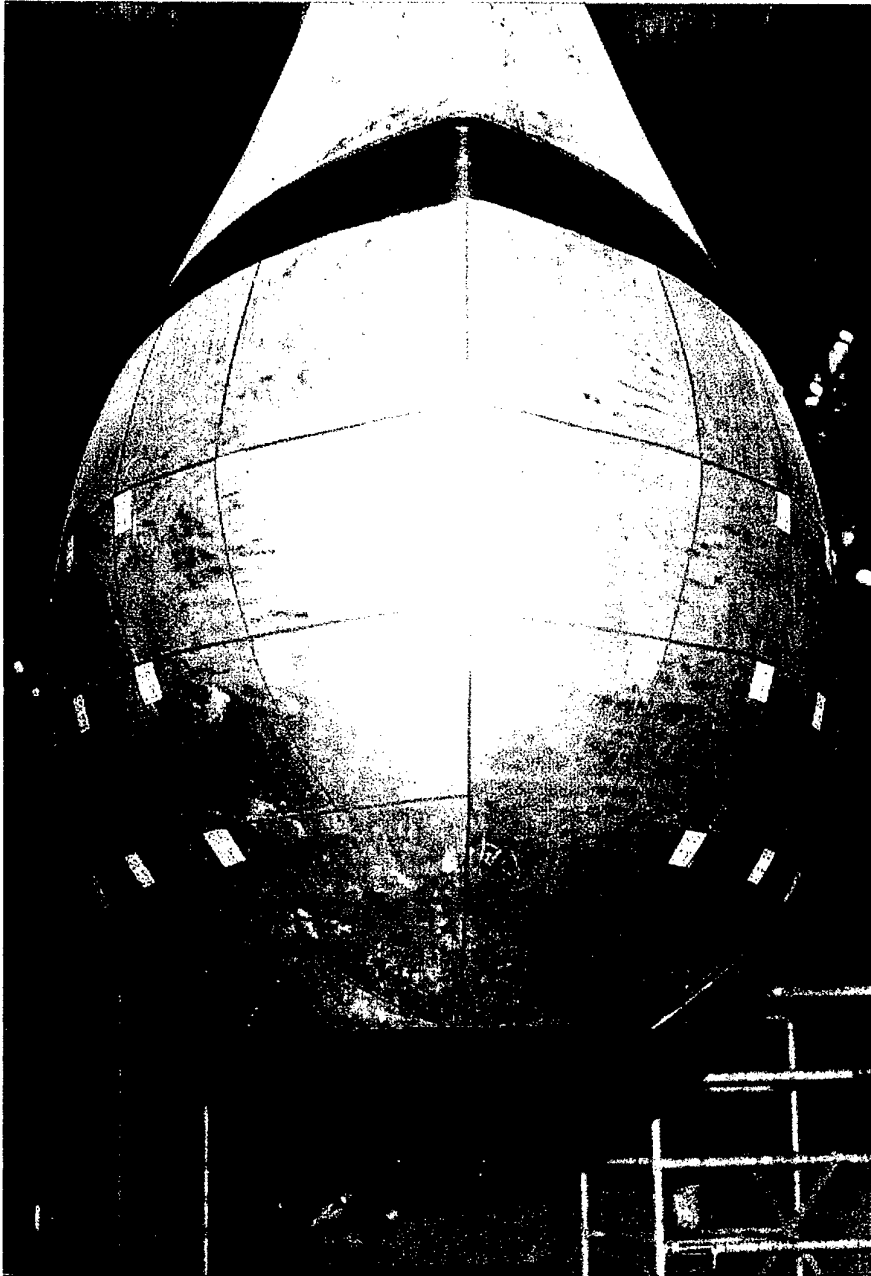
Recommendations:

1. It would be interesting to carry out a calculation for a composite dome under the same impact conditions in order to compare the resultant deformations, stresses, forces and energies.

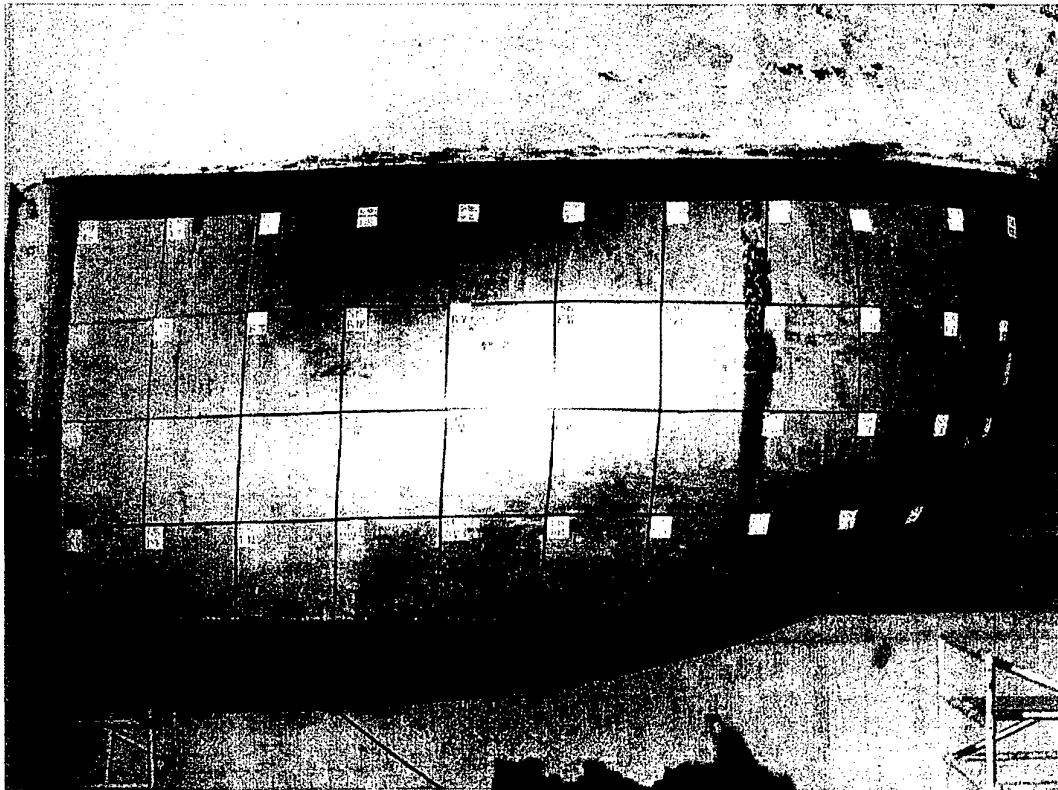
## References

- [1] LS-DYNA3D User's Manual (Nonlinear Dynamic Analysis of Structures in Three Dimensions), Version 936, August 1995.
- [2] LS-DYNA3D Theoretical Manual, LSTC report 1018 Rev. 2, July 1993.
- [3] G.M.H. Graafmans, Materiaal en Impact Eigenschappen van de Sonar Dome van het M-Fregat, S.B. 11056, DMKM/SCHEBO/SO&O, March 1997 (in Dutch).
- [4] R.J. Roark and W.C. Young, Formulas for stress and strain, McGraw-Hill, 1975.
- [5] H.A. Lupker, LPG Rail Tank Cars Under Head-On Collisions, Int. J. Impact Engng, Vol. 9, No. 3, pp. 359-376, 1990.

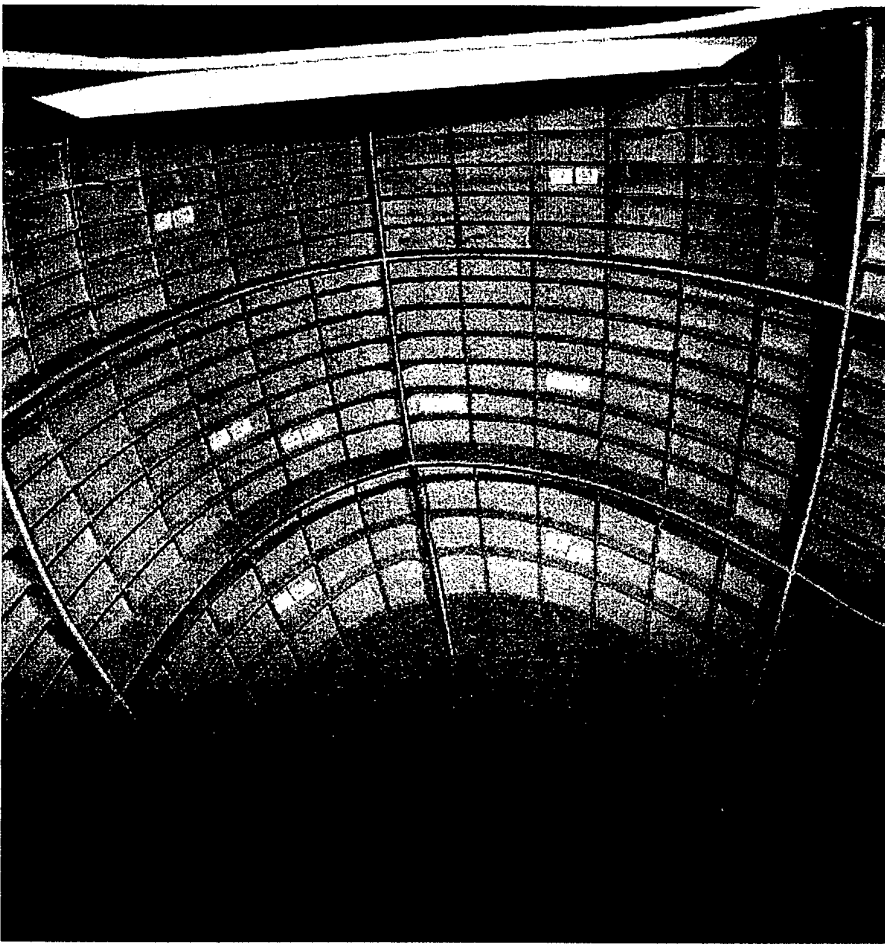
## Figures



*Figure 1 Front view of the undamaged dome*



*Figure 2 View of starboard side of the undamaged dome (squares are numbered A to D from top to bottom and I to XI from left to right)*



*Figure 3 Inside front view of the undamaged dome*



Figure 4 Damaged surface has approximately a circular shape with a right-angled dent in it.

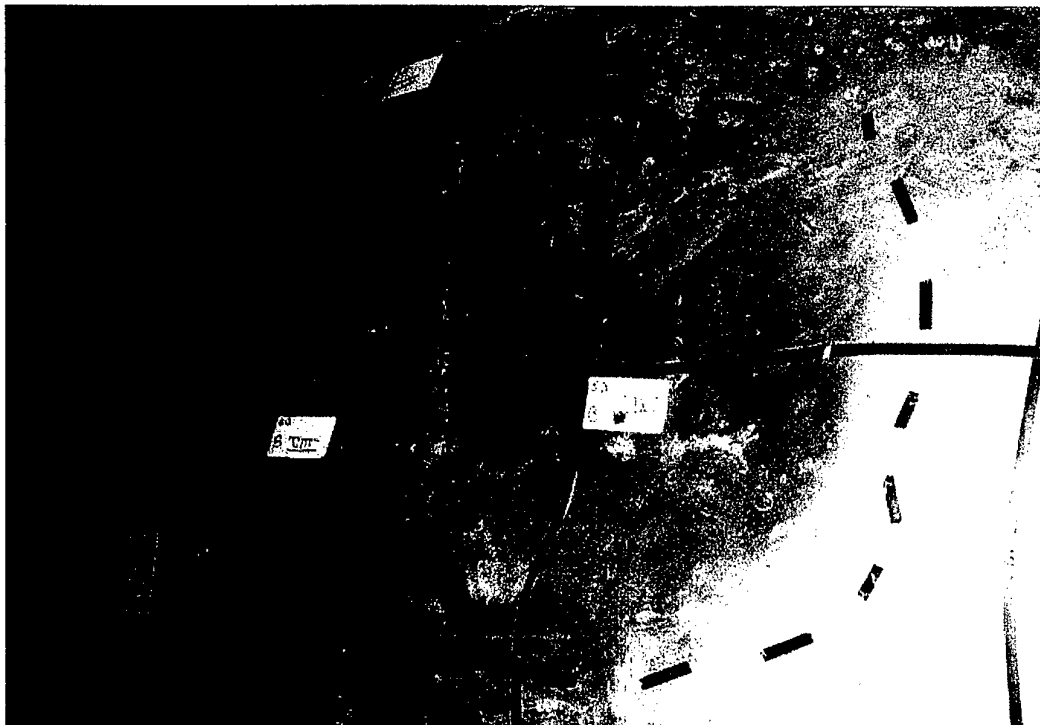
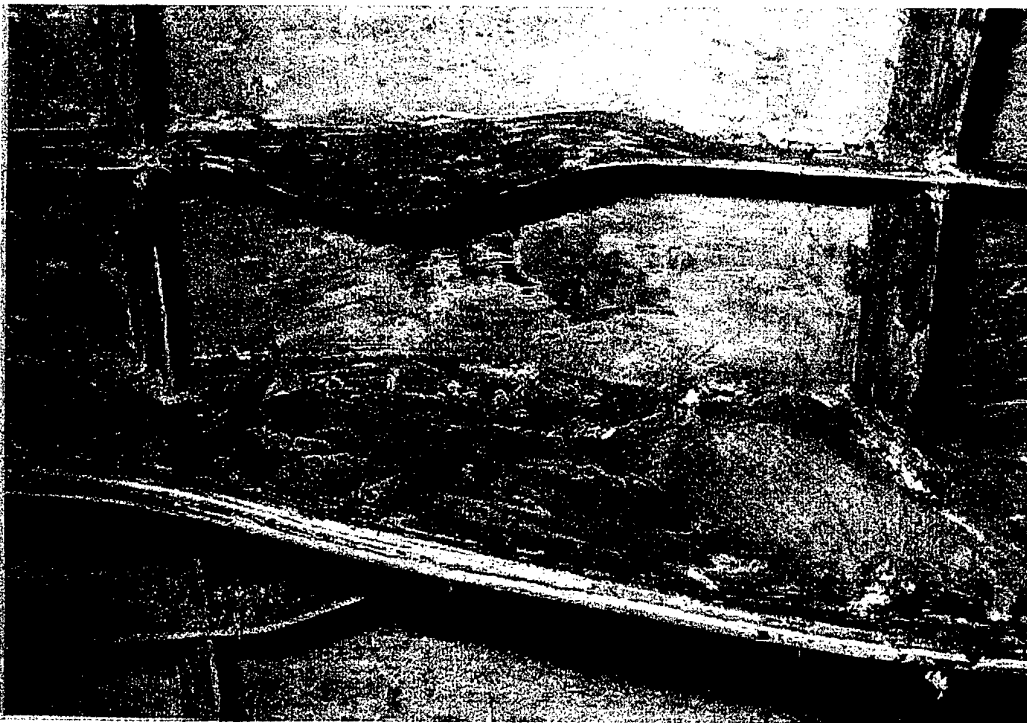


Figure 5 Outside view of damage



*Figure 6 Inside view of damage: tilting of horizontal primary stiffener and local buckling of horizontal secondary stiffeners*



*Figure 7 Crack in connection between hull and longitudinal stiffener*



*Figure 8 Crack in stiffener connection*

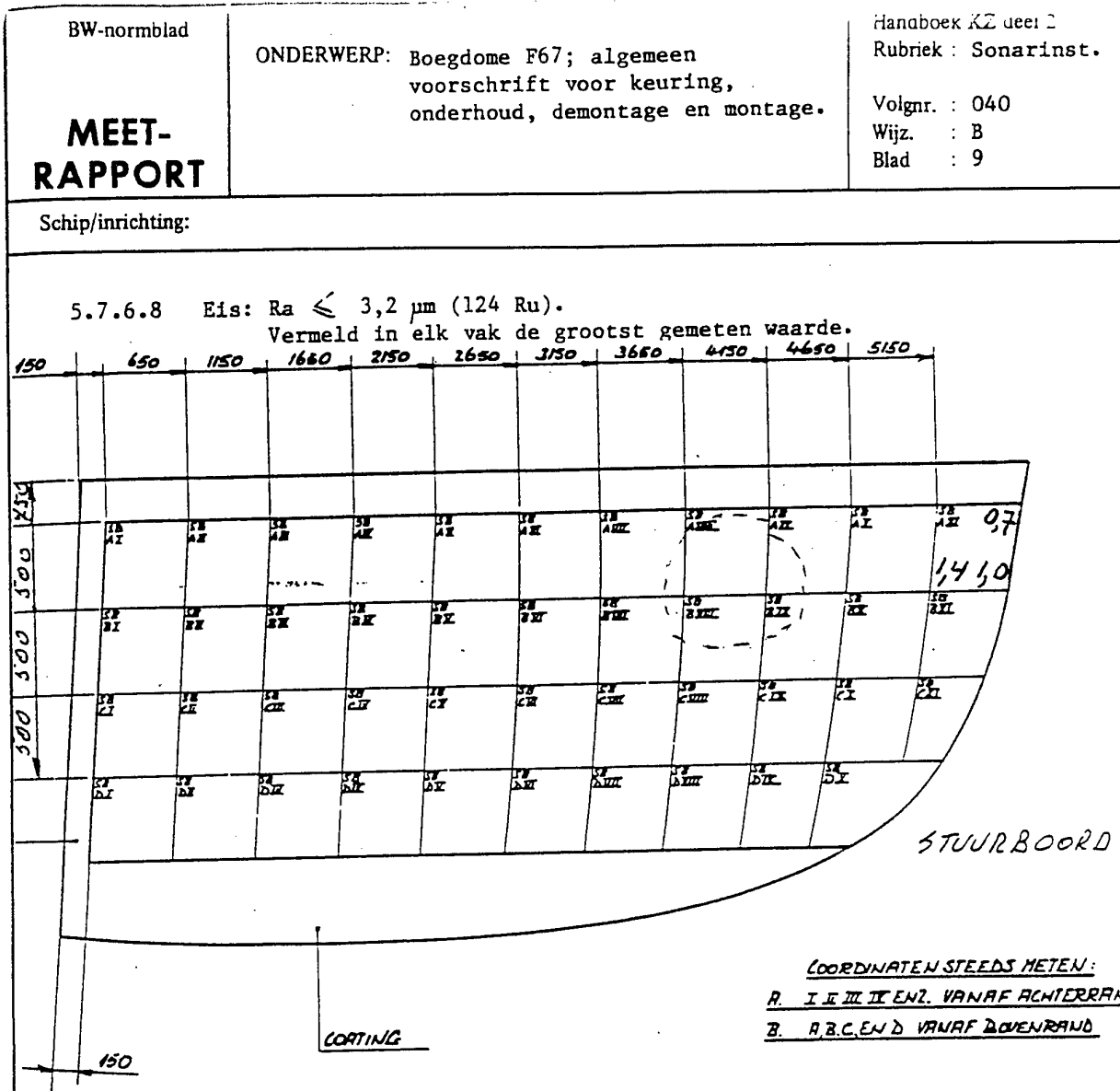


Figure 9 Damaged region of the dome is marked by the dotted curve



Figure 10 Mesh of the dome seen from the bottom in xy-plane (impacting sphere in blue)

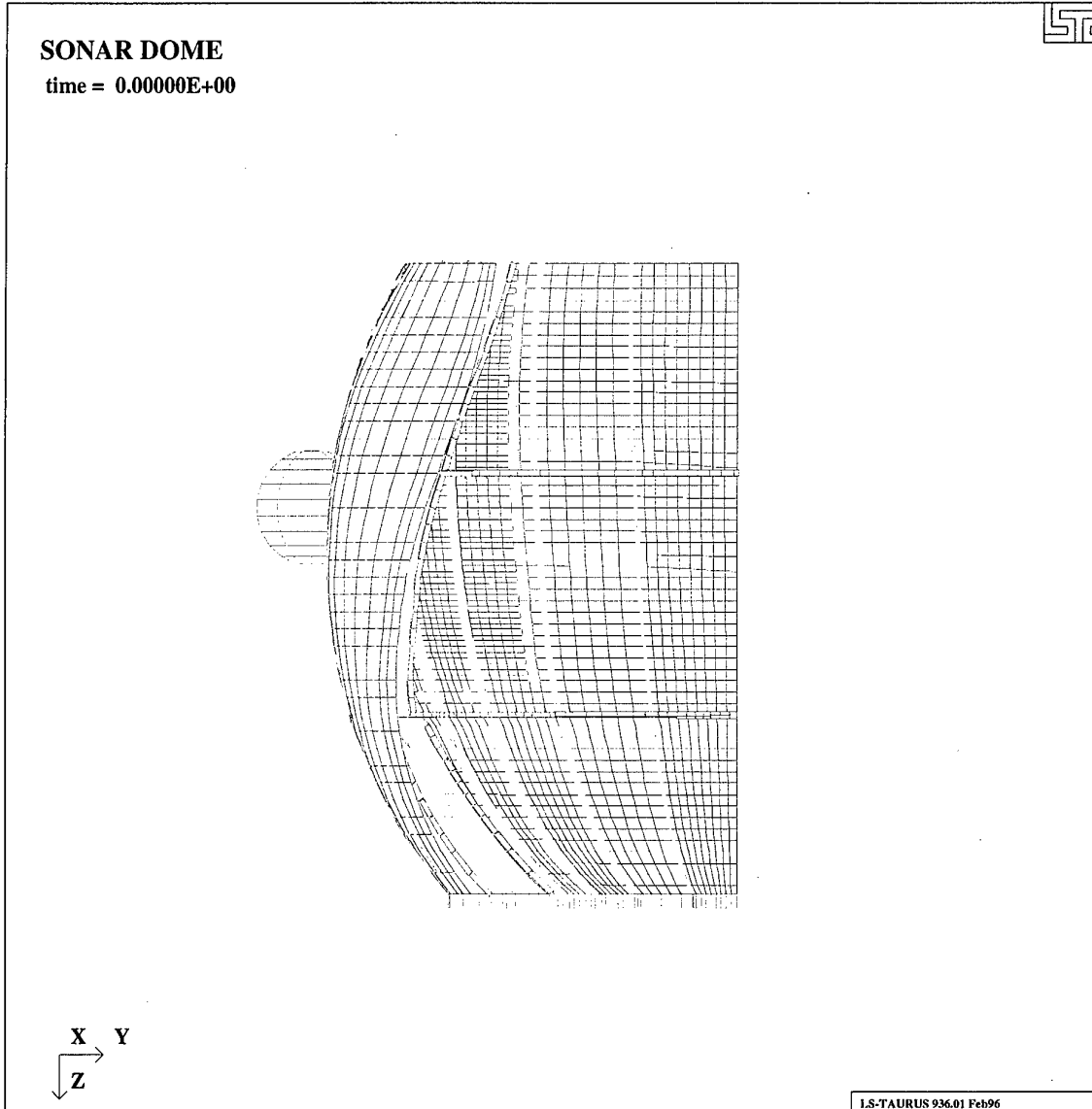


Figure 11 Mesh of the dome seen from the aft in the yz-plane

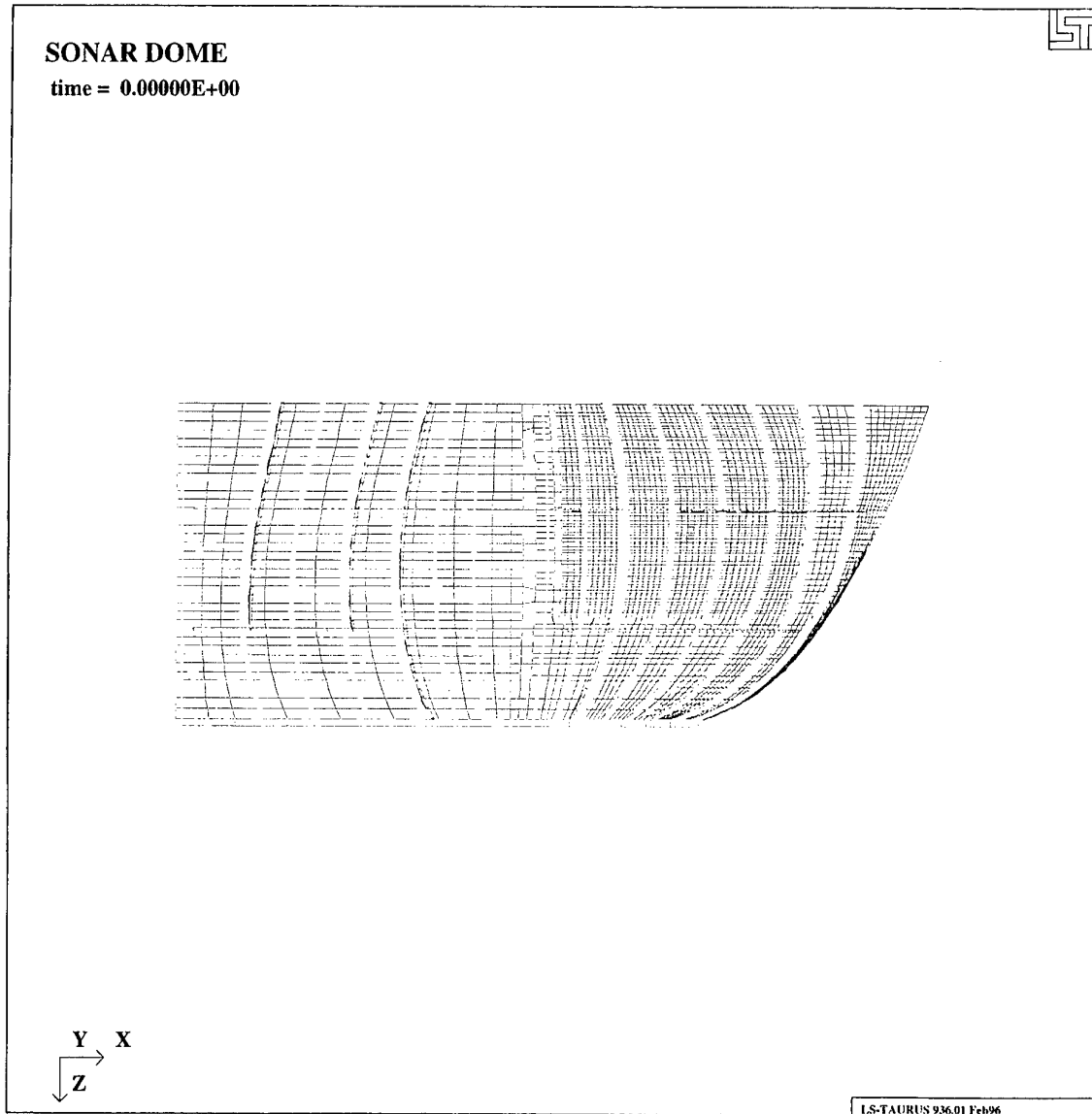


Figure 12 Mesh of the dome seen from the inside in the xz-plane

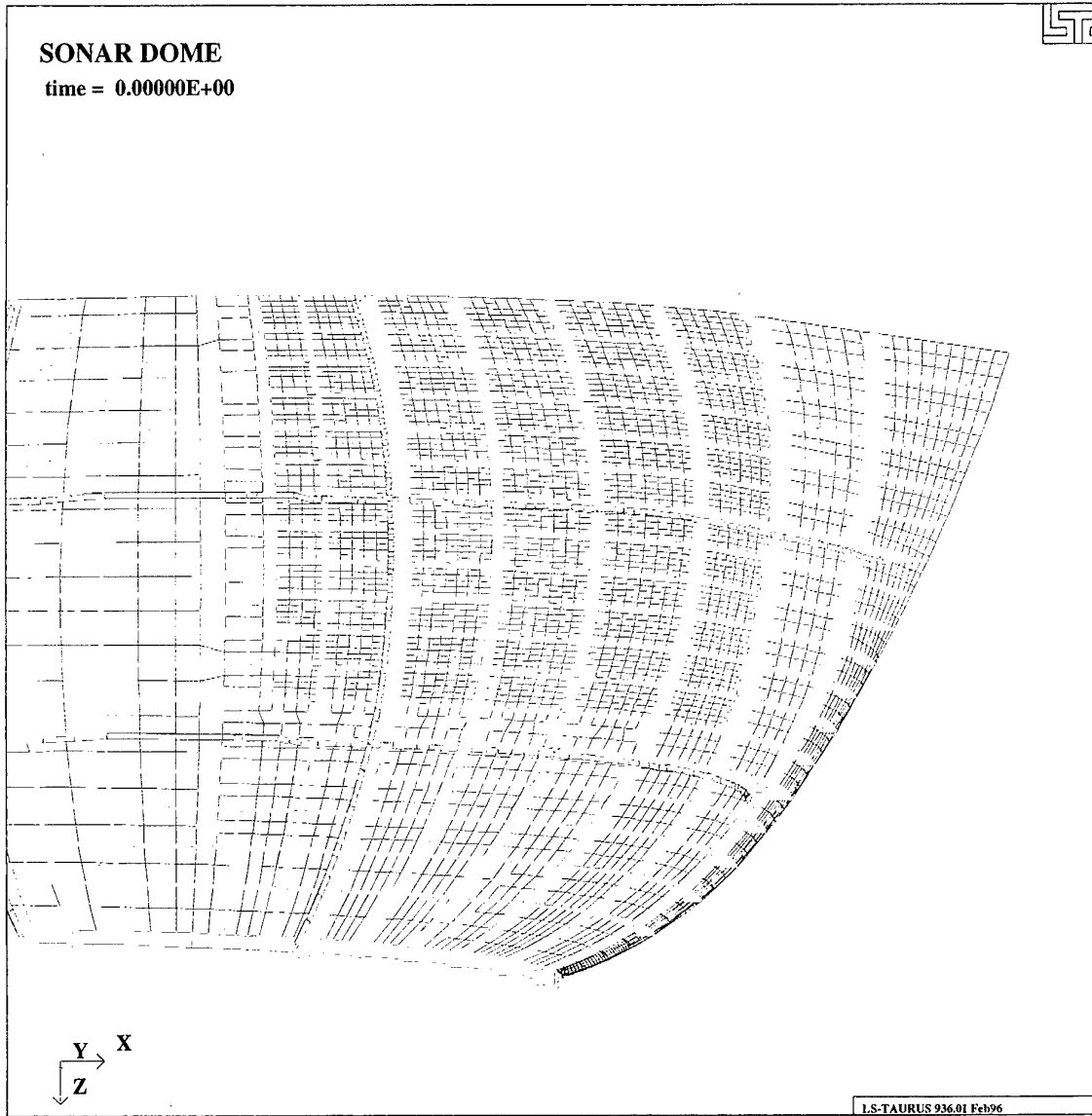


Figure 13 Mesh of the dome seen from the inside near the region of impact

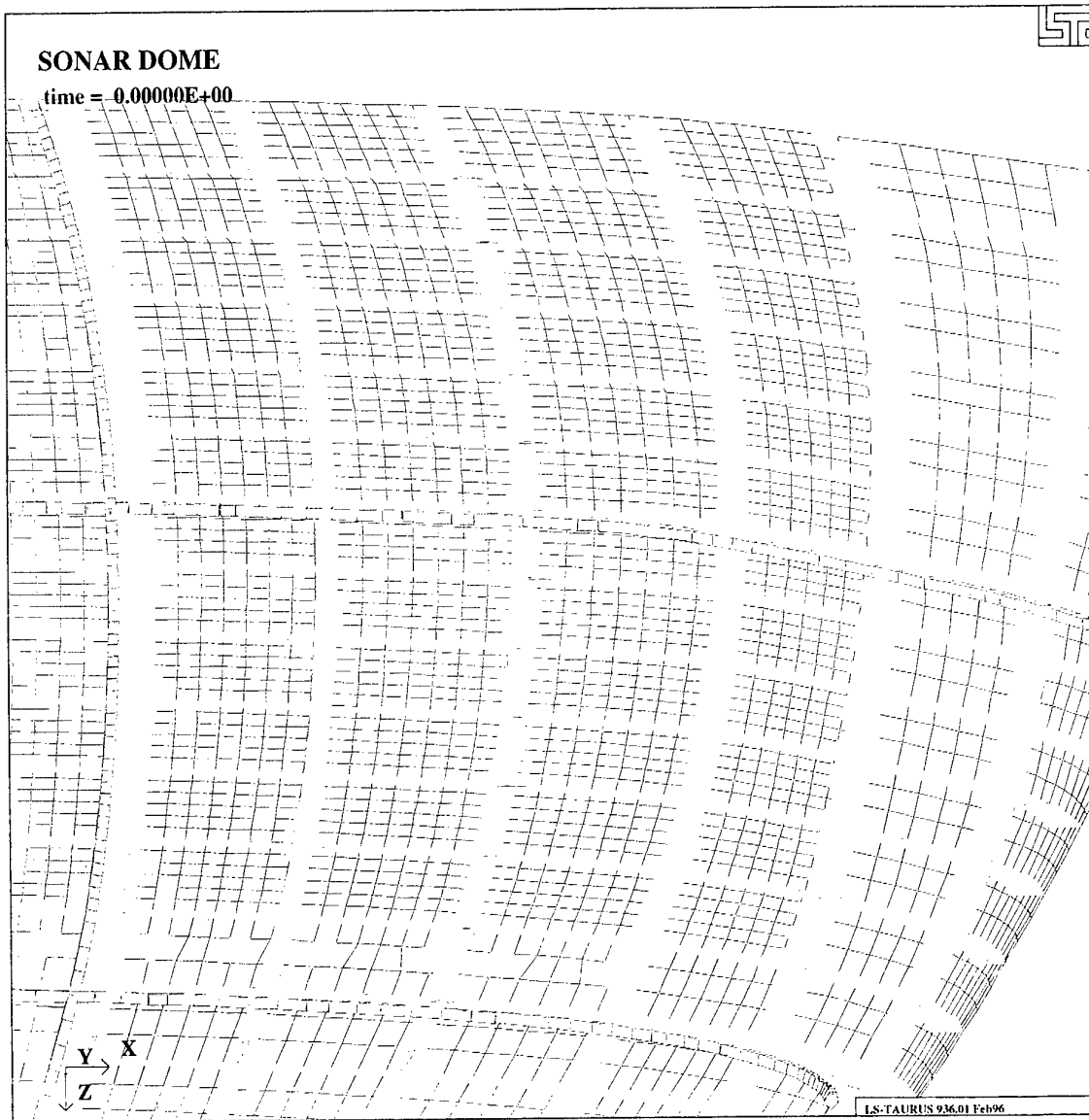


Figure 14 Further detail of the mesh of the dome near the region of impact

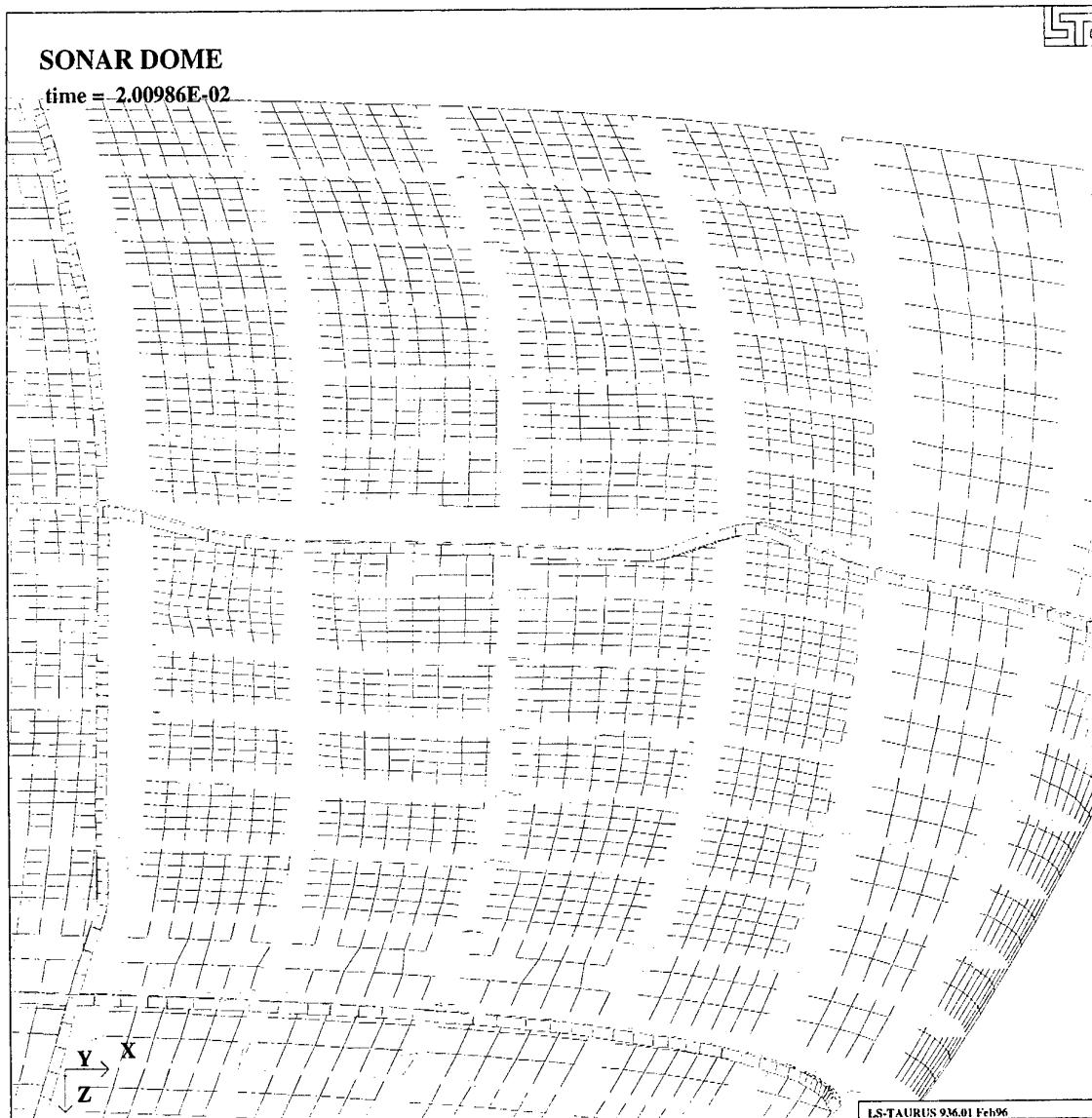


Figure 15 Deformed geometry seen from the inside at  $t=0.02$  s

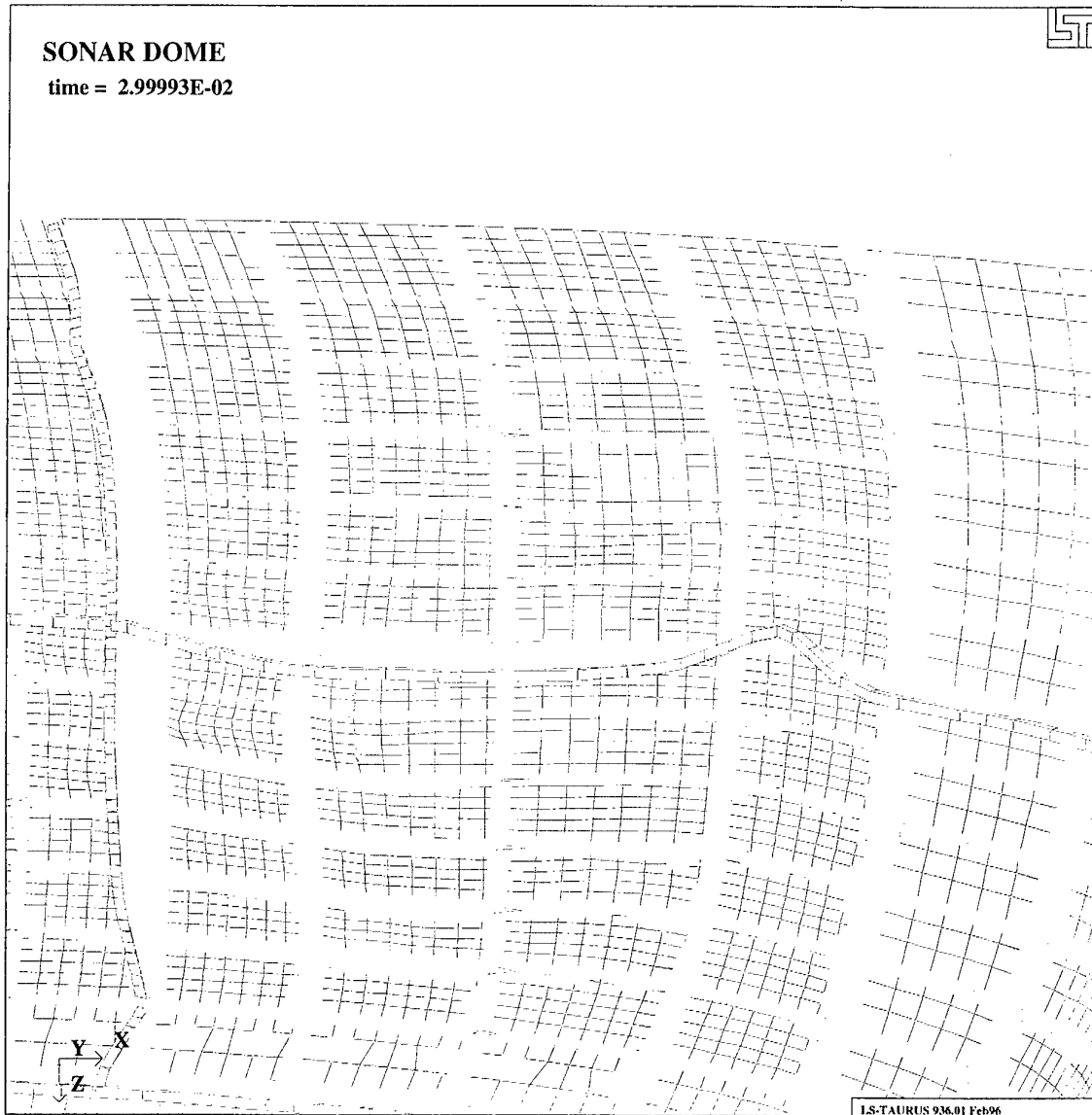


Figure 16 Deformed geometry seen from the inside at  $t=0.03$  s

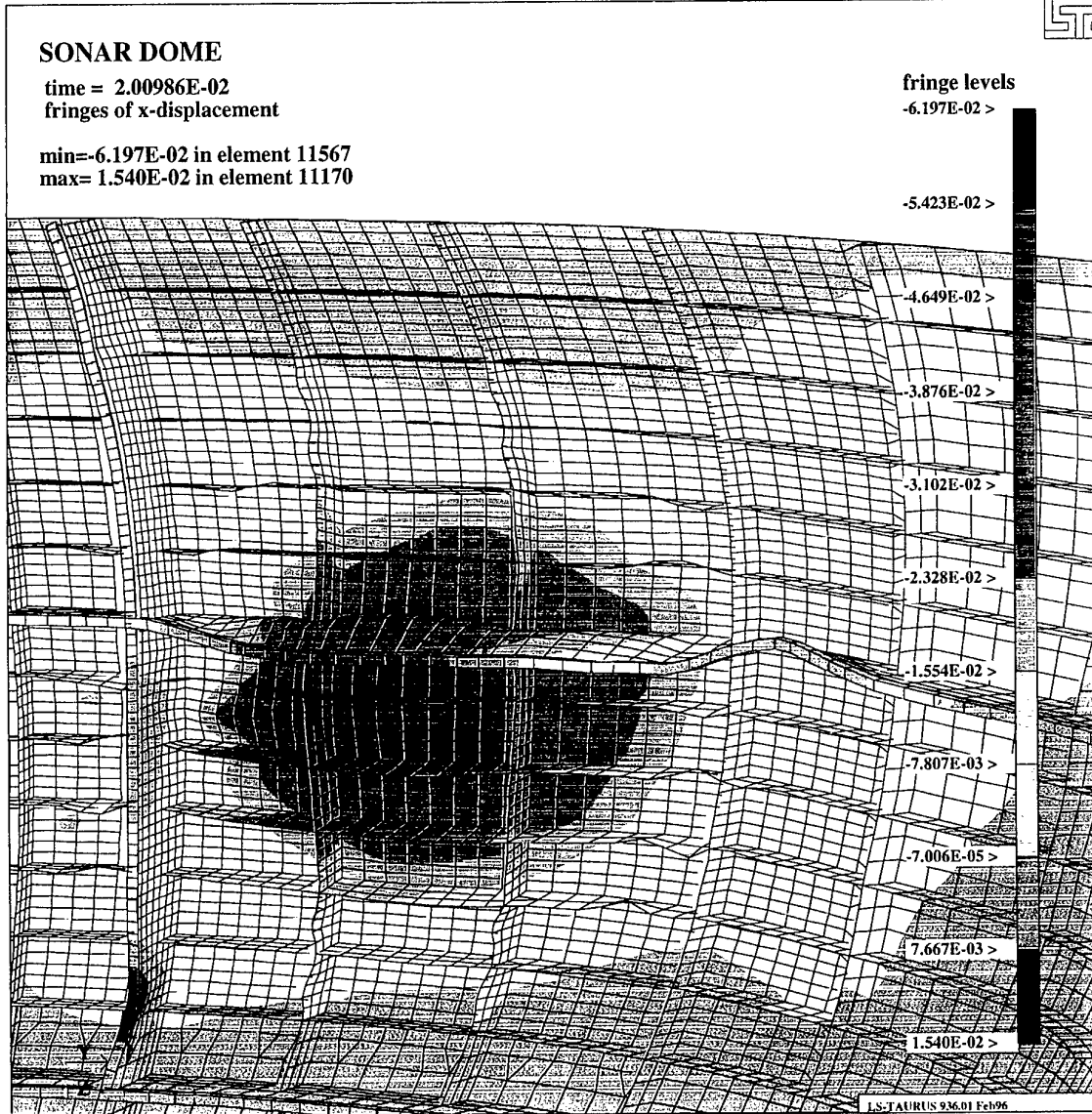


Figure 17 x-displacement [m] at t=0.02 s

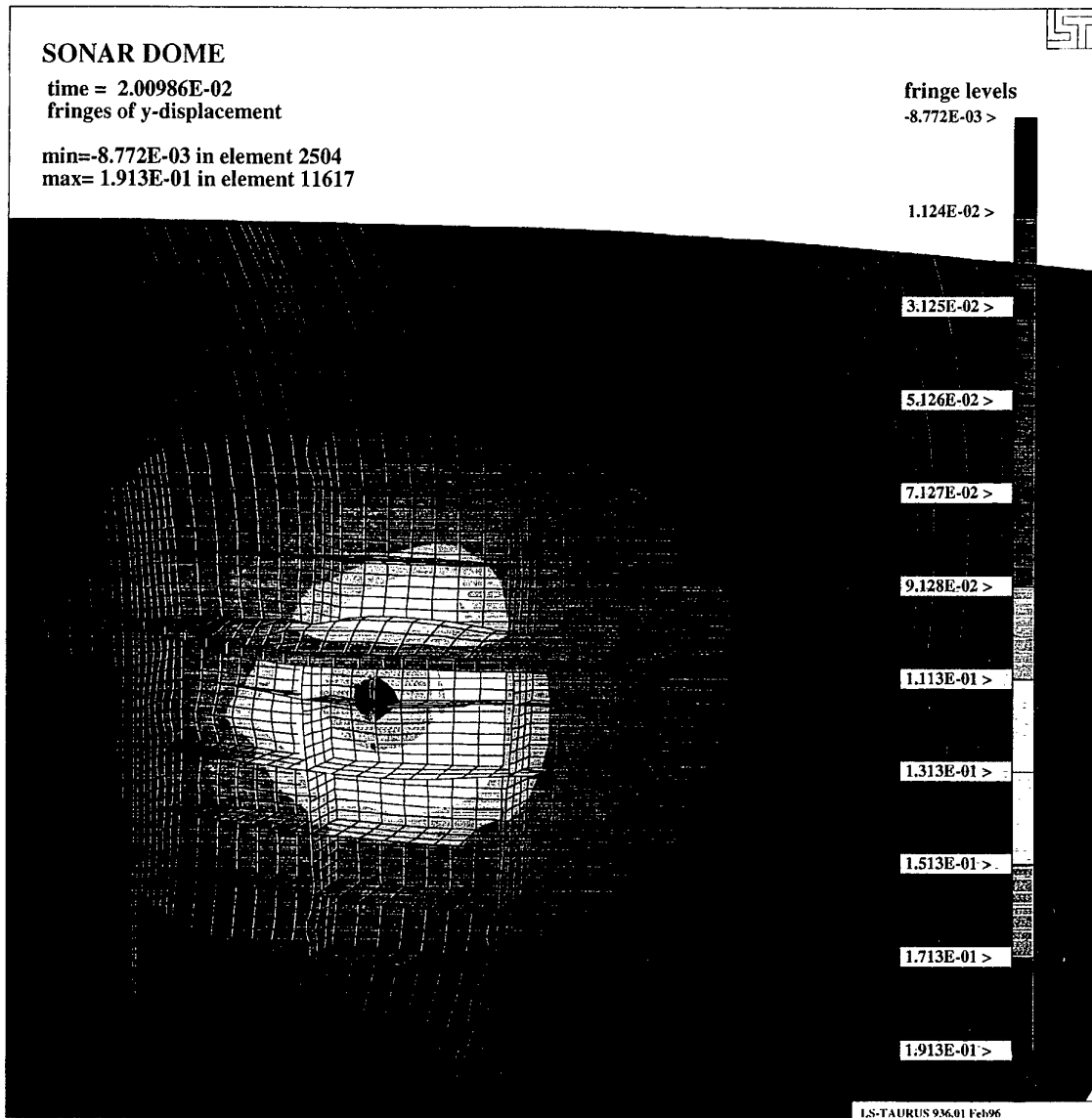


Figure 18 y-displacement [m] at t=0.02 s

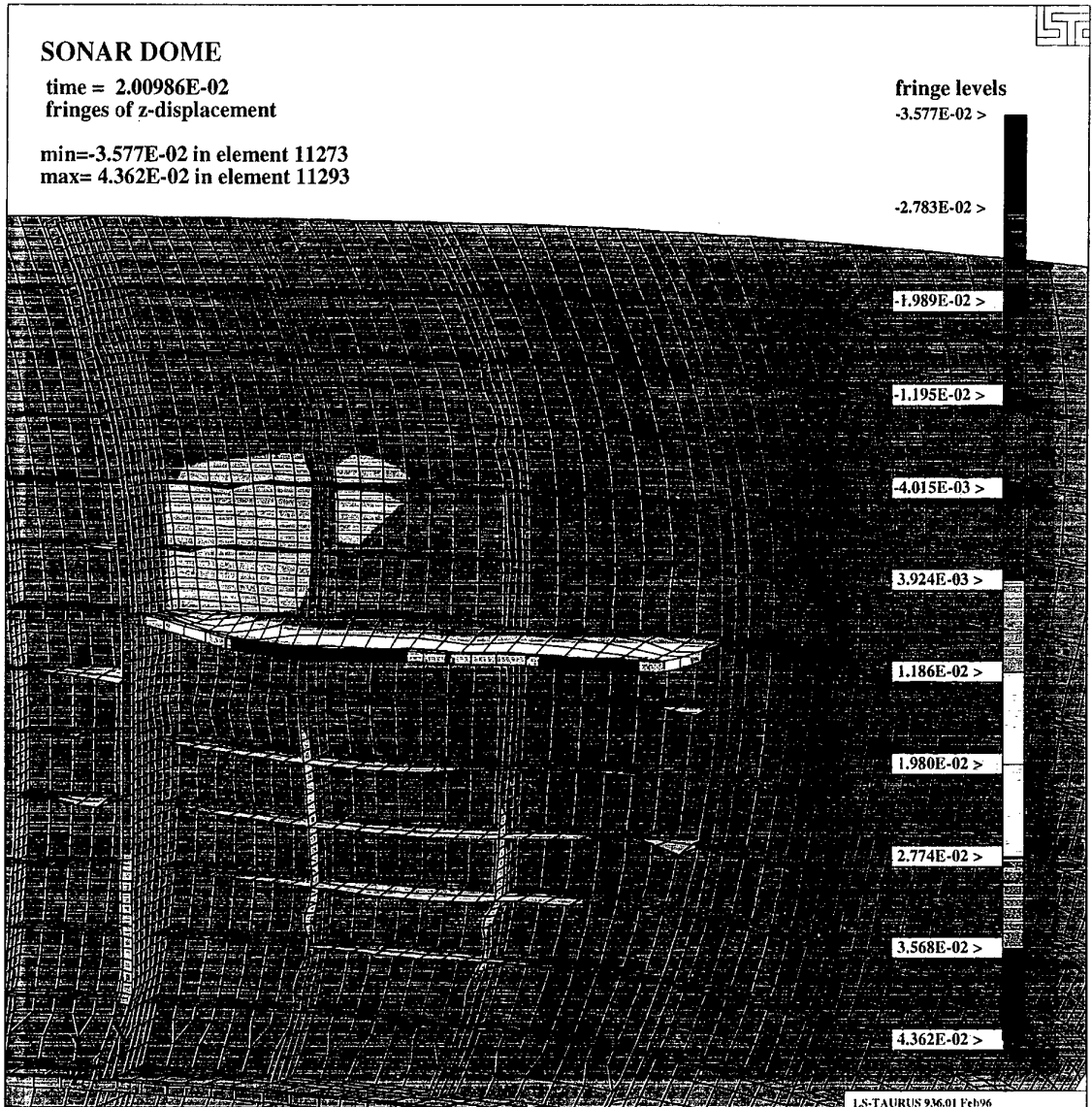


Figure 19 z-displacement [m] at t=0.02 s

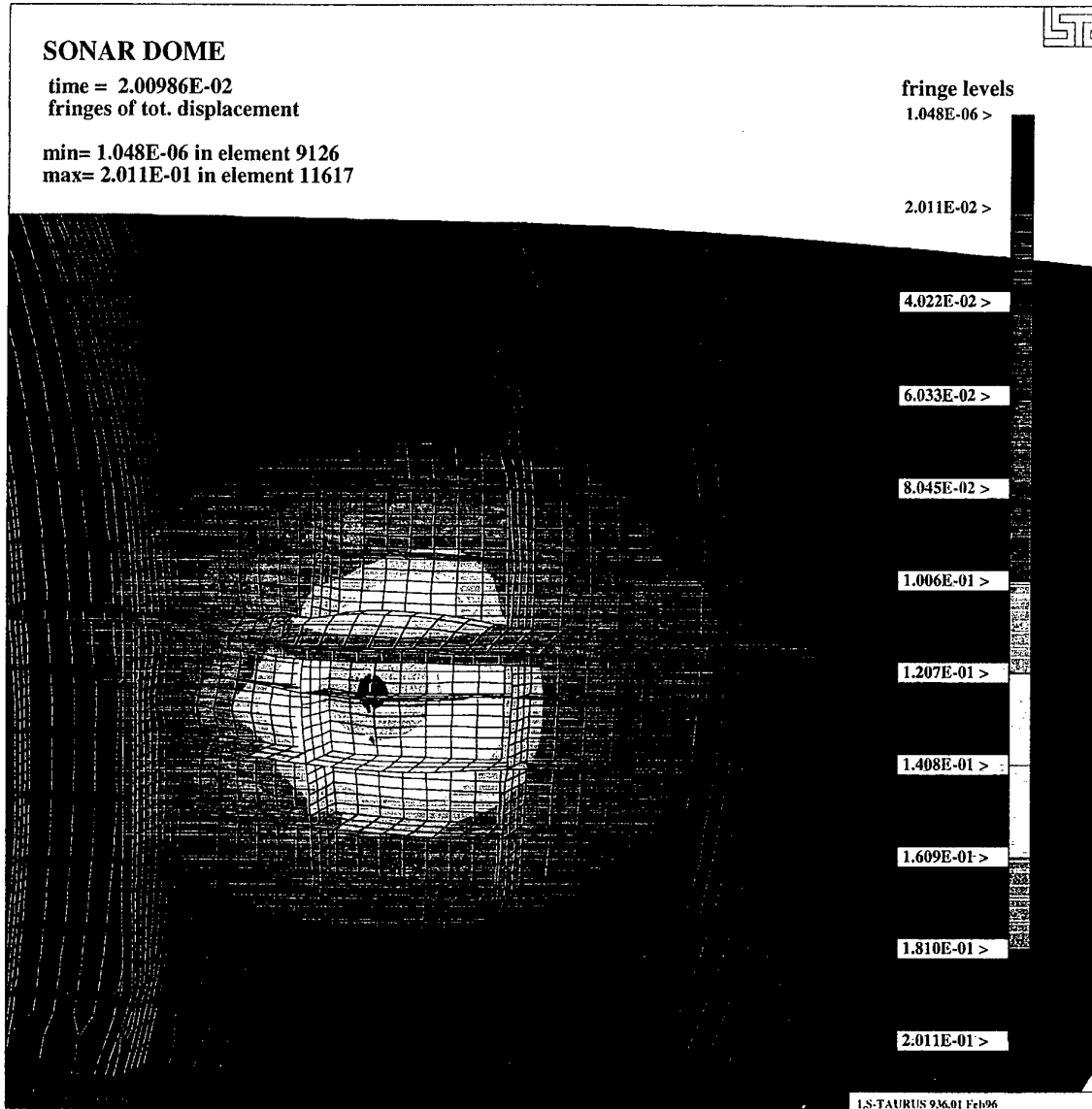


Figure 20 Total displacement [m] at t=0.02 s

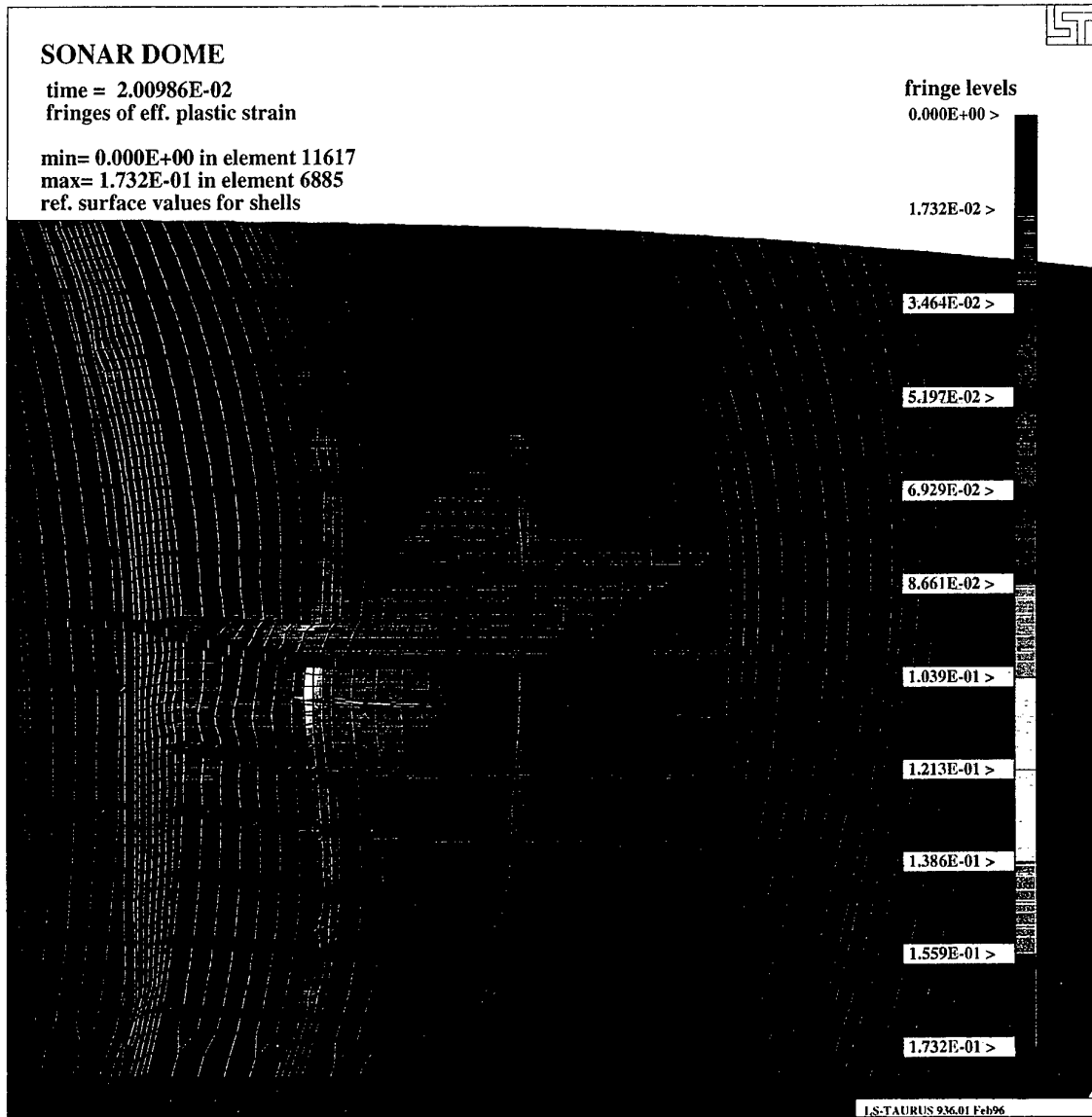


Figure 21 Effective plastic strain [-] at t=0.02 s

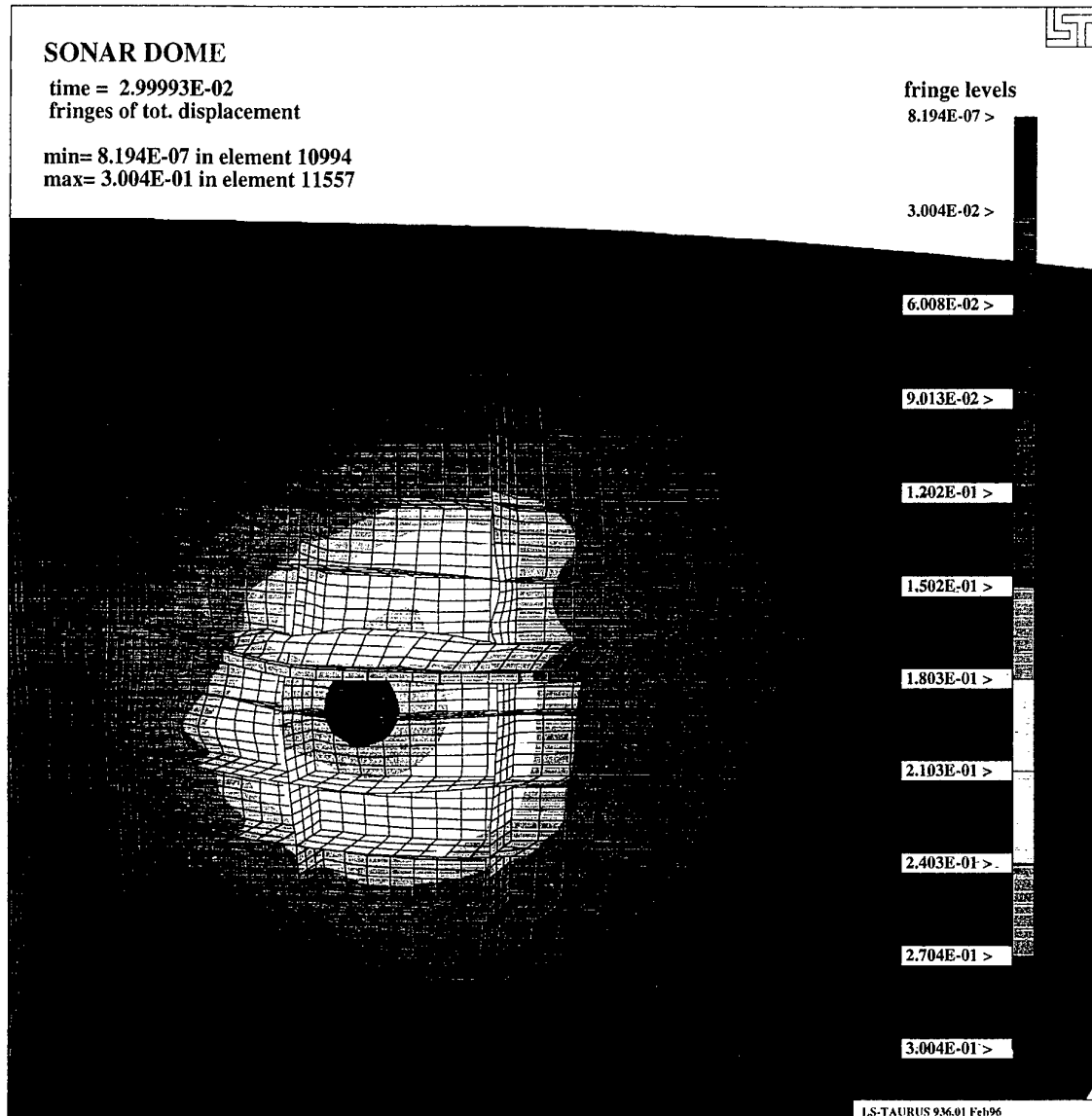


Figure 22 Total displacement [m] at t=0.03 s

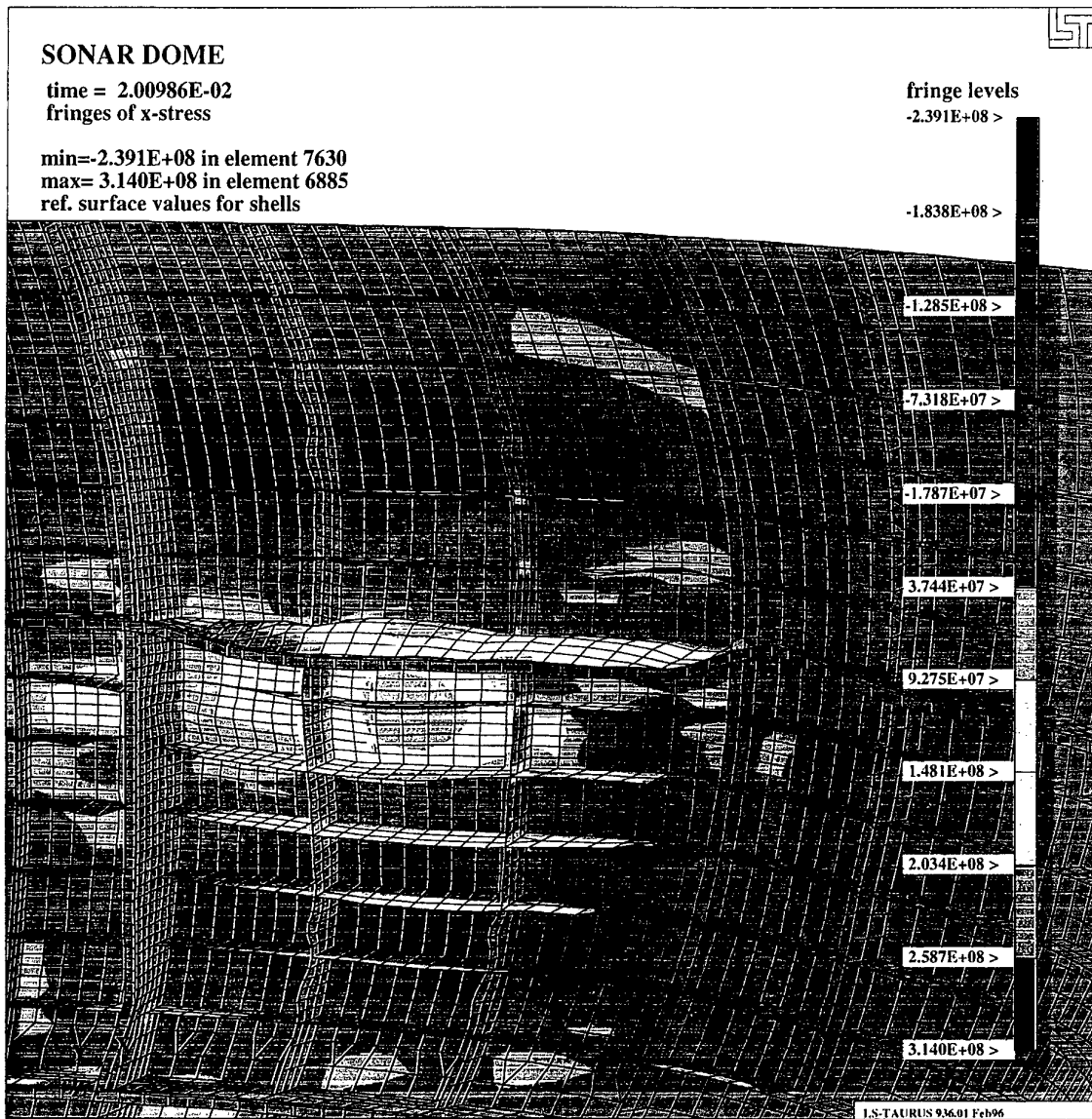


Figure 23 x-stress [Pa] at t=0.02 s

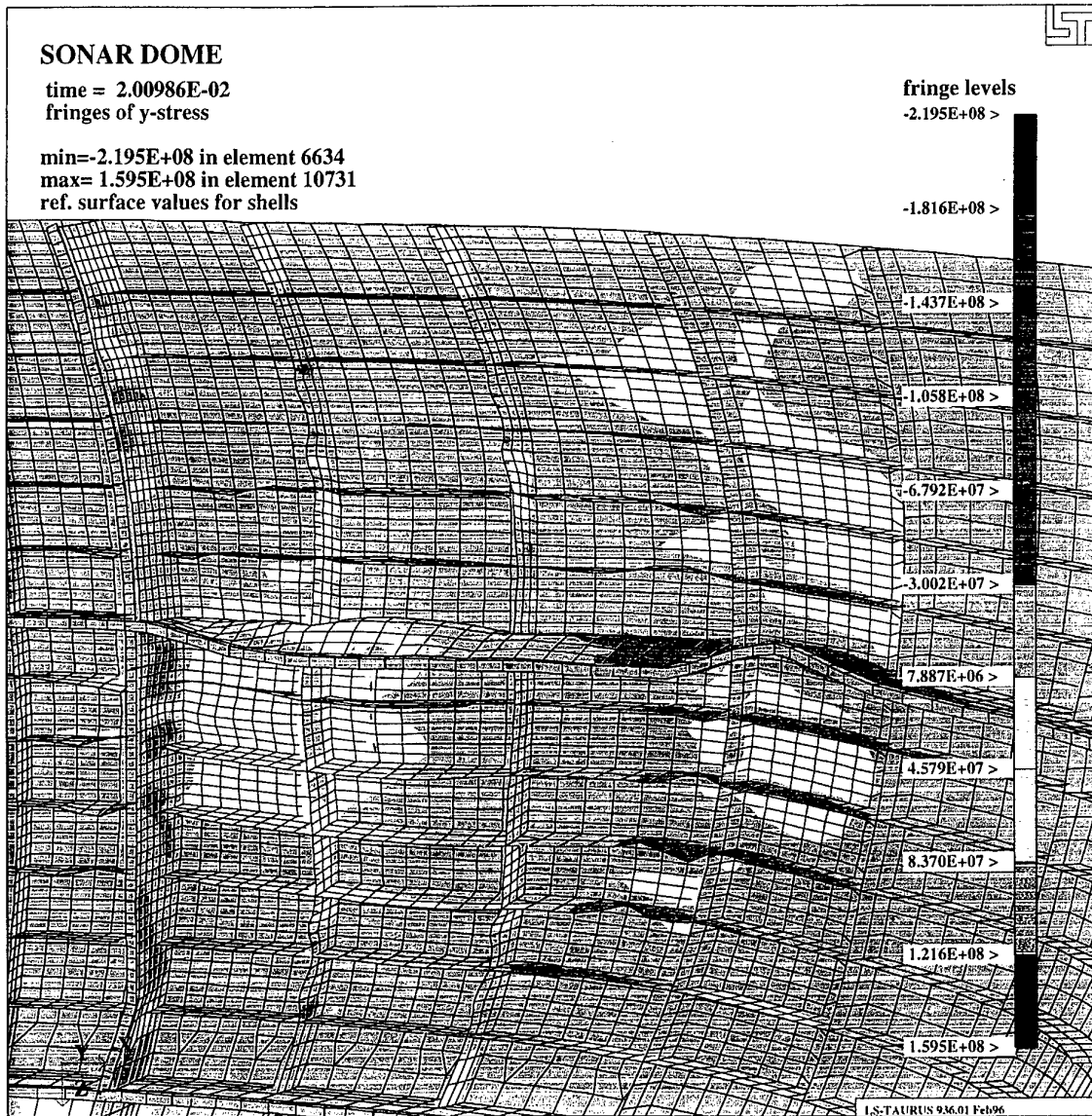


Figure 24 y-stress [Pa] at t=0.02 s



Figure 25 z-stress [Pa] at t=0.02 s

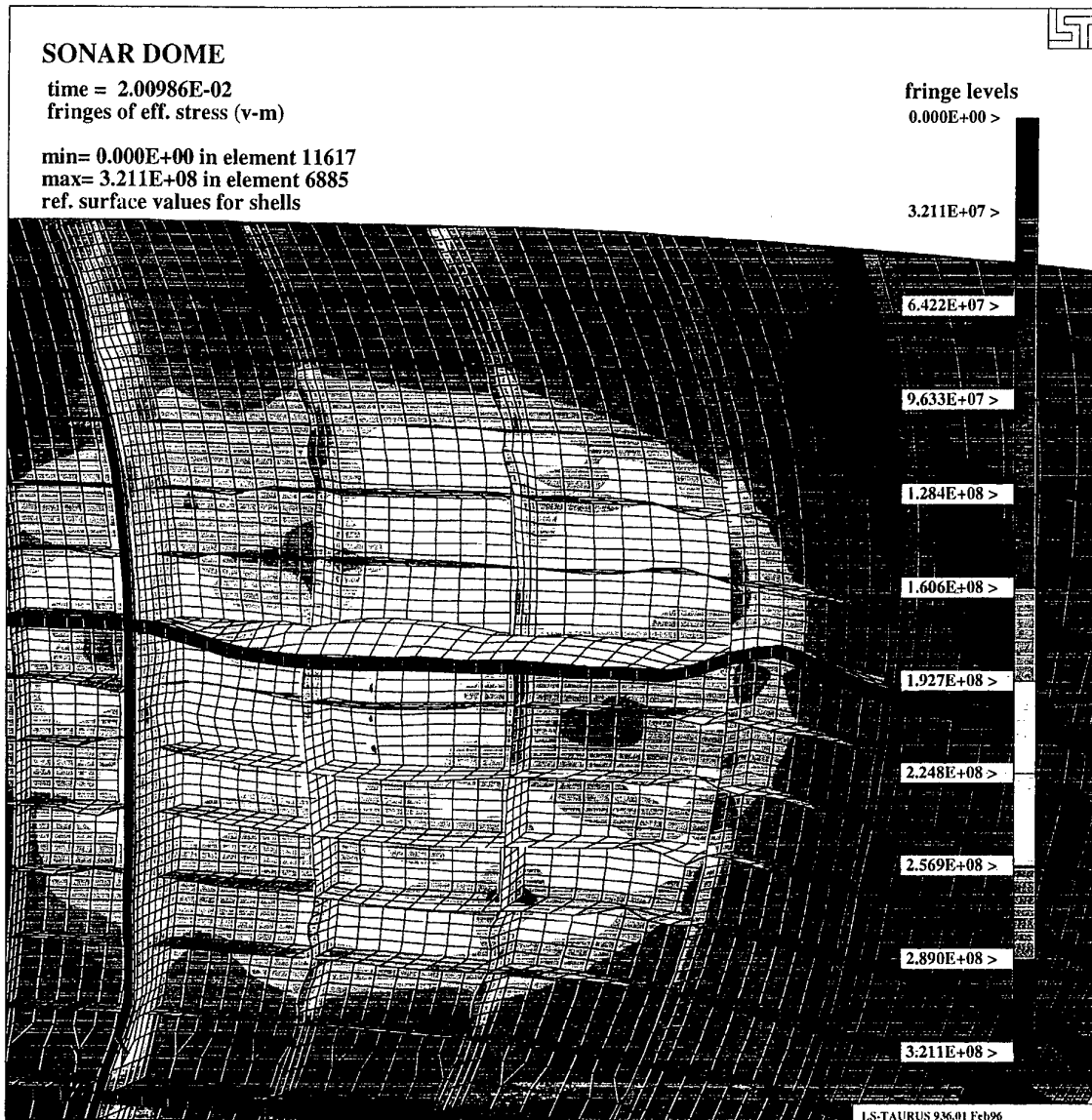


Figure 26 Von Mises stress [Pa] at t=0.02 s

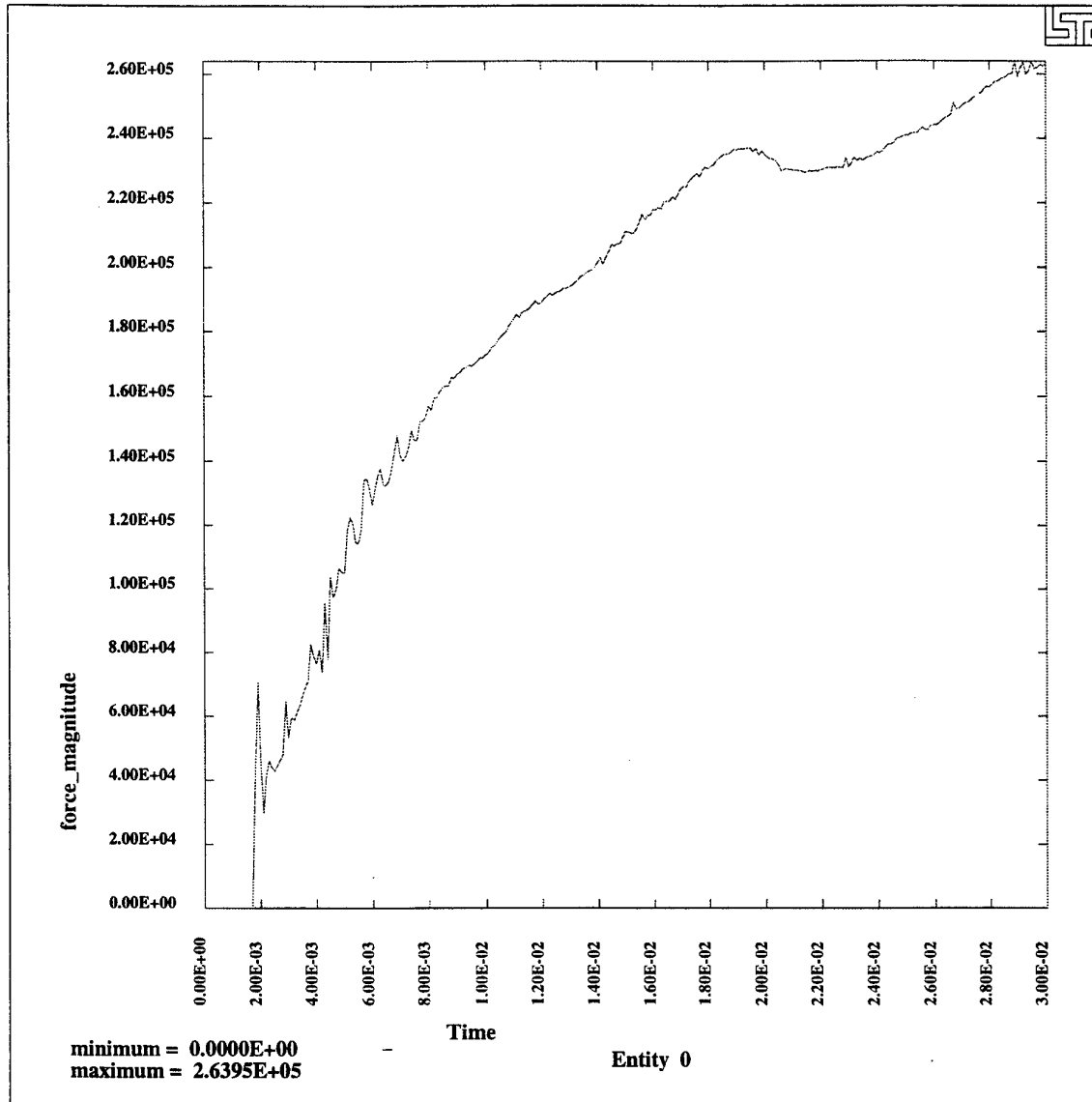


Figure 27 Total contact force [N] versus time [s]

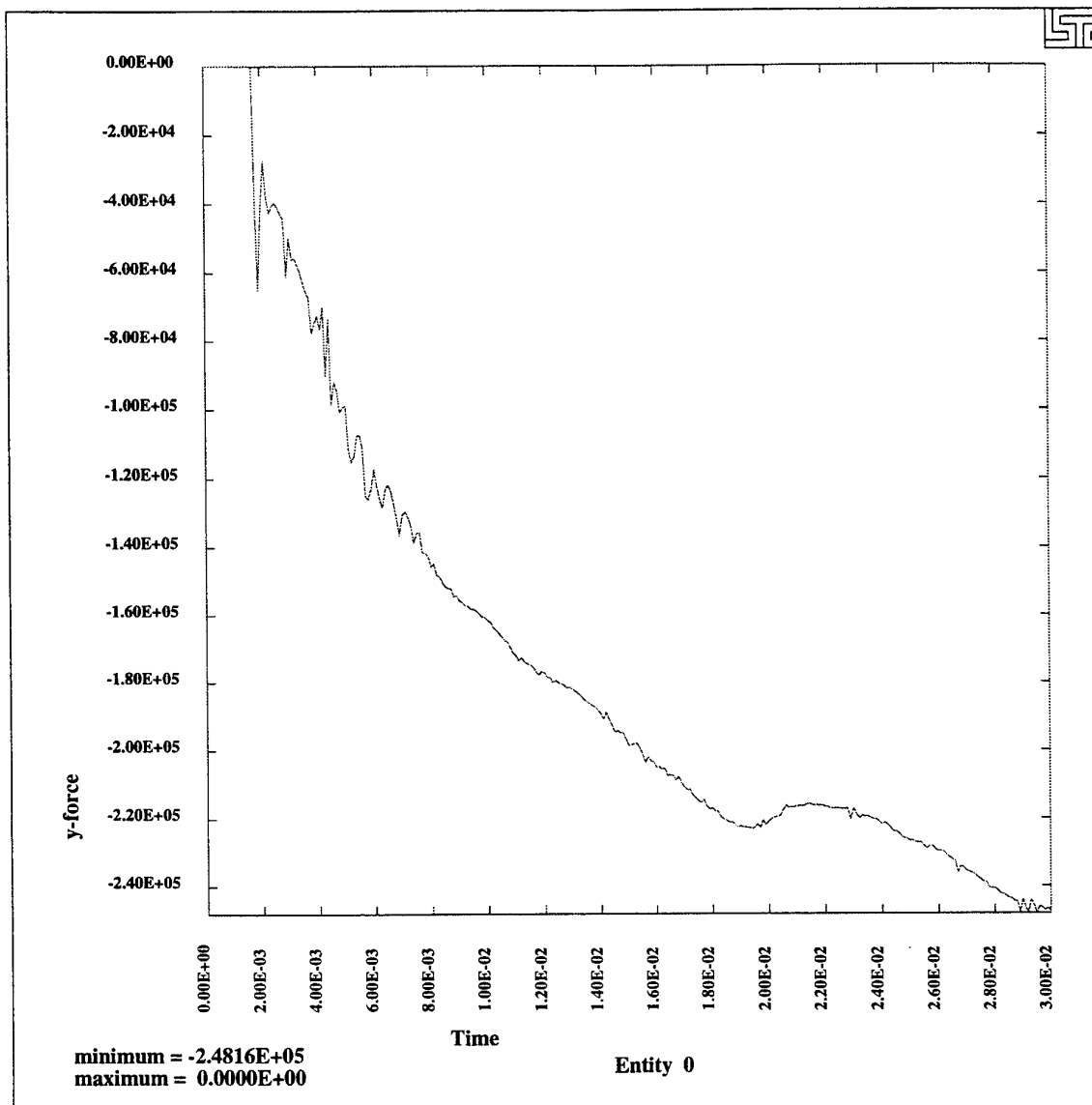


Figure 28 y-force [N] versus time [s]

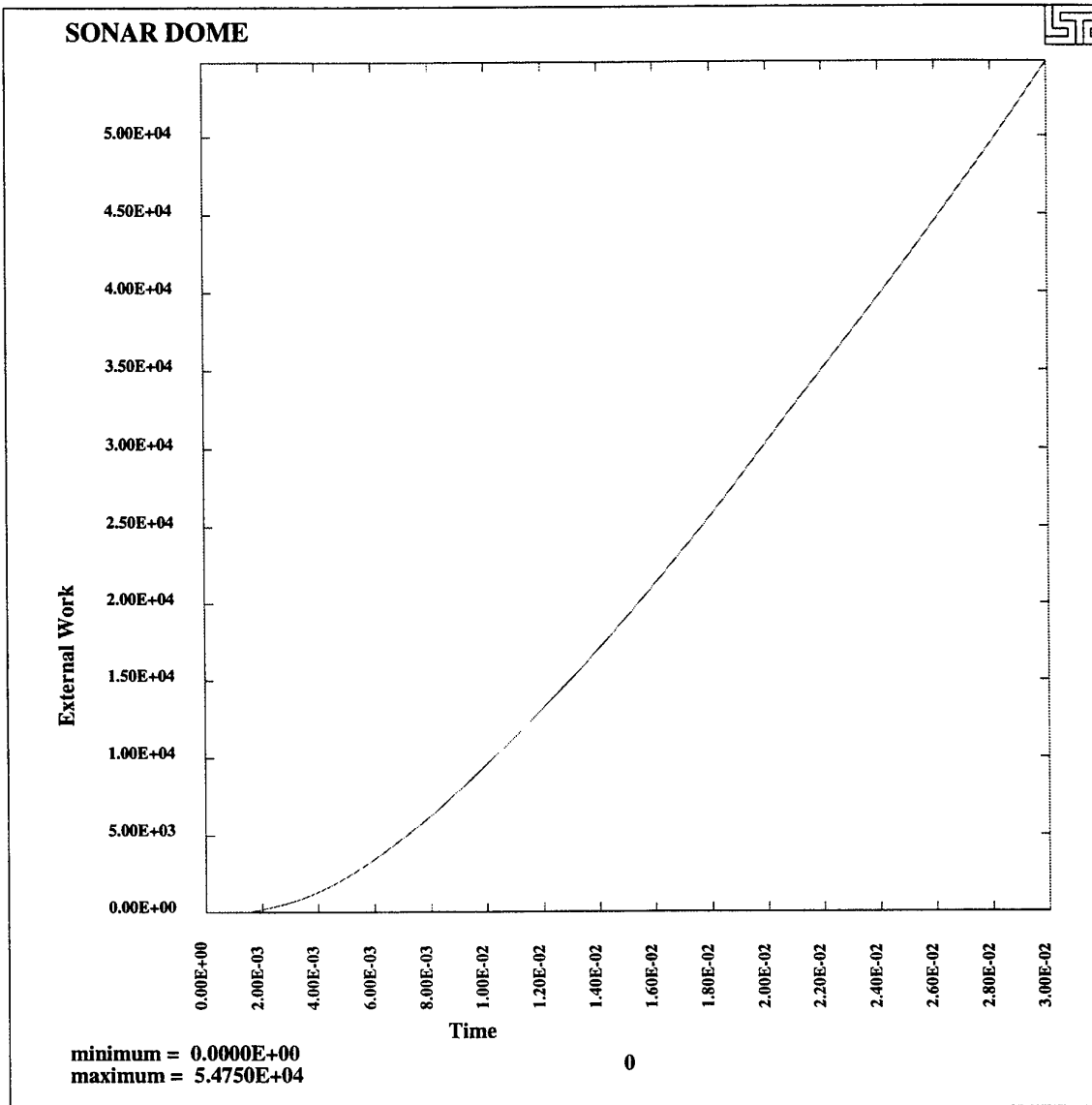


Figure 29 External work [J] versus time [s]

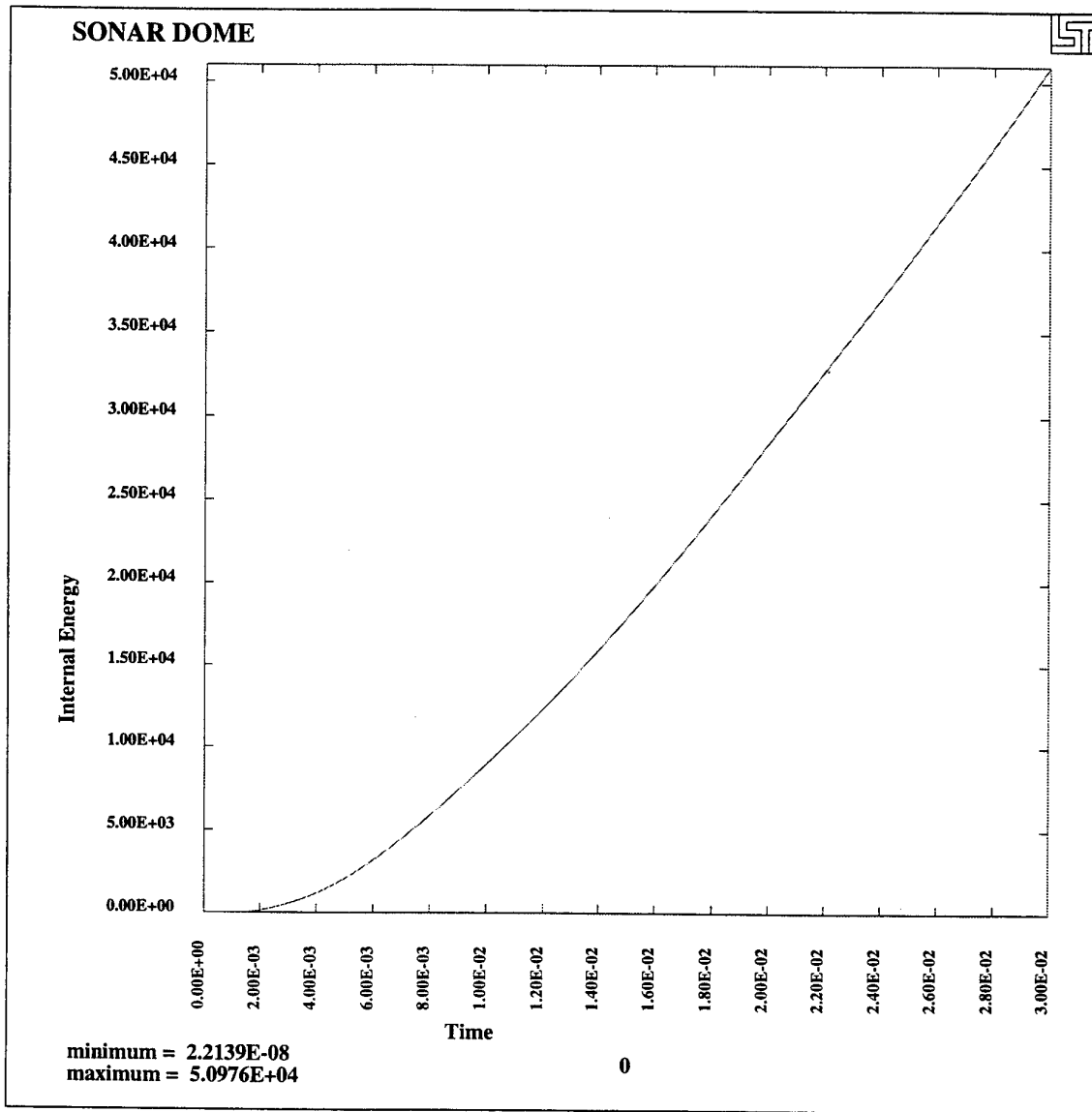


Figure 30 Internal energy [J] versus time [s]

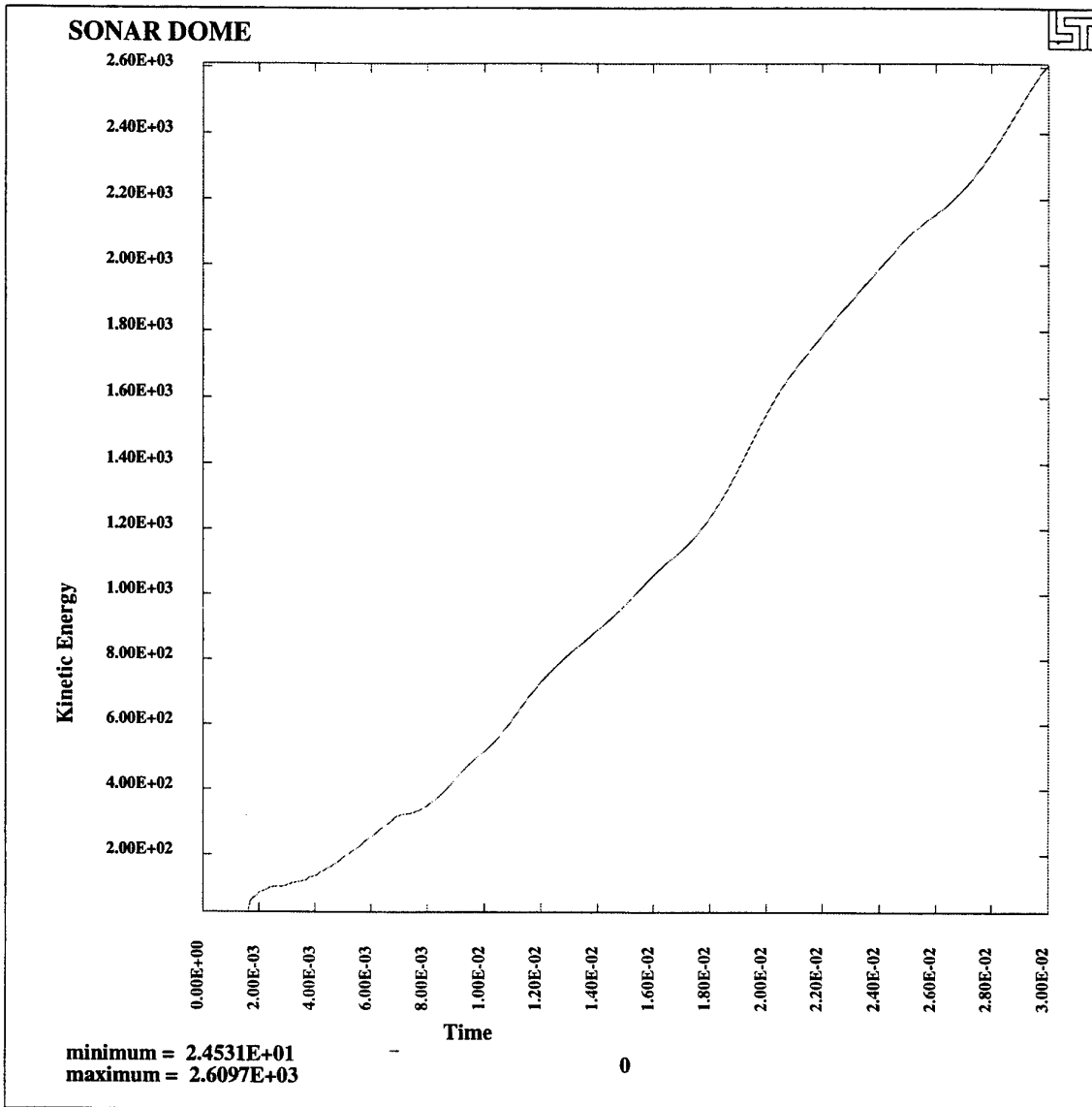


Figure 31 Kinetic energy [J] versus time [s]

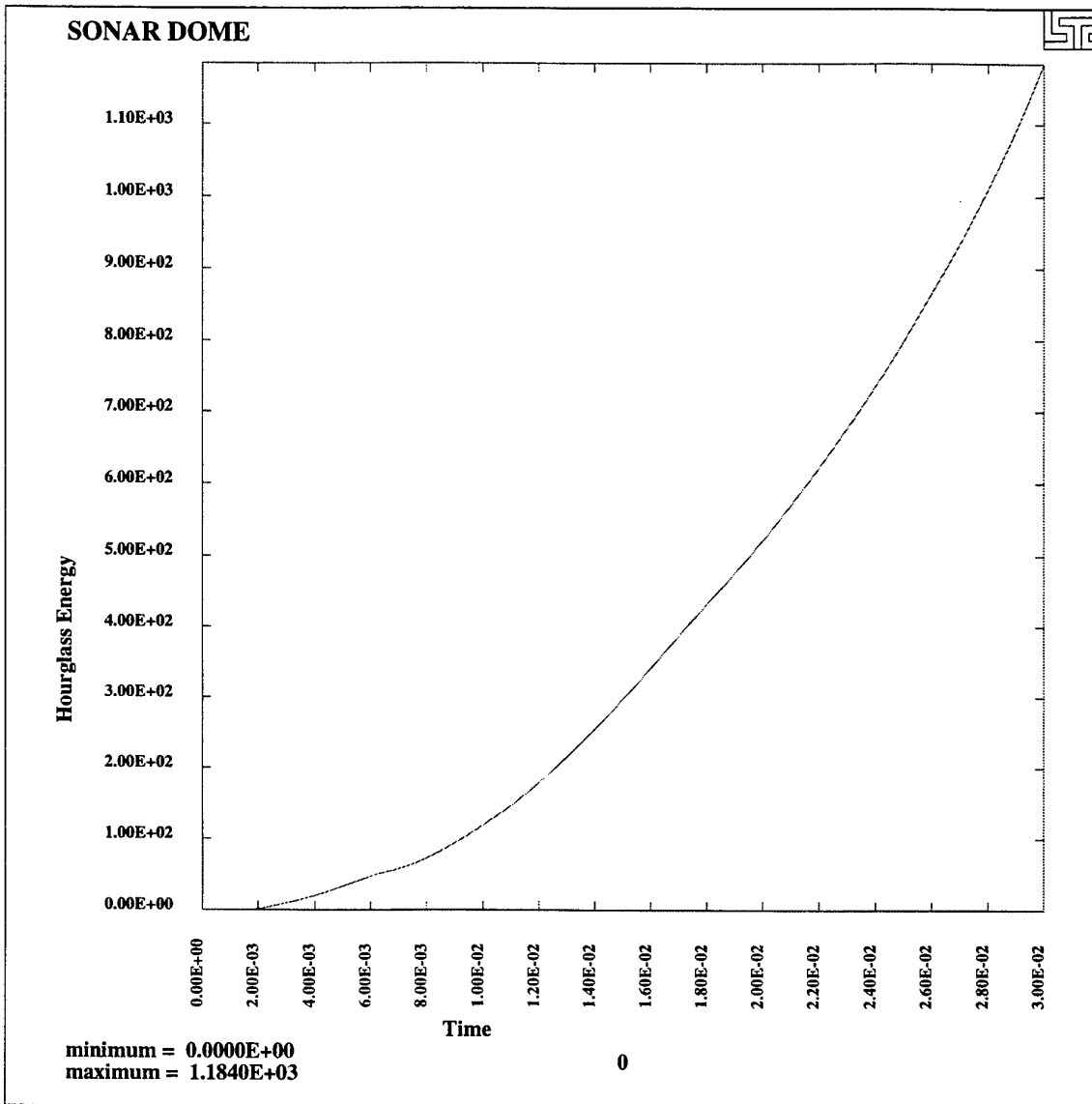


Figure 32 Hourglass energy [J] versus time [s]

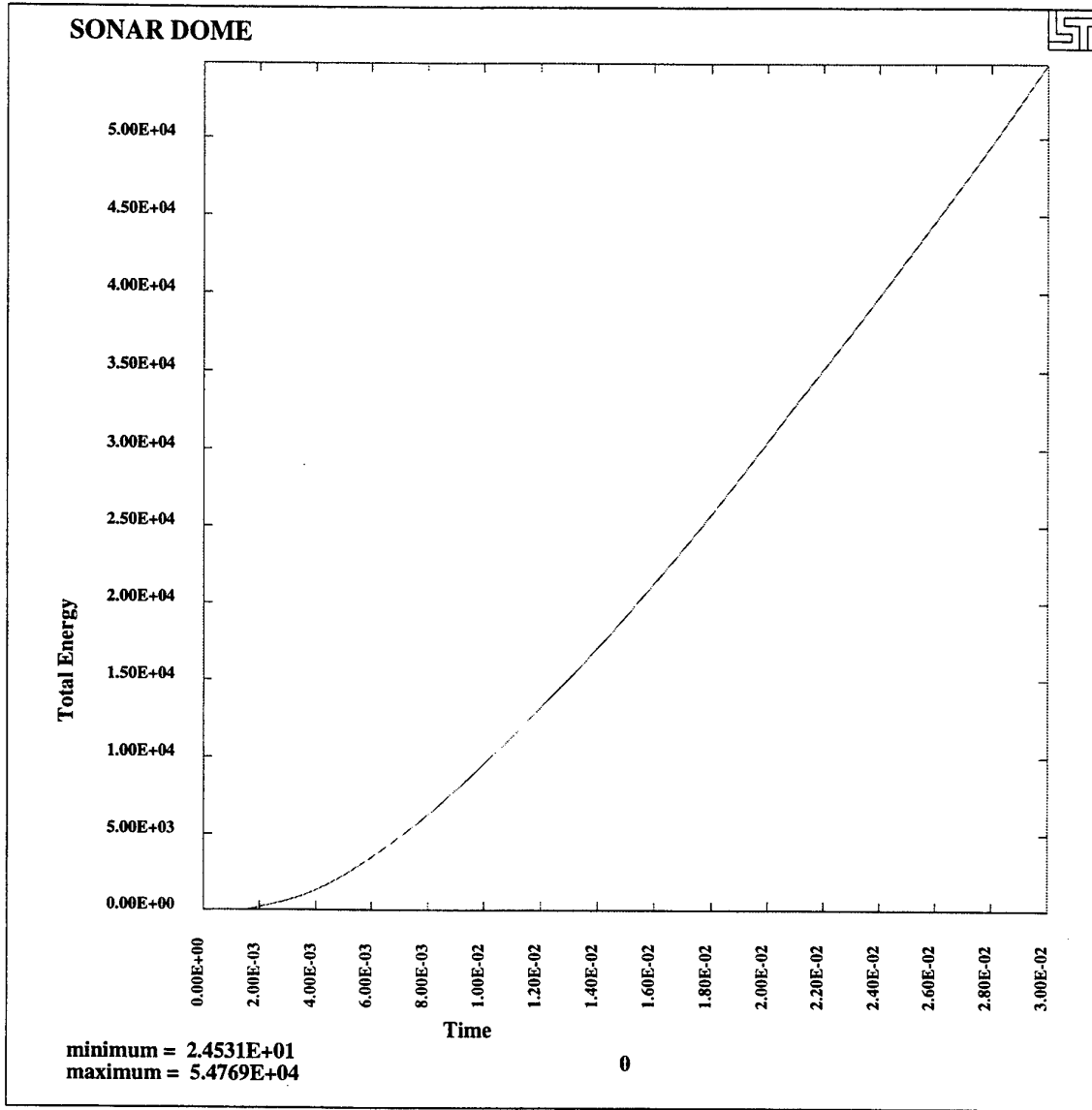


Figure 33 Total energy [J] versus time [s]

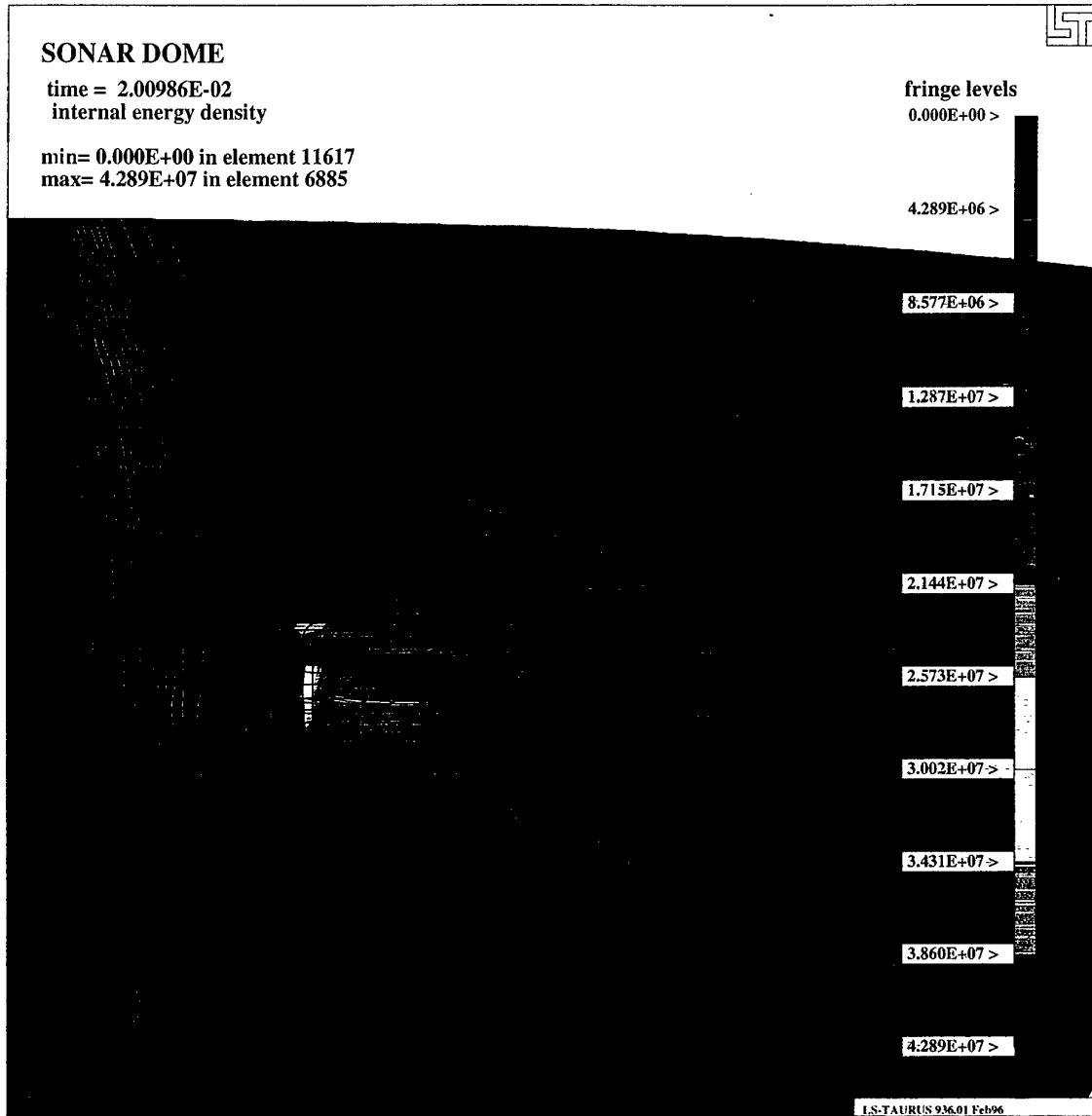


Figure 34 Internal energy density

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