71-5059

NFS

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CERTIFIED MAIL
RETURN RECEIPT REQUESTED

21G-01-0308
GOV-01-55-04
ACF-01-0343

December 13, 2001

Director
Spent Fuel Project Office
Office of Nuclear Material Safety and Safeguards
U.S. Nuclear Regulatory Commission
Washington, DC 20555

References:
1) Docket No. 71-5059
2) Safety Analysis Report for Packaging (SARP) for the Model No. NFS Uranyl Nitrate Tank Trailer Package; NFS letter (21G-00-0004) dated January 20, 2000; T.S. Baer to Director, SFPO/NMSS
3) Response to NRC's Request for Additional Information (RAI) Regarding NFS' SARP (Reference 2); NFS letter (21G-00-0194) dated October 13, 2000; B.M. Moore to Director, SFPO/NMSS
4) Revised Appendix C to NFS' SARP; NFS letter (21G-00-0243) dated November 21, 2000; B.M. Moore to Director, SFPO/NMSS
5) RAI Regarding Renewal of Certificate of Compliance No. 5059 for the Model No. NFS Uranyl Nitrate Tank Trailer Package; NRC letter (TAC No. L22329) dated August 29, 2001; E.W. Brach to B.M. Moore
6) Notification of NFS' Intent to Respond to Reference 5; NFS letter (21G-01-0206) dated September 14, 2001; B.M. Moore to Director, SFPO/NMSS
7) Request for Additional Time to Respond to Reference 5; NFS letter (21G-01-0236) dated October 12, 2001; B.M. Moore to Director, SFPO/NMSS
8) Revised Chapters 3 and 6 to Referenced SARP; NFS letter (21G-01-0246) dated October 19, 2001; B.M. Moore to Director, SFPO/NMSS
9) Extension of Due Date for Response to NRC's RAI (Reference 5); NRC letter dated November 8, 2001; E.W. Brach to B.M. Moore

Subject: **Revised APPENDIX C to Referenced SARP**

NMSS01 Public

Dear Sir:

Nuclear Fuel Services, Inc. (NFS) notified you in Reference 6 of its intent to respond to the August 29, 2001, Request for Additional Information (RAI) (Reference 5). NFS initially believed that it could provide a partial response by October 15, 2001. However, in Reference 7, NFS requested additional time to respond to the RAI – a one-week extension of the due date until October 22, 2001, to address the criticality-related issue in the RAI, and a two-month extension of the due date until December 15, 2001, to address the structural-related issues. Though NFS had not yet received a reply from the NRC regarding the extension request, NFS proceeded to submit (Reference 8) on October 19, 2001, information to address the criticality-related issue in the RAI. NFS was pleased to receive NRC approval (Reference 9) of the extension to the due date a short time later.

Please find enclosed 6 copies of the subject information prepared by NFS in response to the structural-related issues in the RAI. Revised pages, as well as new pages, are provided in Attachment I for updating of the current SARP (References 2, 3, 4, and 8). Vertical lines in the right margin of the revised pages indicate the locations of changes. Please note that the current SARP is not being replaced in its entirety by this submittal; therefore, appropriate insertion of the new information is required in order to have a complete version of the SARP. These changes are considered to be part of Revision 2 of the SARP.

If you or your staff have any questions, require additional information, or wish to discuss this matter, please contact me, or Mr. Rik Droke, Licensing and Compliance Director at (423) 743-1741. Please reference our unique document identification number (21G-01-0308) in any correspondence concerning this letter.

Sincerely,

NUCLEAR FUEL SERVICES, INC.

/BMM

B. Marie Moore
Vice President, Safety and Regulatory

RPD/lsn

cc:

Regional Administrator
U.S. Nuclear Regulatory Commission
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Atlanta Federal Center
61 Forsyth Street, SW
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Attachment I

Revised/New Pages for the Safety Analysis Report for Packaging for the Model No. NFS Uranyl Nitrate Tank Trailer

RECORD OF REVISIONS

Revision 0 (Unique Document Identification No. 21T-00-0020) Original Document

Revision 1 (Unique Document Identification No. 21T-00-0794) Page Changes

Page(s)	Section	Description of Change
1-10 & 1-11	1.2.3	Expanded Table 1-1 to include each isotope.
2-5	2.5.2	Revised Tie-Down Device analysis.
2-6	2.6.7	Revised Free Drop analysis.
2-7	2.6.9	Inserted package weight exclusion.
	2.6.10	Added basis (testing) for conclusion.
3-2	3.2.3	Corrected the maximum pressure to include effects of radiolytic gases and surging.
3-6	Appendix 3-3	Added information on contents being at 100°F, radiolytic gases, and surging.
4-3	4.2.2	Corrected information to meet definition of maximum normal operating pressure.
4-3	4.2.2.1	Corrected reference to radiolytic gas appendix.
4-3	4.2.2.1	Corrected values from revised Appendix 4-3.
4-11 - 4-13	Appendix 4-3	Completely revised.
4-14	Appendix 4-4	Changed reference for radiolytic gas generation data.
	Appendix A - D	Added new appendices.

Revision 2 (Unique Document Identification No. 21T-01-1089) Page Changes

Page(s)	Section	Description of Change
3-1 – 3-3	3.1-3.3	Revised evaluation and results.
3-4	Tables 3-1 & 3-2	New summary tables for thermal analysis results and material properties
3-5	Tables 3-3 & 3-4	New summary tables for dimensions, applied heat loads and initial conditions.
3-6 & 3-7	Appendix 3-1 & 3-2	Revised to conservatively apply 10CFR71 requirements
3-10—3-15	Appendix 3-4	New calculations and summary for worst case liquid boil-off during a fire event.
6-2	6.1	Inserted references to new Appendices 6-9, 6-10, and 6-11.
6-3	6.3.1.1	Renumbered Appendix 6-9 to 6-13. Included reference to KENO VI and MCNP462 comparison calculations.

Page(s)	Section	Description of Change
6-4 & 6-5	6.3.1.2.1	Renumbered Appendix 6-9 to 6-13.
6-6	6.3.1.2.2	Revised temperature, time and chapter number for consistence with revised Chapter 3.
6-7 & 6-8	6.3.1.2.3	Added new description for additional calculations and method.
6-8	6.4.1	Revised appendix reference and added more details for XSDRN, KENO Va, KENO VI, and MCNP462 comparison calculations.
6-9	6.4.3	Added reference to new Appendices 6-9, 6-10, and 6-11.
6-14—6-22	Appendix 6-1	Added new input case examples from additional calculations.
6-31	Appendix 6-9	Added new summary of results for additional thermal HAC calculations for spherical configuration.
6-32	Appendix 6-10	Added new summary of results for additional thermal HAC calculation for vertical cylinder configuration.
6-33	Appendix 6-11	Added new summary of results for additional thermal HAC calculation for 45° tilted cylindrical configuration.
6-34	Appendix 6-12	Added new summary of results and comparison to Chapter 3 Thermal HAC results.
6-35	Appendix 6-13	Revised Appendix 6-9 to 6-13 with insertion of new appendices 6-9, 6-10, 6-11, and 6-12.
6-40 – 6-42	Appendix 6-13	Added new Figure 6-10, 6-11, and 6-12.
	APPENDIX C	Replaced in its entirety due to new formatting and revised analyses.

TABLE OF CONTENTS (continued)

APPENDIX C

Analysis of Fruehauf MC311 Insulated Stainless Steel Cargo Tank Semi-Trailer

Container Technology Incorporated

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Nuclear Fuel Services Inc
ANALYSIS

REVISED and FORMATTED
November 19, 2001

Fruehauf MC 311 Insulated Stainless Cargo Tank Semi-Trailer

PREFACE and COMMENTS FOR THIS REFORMATTED AND REVISED Report and ANALYSIS

Part 1 of this current report contains reformatted original calculations relating to "10g longitudinal and 5g lateral acceleration of "packaging attachments". Since "packaging" in the context of this report is a highway tank trailer, few of which are intentionally designed for service of that severity, and since it is not clear from the context of the regulations what constitutes "attachments" relating to tank trailers, the original calculations were excluded from the original draft. They are included here as Part 1 for reference only. Although this type analysis is irrelevant to the reality of highway transportation, the calculations nevertheless indicate that the tank trailer has adequate strength to withstand these forces at the points of attachment to the tandem support structure. (PART 4 OF ORIGINAL REPORT, OCT. 6, 2000) BY NFS

Parts 2 & 3 of this Report and Analysis were originally submitted and captioned parts 1 & ~~2~~ 3 BY NFS in a report dated October 6, 2000. These parts have been re-captioned and reformatted for binding.

Part 2 of this current report contains a scaled pictorial demonstration that an overturn would result in a more severe impact condition than a one foot drop.
Because of the length to width to height aspect ratio of tank semi-trailers, an overturn is the only outcome to be expected in an accident situation wherein the trailer does not maintain it's operational orientation.

Part 3 of this current report contains calculations relating to how well the trailer conforms to *current* DOT regulations relating to structural integrity requirements for new construction of highway cargo tanks. In spite of the fact that the trailer was constructed under less severe specifications, the calculations in Part 3 indicate that it would essentially conform to the more stringent DOT standards of structural integrity that are now in effect.

Part 4 of this report contains analysis, calculations and conclusions relating to the damage effects of a one foot drop of the trailer on to it's landing gear legs. (PART 2 OF ORIGINAL REPORT OCT. 6, 2000) BY NFS

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ANALYSIS

Fruehauf MC 311 Insulated Stainless Cargo Tank Semi-Trailer

May 11, 2001

Revised

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Nuclear Fuel Services Inc

ANALYSIS
Fruehauf MC 311 Insulated Stainless Cargo Tank Semi-Trailer

May 11, 2001 Revised

Part 1
Analysis of effects of 10 g longitudinal load, a 5 g lateral load and a 2 g vertical load on the attachments of the tank.

Notes and calculations for Part 1

Part 2
Determination of what accident or roll-over conditions that would be less severe than a one foot drop in the upright position.

Notes and calculations for Part 2

Part 3
Analysis of the existing trailer, built to an older standard, (MC311) under operating conditions (DOT 411 see US DOT 49CFR)

Notes and calculations for Part 3

Part 4
Determination of the effects of a one foot drop in the upright position

Appendices to item Part 4

ANALYSIS
Fruehauf MC311 Insulated Stainless Steel Cargo Tank Semi-Trailer

Part 1

Analysis of effects of 10g longitudinal load, 5g vertical load & 2g lateral load on the attachments of the tank.

Commentary, conclusion, calculations and supporting diagram for Part 1

PART 1 ANALYSIS OF EFFECTS OF 10g LONGITUDINAL LOAD, 5g LATERAL LOAD & 2g VERTICAL LOAD ON THE ATTACHMENTS OF THE TANK

COMMENTARY:

This requirement presumes the occurrence of unrealistic if not impossible conditions in the operation of cargo tank trailers. Under the 10 g and 5 g mandated conditions of the analysis, the tank would accelerate due to longitudinal or lateral forces applied to the center of the tank; "the attachments would be forced "along for the ride" (no tire to ground coefficient of friction would be sufficient to sufficiently attenuate such accelerations), but the inertial resistance force vectors at the mass center of each attachment would create a "pendulum" effect, creating a "racking" or bending moment depending on the attachment method of each particular structure.

The most severe effect on attachments under longitudinal acceleration conditions would occur at the connection of the tank to the three sets of connections to the reinforcing rings at the rear of the trailer. These consist of flanged gussets and lateral bolster plates just above the ten inch deep tandem subframe structure. The weakest orientation is in the 10 g direction. In order for this "rectangular" configuration to "rack" or be deflected, it must assume a parallelogram configuration requiring that the vertical and horizontal components rotate in relation to each other. By inspection, it is apparent that resistance to this distortion must occur at the top of the subframe at the base of the flanged gussets mentioned above. Nevertheless, the consequent bending moments in the vertical longitudinal plane create stresses that do not exceed the yield strength of the material of construction. Horizontal plane shear stress are created that do not exceed 62% of the ultimate tensile stress.

The stiffness of the same connections in the lateral direction, (at right angle to the direction of the above analysis) is many times greater, but is subjected to half the accelerative force; the flexural stress so created is therefore below the yield strength of the material of construction.

The horizontal plane sectional area of the connections is adequate to support twice the weight (up or down)of the tank trailer without exceeding the yield strength of the material of construction.

CONCLUSION:
THE CITED REGULATION IS NOT A REASONABLE CRITERIA FOR CARGO TANK TRAILERS. THE CONNECTION OF THE TANK TO IT'S LEAST STRONG SUPPORT (at the tandem) WILL ELASTICALLY SUSTAIN THE FLEXURAL & COMPRESSIVE STRESSES (AS WELL AS PLASTIC SHEAR STRESS) CAUSED BY 10g ACCELERATION.

ANALYSIS OF APPLICATION OF 10CFR §71.45 TO TANDEM SUPPORT OF FRUEHAUF MC312 3700 uswg CARGO TANK STAINLESS STEEL SEMI-TRAILER REF Part 3 Fig. 2

The above cited regulation is concerned with the effects of acceleration of a packaging on the restraint (tie-down) devices that afix the packaging to the vehicle providing means of transport. In particular, the regulation anticipates a situation where the center of the package would be accelerated by a force vector of 11.358 times the package weight, this force vector consisting of a 2 times the loaded weight as a vertical component, 10 times the loaded weight as a horizontal component in the direction of travel, and 5 times the loaded weight as a horizontal component transverse to the direction of travel. If a package were to be accelerated as stated, the vehicle inertia would provide an equal and opposite inertial force for each component direction. These opposing forces would act to seperate the packaging from the vehicle, and the seperation force would create stress in the connecting devices. When the the packaging and vehicle is defined as a cargo tank semi-trailer, the determination of what is 'packaging' , what is 'tie-downs' and what is 'vehicle' provides a challange of interpretation. For this analysis the interpretation is as follows:

<u>packaging</u>: the jacketed and insulated tank vessel and reinforcing rings, including all devices that are both supported BY and connected to the tank vessel or it's reinforcing rings

vehicle: the tandem subframe box, suspension, axles, wheels tires that provide support to the tank vessel directly or through it's reinforcing rings AND the upper coupler plate and subframe which provide a means for one end of the tank (packaging) to be supported by the truck tractor.

<u>tie-downs</u>: the bolster plates and gussets that connect the tandem subframe box to the three tank reinforcing rings atthe rear AND the welded connection of the upper coupler plate and subframe directly to the tank pads and tank reinforcing rings at the front.

For perspective, a comparison of the mass (weight) of '<u>packaging</u>' and '<u>vehicle</u>' is listed here:

packaging	loaded tank & supported equipment = 59,650 lbs (nearly 60,000 lb)
vehicle-tandem	4,200 lb (aproximately one-fourteenth of the packaging weight)
vehicle- upper coupler	550 lb (the 'packaging' weighs one hundred eight times as much and the tandem 'vehicle' weighs seven and a half times as much as the upper coupler 'vehicle')
vehicle- upper coupler AND truck	16,150 lb (the 'packaging' weighs nearly 3.7 times much as the upper coupler AND truck 'vehicle'

Inspection of this comparison supports the analogy of viewing each of the 'vehicles', as here defined, as a pendulum" suspended from the 'packaging' and each of the tie-down devices as a rigid joint or "frozen hinge", the purpose of which is to restrain the "pendulum" vehicle from "swinging", i.e. moving with respect with the tank vessel 'packaging'. The tie-dow' failure criteria is determination of stress level beyond the yield.

At the upper coupler, (because of the fifth-wheel-plate hinged at the tractor), a moment can only be transmitted through the king-pin in the lateral-vertical plane, (the plane in which lies the 5g accelerating force). If the "vehicle' is deemed to be the truck AND upper coupler as a rigid entity, the mass center of the combination is considered to be half of the fifth-wheel height or (1/2 of 50in) +4in=29in

Because of the hinged plate, the moment producing 10g accelerative force is considered to act at the fifth wheel level, a distance of only 14 in.

A similar analysis at the tandem reveals that for both the 10g and 5g directions, the "pendulum" lengths are the same. However, inspection of the tie down connection at that location reveals that it is is an open system of bolster plates and gussets with a fraction of the rigidity or stiffness of the the connections to the shell at the front of the trailer. In fact, an examination of the properties of the moment resisting section in the 10g direction shows that it is the least robust flexural section available to resist moments. This location direction, and section modulus will be the focus of this analysis. The analysis consists of evaluating the NECESSARY section modulus of the section for the material to be stressed to it's yield point. This NECESSARY section modulus, once determined is then compared to the actual section modulus to determine the adequacy of the section as a tie down.

NOMENCLATURE AND DEFINITION OF TERMS

ik plane area, each set required for ***i*** axis acceleration	ai_{gst}
design safety factor	dsf
tank contact height	h_{gst}
load factors ***i, j, k*** axis	lf_i lf_j lf_k
acceleration moment each gusset set, ***i*** axis	mij_{gst}
acceleration moment each gusset set, ***k*** axis	mjk_{gst}
number of tank supports	N
number of gusset sets	N_{gst}
MASV	Sm
yield A36	Sy
i axis shear force in ***ik*** plane	vFi
loaded tank weight	W
landing gear weight	W_{lg}
upper coupler weight	W_{ko}
tandem weight	W_{tdm}
actual ***i*** axis section modulus	Zi_{gst}
actual ***k*** axis section modulus	Zk_{gst}
section modulus, each set required for ***i*** axis acceleration	zi_{gst}
section modulus, each setrequired for ***k*** axis acceleration	zk_{gst}

<u>Axis Convention</u>
***i* axis:** longitudinal
***j* axis:** vertical
***k* axis:** lateral
primary units inches, lbs unlessotherwise noted

Analysis of strength of structure between 10 inch channel tandem box rails and tank ring contact point aproximately 18 inches above frame.
Rotation in the ***ij*** plane and shear in ***ik*** plane from ***i*** axis 10g acceleration and rotation in the ***jk*** plane and shear in the ***ik*** plane from ***k*** axis 5g acceleration

yield A36	tank contact height	tandem weight	upper coupler weight	landing gear weight	loaded tank weight	design safety factor	number of gusset sets
$Sy := 36000$	$h_{gst} := 18$	$W_{tdm} := 4200$	$W_{ko} := 550$	$W_{lg} := 550$	$W := 59100$	$dsf := 1$	$N_{gst} := 3$

ultimate A36	shear factor	load factors ***i, j, k*** axis	MASV	number of tank supports
$Su := 58000$	$kv := 0.62$	$lf_i := 10$ $lf_j := 2$ $lf_k := 5$	$Sm := \frac{Sy}{dsf}$ $Sm = 36000$	$N := 1$

RESISTANCE TO RACKING BETWEEN TANK and TANDEM GUSSET SUPPPORTS AT CONNECTION TO TANDEM BOX RAILS, [*ij* & *jk* plane]

NOTE ASSUMED CONDITION :UNRESTRAINED TANDEM; tandem inertially resists 10g acceleration of the CENTER of the loaded tank; the tandem is assumed to behave as a pendulum exerting a racking force on the connection to the tandem.

acceleration moment each gusset set, *i* axis	section modulus, each set required for *i* axis acceleration	actual *i* axis section modulus
$mij_{gst} := \left[\frac{(lf_i)\cdot\left(\frac{W_{tdm}}{N}\right)}{N_{gst}}\right]\cdot h_{gst}$	$zi_{gst} = \left(\frac{mij_{gst}}{Sm}\right)$	$Zi_{gst} := 12.353$
$mij_{gst} = 252000$	$zi_{gst} = 7$	TRUE $(Zi_{gst} \geq zi_{gst}) = 1$

acceleration moment each gusset set, *k* axis	section modulus, each set required for *k* axis acceleration	actual *k* axis section modulus
$mjk_{gst} := \left[\frac{(lf_k)\cdot\left(\frac{W}{N}\right)}{N_{gst}}\right]$	$zk_{gst} = \left(\frac{mjk_{gst}}{Sm}\right)$	$Zk_{gst} := 135.597$
$mjk_{gst} = 98500$	$zk_{gst} = 2.736$	TRUE $(Zk_{gst} \geq zk_{gst}) = 1$

RESISTANCE TO SHEAR OF TANDEM GUSSET SUPPPORTS AT CONNECTION TO TANDEM BOX RAILS, [*ik* plane]

i axis shear force in *ik* plane	*ik* plane area, each set required for *i* axis acceleration	actual *ik* plane area each gusset set (attached)
$vFi := \left[\frac{(lf_i)\cdot\left(\frac{W}{N}\right)}{N_{gst}}\right]$	$ai_{gst} = \frac{vFi}{Su\cdot(kv)}$	$Ai_{gst} := (12.0375 - 6.375)$
$vFi = 197000$	$ai_{gst} = 5.478$	TRUE $(Ai_{gst} \geq ai_{gst}) = 1$

k axis shear force in *ik* plane will be 1/2 *i* axis shear force, but *ik* plane area is the same

CONCLUSION :

Present tandem support structure will withstand 10g acceleration in the longitudinal each *i* direction without exceeding yield strength of tandem support material in flexure.

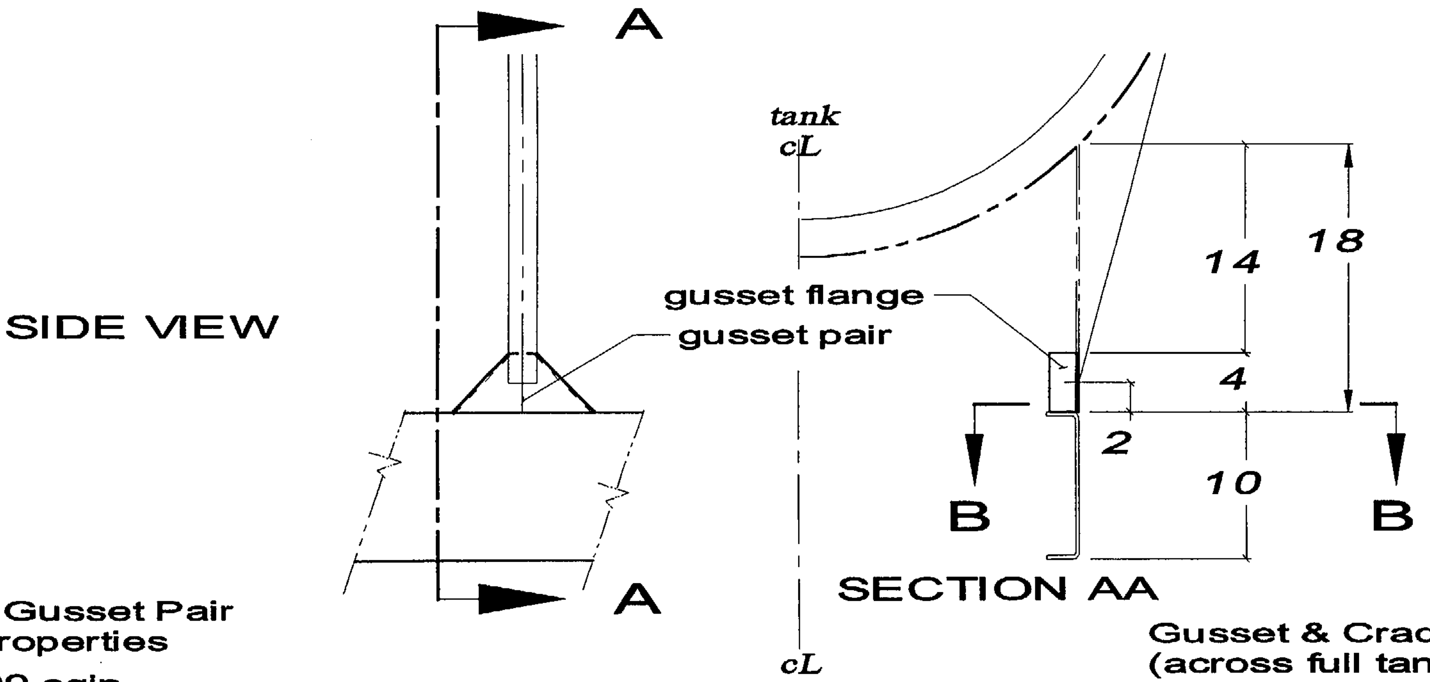
Area of each gusset set at junction with tandem box frame rails is adequate to withstand 10g acceleration in the longitudinal each *i* direction without exceeding 62% of ultimate strength of tandem support material in shear.

original calculations performed for CTEC in Lubbock, Texas
revised/reformatted for Container Technology Incorporated

Thomas A. Rogers P E 7/27/2001 date
Texas Registration 79212

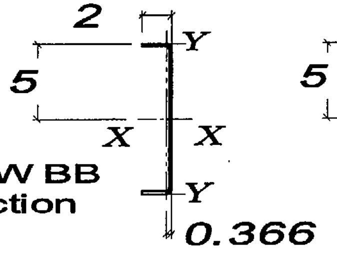
THOMAS A. ROGERS
79212

revision note: recaptioned & reformated for binding; no calculation revisions



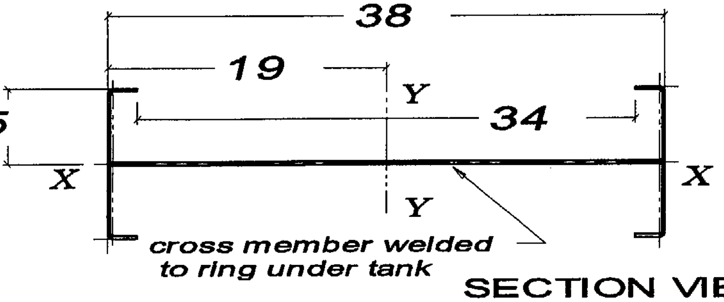
One Side Gusset Pair
Section Properties

area=2.509 sqin
Ixx=30.872in^4
c=5.0in
zx=6.174^3



SECTION VIEW BB
gusset pair section
one side only

3/16 carbon steel
(A36) plate construction



SECTION VIEW BB
across full tandem structure

Gusset & Cradle Properties
(across full tandem structure)
area=12.0375sqin
unattached area=6.375sqin
attached area =5.6625sqin
Ixx=61.765in^4
Iyy=2576.35in^4
cx=19in
cy=5.0in
zx=12.353in^3
zy=135.597in^3

Part 1 Fig. 1
TANDEM UNDERSTRUCTURE DIMENSIONS & DETAIL

ANALYSIS
Fruehauf MC 311 Insulated Stainless Cargo Tank Semi-Trailer

Part 2

Determination of what accident or roll-over conditions that would be less severe than a one foot drop in the upright position.

Notes and calculations for Part 2

PART 2 DETERMINATION OF WHAT ACCIDENT OR ROLL OVER CONDITIONS THAT WOULD BE LESS SEVERE THAN A ONE FOOT DROP IN THE UPRIGHT POSITION

COMMENTARY:

Inspection of the diagram Fig. 1, attached to this Part 2, reveals that as the sequence of overturn progresses (right to left), the trailer center of mass actually rises at to a maximum height at the point of unstable equilibrium, just before it begins to fall on its side (second from right, labled "equilibrium for impending overturn")

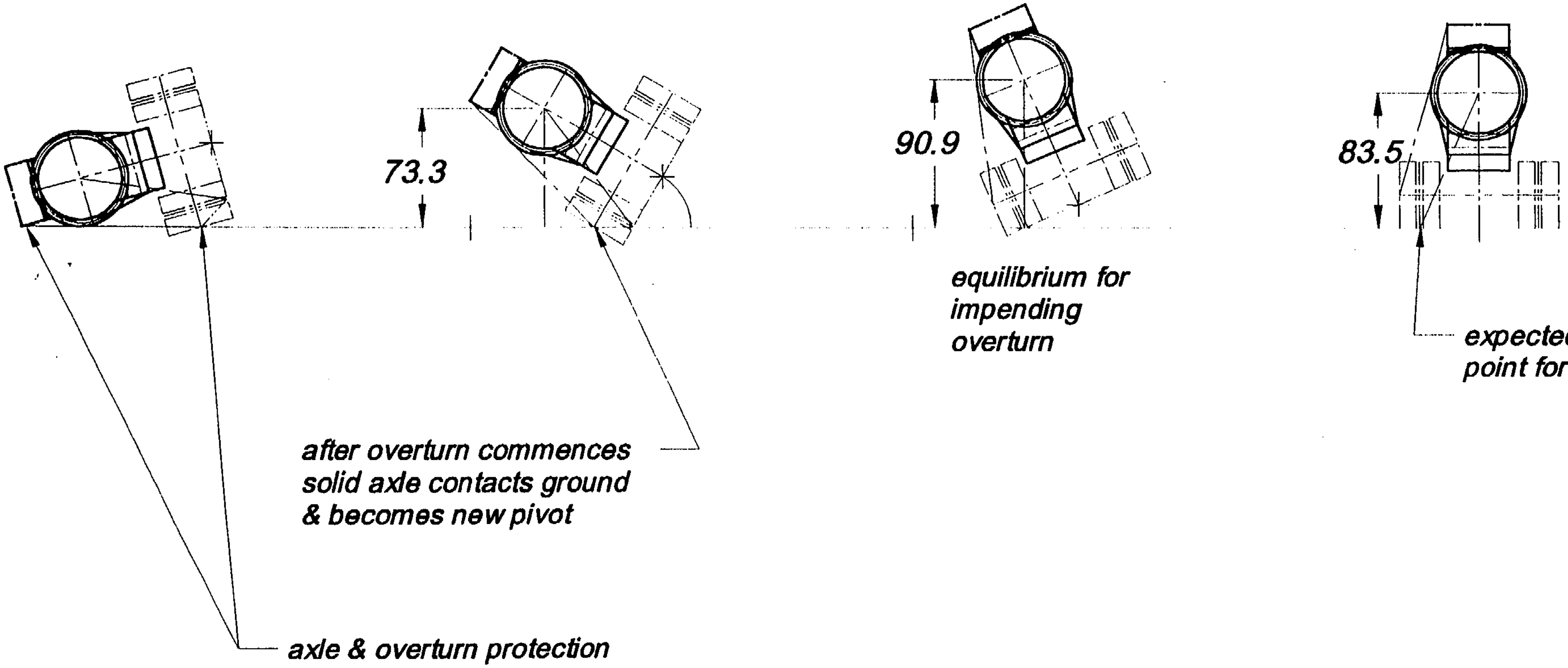
The third (from right) illustration shows that the first structrual part to hit the ground (ignoring fenders and other non-structural equipment) is the end of the axle. At this point one of two sequences can occur, as follows:

> If the spring to which the axle is attached is not pushed out of the spring hanger at the tandem box frame, the end of the axle becomes the new pivot point as the trailers rotational momentum continues to roll the trailer. The trailer will then most likely fall freely directly on to its side.
>
> If the spring does become dislodged, (this will not occur instantly), the trailer will behave somewhat as if it were rolling on to an elastic surface while the spring is being pushed out of the hanger. Once the axle and spring are detached, the axle will no longer be able to serve as a supporting pivot and the trailer will drop directly on to its side. If rotational momentum has not been absorbed, the trailer will continue to roll until the overturn contacts the ground. Depending on the unabsorbed rotational energy, the trailer will continue to rotate to the extent allowed by the overturn, which will then become a new pivot.

Either of the two foregoing scenarios will result in severe damage to the tank because the drop distance is grossly in excess of one foot. This is not to imply that the trailer will rupture and lose its contents; the insulation and jacket will almost guarantee a somewhat softer landing than could otherwise be expected, but the gross damage will far exceed any that can be expected from a one foot drop on to wheels and landing gear; these are structures that are designed to bear the weight of the loaded trailer, whereas the side of the tank is NOT designed to sustain that type of abuse.

CONCLUSIONS:

THERE IS NO CONDITIONS THAT WOULD BE LESS SEVERE THAN A ONE FOOT DROP IN THE UPRIGHT POSITION



equilibrium for
impending
overturn

expected pivot
point for overturn

after overturn commences
solid axle contacts ground
& becomes new pivot

axle & overturn protection
device support trailer and
absorb overturn energy
unless axle yields, if so
then side of trailer bears
brunt of overturn energy

OVERTURN SEQUENCE
NO OVERTURN EVENT
DROPS TRAILER ONE
ONE FOOT OR LESS

NFS overturn part2 fig1
Container Technology Incorporated

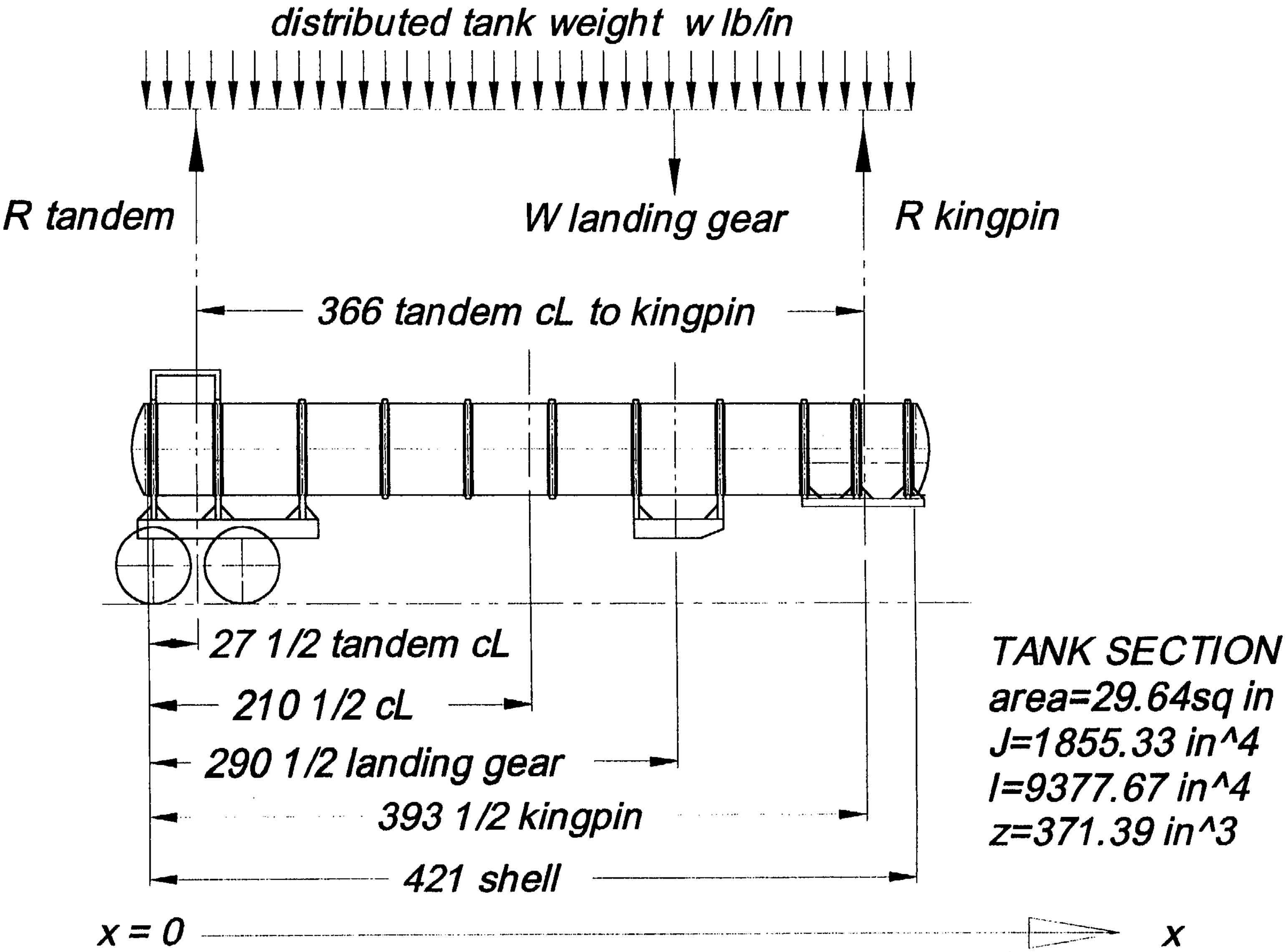
PART 2 Fig. 1

ANALYSIS
Fruehauf MC 311 Insulated Stainless Cargo Tank Semi-Trailer

Part 3

Analysis of the existing trailer, built to an older standard, (MC311) under operating conditions (DOT 411 see US DOT 49CFR).

Notes and calculations for Part 3



Part 3 Fig.1
TANK REPRESENTED AS A
SIMPLY SUPPORTED BEAM

NFS part 3 fig.1 tank beam
Container Technology Incorporated

NUCLEAR FUEL SERVICES PART 3
MC 311 CARGO TANK ANALYSIS REFERENCE Part 3 Fig. 1
ANALYSIS OF THE EFFECT OF APPLICATION OF CERTAIN CURRENT [49CFR] LOAD CONDITIONS THAT CAUSE FLEXURAL STRESS IN THE TANK SHELL.

NOTE: THESE LOAD CONDITION REQUIREMENTS WERE NOT IN EFFECT AT THE TIME THAT MC311 WAS A CURRENT STANDARD

DEFINITIONS& NOMENCLATURE [See attached diagram: "*TANK REPRESENTED AS A BEAM*"

design safety factor	dsf
height of cL	H
moment of inertia, in^4	I
support length, in	L
tank shell length, in	Ls
kingpin offset, in	$Lf_{kingpin}$
tandem offset, in	Lf_{tandem}
rear head seam to landing gear, in	L_{lg}
operating & emergency load factors	lf_i lf_j lf_k
combined braking and weight moments modified by $(1+lf_j)$ load factor down and (lf_i) forward	$mj(x)$
braking moment variance with x	$mj_i(x)$
surge load factor	Ns
tank radius, in	Ro
kingpin reaction, lb	$R_{kingpin}$
tandem reaction , lb	R_{tandem}
reaction effect of braking moment	ΔR
maximum allowable operating stress, psi	Sm_T
ss yield strength, psi	Sy_T
ss ultimate strength, psi	Su_T
stress from combined braking and weight moments modified by $(1+lf_j)$ load factor down and (lf_i) forward	$\sigma m(x)$
stress from emergency braking moment variance with x, psi	$\sigma mji_e(x)$
unmodified shear variance along shell length	$v(x)$
landing gear weight, lb	W_{lg}
tank as beamweight, lb	W_T
distributed tankweight, lb/in	w_T
i axis load points	$X0$ $X1$ $X2$ $X3$ $X4$
shell length variable along i axis	x
tank section modulus, in^3	Z

axis convention
i longitudinal in direction of travel
j upward, normal to i
k lateral, normal to i & j

REVISION NOTE:
LONGITUDINALCOMPRESSION DEVICES NOT INCLUDED IN CALCULATION OF TANK SECTION.
SEE CONCLUSION.

VESSEL SECTION DATA [See attached diagram: "*TANK REPRESENTED AS A BEAM*" Part 3 Fig.1

moment of inertia, in^4	tank radius, in	tank section modulus, in^3	height of cL	ss yield strength, psi	ss ultimate strength, psi	design safety factor
$I := 9377.67$	$Ro := 25.25$	$Z := \frac{I}{Ro}$	$H := 83$	$Sy_T := 30000$	$Su_T := 75000$	$dsf := 4$
		$Z = 371.393$				

tank shell length, in	support length, in	maximum allowable operating stress, psi	operating & emergency load factors			surge load factor
$Ls := 421$	$L := 366$	$Sm_T := \frac{Sy_T}{dsf}$	$lf_i := 0.35$	$lf_j := 0.35$	$lf_k := 0.20$	$Ns := 2$
		$Sm_T = 7500$				

tandem offset, in	kingpin offset, in	tank as beam is symetrically supported	rear head seam to landing gear, in
$Lf_{tandem} := 27.5$	$Lf_{kingpin} := 393.5 - Ls$	$\lvert Lf_{kingpin} \rvert = \lvert Lf_{tandem} \rvert = 1$	$L_{lg} := 290.5$
	$Lf_{kingpin} = -27.5$		

SUPPORT REACTIONS

tank as beam weight, lb	distributed tank weight, lb/in	landing gear weight, lb
$W_T := 59100$	$w_T := \frac{W_T}{Ls}$	$W_{lg} := 550$

kingpin reaction, lb

$$R_{kingpin} := \left(\frac{W_T}{2}\right) + \left(\frac{L_{lg} - Lf_{tandem}}{L}\right) \cdot W_{lg}$$

$$R_{kingpin} = 29945$$

tandem reaction , lb

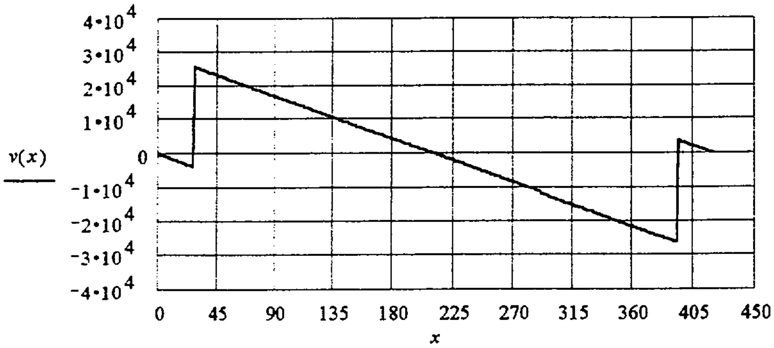
$$R_{tandem} := \left(W_T + W_{lg}\right) - R_{kingpin}$$

$$R_{tandem} = 29705$$

shell length variable along ***i*** axis	***i*** axis load points				
$x := 0 .. Ls$	$X0 := 0$	$X1 := Lf_{tandem}$	$X2 := L_{lg}$	$X3 := Ls - \lvert Lf_{kingpin} \rvert$	$X4 := Ls$
		$X1 = 27.5$	$X2 = 290.5$	$X3 = 393.5$	$X4 = 421$

UNMODIFIED SHEAR VARIANCE ALONG SHELL LENGTH

$$v(x) := -\left(w_T\right)\cdot x - \left[R_{tandem}\cdot(x \ge X1)\right] - \left[W_{lg}\cdot(x \ge X2)\right] + \left[R_{kingpin}\cdot(x \ge X3)\right]$$



$v(X1 - 0) = 25844.33$

$v\left(\frac{L}{2} + 28.60259\right) = 0$

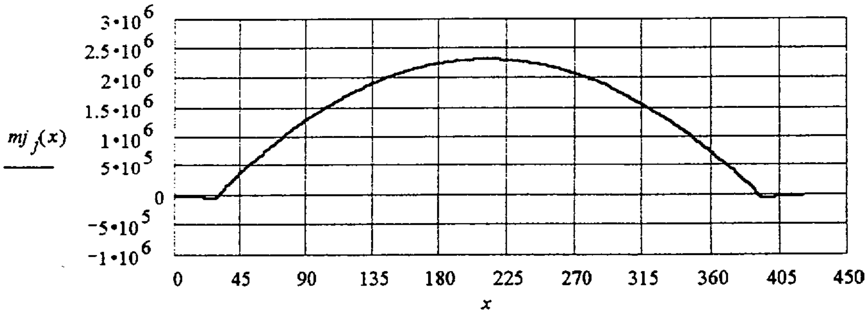
$\frac{L}{2} + 28.60259 = 211.603$

$v(X3 - 0.0001) = -26084.753$

$X3 - 0.0001 = 393.5$

UNMODIFIED MOMENTS FROM WEIGHT ONLY

$$mj_j(x) := \left[-\left(\frac{w_T}{2}\right)\cdot x^2\right] + \left[R_{tandem}\cdot(x - X1)\cdot(x \ge X1)\right] - \left[W_{lg}\cdot(x - X2)\cdot(x \ge X2)\right] + \left[R_{kingpin}\cdot(x - X3)\cdot(x \ge X3)\right]$$



MODIFIED WEIGHT MOMENTS FROM EMERGENCY CONDITIONS

[downward load factor is twice operating [(1+2x*lf$_j$*) = 1.7]

$$\sigma j_e(x) := \left(\frac{1.7\cdot mj_j(x)}{Z}\right)$$

$$\sigma j_e\left(\frac{L}{2} + 28.60259\right) = 10647$$

LOAD FACTOR MODIFIED MOMENT FROM EFFECTS OF WEIGHT & BRAKING

reaction effect of braking moment

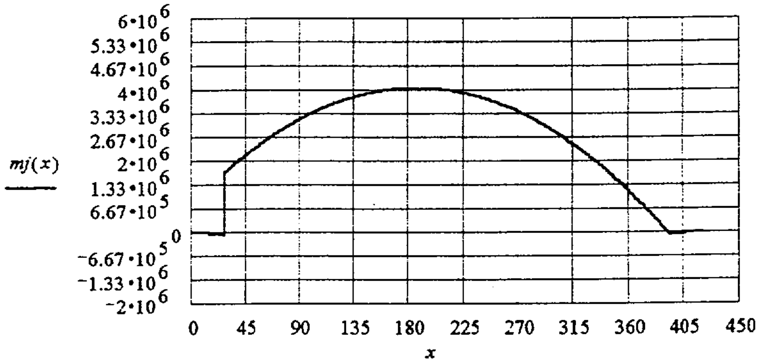
$$\Delta R = \left(lf_i\right)\cdot\left[\frac{\left(W_T + W_{lg}\right)\cdot H}{L}\right]$$

braking moment variance with x

$$mj_i(x) := \left[\left[\left(W_T + W_{lg}\right)\cdot H\right]\cdot\left(\frac{X3 - x}{X3 - X1}\right)\cdot(x \geq X1)\cdot(x < X3)\right]$$

combined braking and weight moments modified by $(1+lf_j)$ load factor down and (lf_i) forward

$$mj(x) = \left(1 + lf_j\right)\cdot mj_j(x) + \left(lf_i\right)\cdot\left(mj_i(x)\right) \qquad \sigma m(x) := \frac{mj(x)}{Z}$$



$mj(186) = 4060295$

$\sigma m(186) = 10933$

EMERGENCY BRAKING

$$\sigma mji_e(x) := \frac{\left(2\cdot lf_i\right)\cdot mj_i(x) + mj_j(x)}{Z} \qquad \sigma mji_e(209) = 10965$$

COMPARISON OF CONDITIONS OF GREATEST FLEXURAL STRESS SHOWS THAT EMERGENCY BRAKING PLUS WEIGHT IS THE MORE SEVERE CONDITIONS.
THE GREATEST FLEXURAL STRESS OCCURS UNDER OPERATING CONDITIONS OF 1.00 g downward and 0.70 g braking

CONCLUSIONS :
TANK SHELL MATERIAL IS NOT OVERSTRESSED BEYOND (*dsf*=4) ALLOWABLE DOT LIMITS (49CFR) UNDER CERTAIN OPERATING AND EMERGENCYCONDITIONS APPLIED TO DOT 400 SERIES CARGO TANKS. Revision Note: Ommision of Compression Struts From Tank Section Calculations Results In Conservative Values for Stress Calculations.

SUMMARY of MAXIMUM STRESS POINTS

$\sigma j_e\left(\frac{L}{2} + 28.60259\right) = 10647$

$\sigma mji_e(209) = 10965$

$\sigma m(186) = 10933$

$p := 35 \qquad D := 50.5 \qquad t_s := \frac{3}{16}$

$\sigma_p := p\cdot\left(\frac{D}{2}\right)\cdot\frac{1}{t_s} \qquad \sigma_p = 4713$

$\sigma := \sigma_p + \sigma mji_e(209) \qquad \sigma = 15679$

$\left(\sigma \leq \frac{Su_T}{dsf}\right) = 1$

original calculations performed for CTEC in Lubbock, Texas
revised/reformatted for CONTAINER TECHNOLOGY INCORPORATED

Thomas A Rogers P E 9/27/00 date
Texas Registration 79212

THOMAS A. ROGERS
79212

revision note: recaptioned & reformated for binding; no calculation revisions

NFS PART 3
CURRENT 49CFR RULES APPLIED TO VESSLEL STRUCTURAL INTEGRITY RELATED TO SURGE IN ACCIDENT CONDITIONS

NUCLEAR FUELS SERVICES
3700uswg CARGO TANK EFFECTS OF TANK WALL STRESS FROM MAXIMUM ALLOWABLE WORKING PRESSURE PLUS A LIQUID SURGE at TWO g DUE TO SUDDEN DECELERATION

shell volume, uswg

$V := 3700$

diameter shell, in

$D := 50.5$

capacity of shell, uswg per inch

$gpi := \left(\frac{1}{231}\right)\cdot\left(\frac{\pi}{4}\right)\cdot D^2$

$gpi = 8.671$

% fill by volume

$fd := 0.97$

gallon weight of lading, lb

$wg_{lading} := 13$

shell volume area. sq in

$A_s := \left[\left(\frac{\pi}{4}\right)\cdot D^2\right]$

$A_s = 2002.962$

lading weight, lb

$W_{lading} := V\cdot\left[fd\cdot\left(wg_{lading}\right)\right]$

$W_{lading} = 46657$

volume of lading, uswg

$V_{lading} := V\cdot(fd)$

$V_{lading} = 3589$

tank MAWP, psi

$p := 35$

height of lading, in (if vessel is vertical)

$h_{lading} := \frac{V_{lading}}{gpi}$

$h_{lading} = 413.917$

surge load factor

$N_{surge} := 2$

load factor modified weight of column, lb

$W_{surge} := \left(N_{surge}\right)\cdot W_{lading}$

$W_{surge} = 93314$

weight on one inch area, lb

$p_{surge} := \frac{W_{surge}}{A_s}$

$p_{surge} = 46.588$

shell thickness, in

$t_s := \frac{3}{16}$

hoop stress at p_{surge}

$\sigma := \frac{\left(p_{surge} + p\right)\cdot\left(\frac{D}{2}\right)}{t_s}$

$\sigma = 10987$

yield strength, psi of SA-240 304 ss

$Sy := 30000$

two g sure pressure is less than 0.75% of yield material of material used

$\left[\sigma \le \left(\frac{3}{4}\right)\cdot Sy\right] = 1$

CONCLUSION :
TANK MAWP PLUS A TWO g LIQUID SURGE DUE SUDDEN BRAKING DOES NOT ELEVATE THE HOOP STRESS TO A VALUE GREATER THAN 75% OFTHE YIELD STRESS OF THE TANK MATERIAL

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original calculations performed for CTEC in Lubbock, Texas
revised/reformatted for CONTAINER TECHNOLOGY INCORPORATED

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Part 4

DAMAGE EFFECTS TO 3700 uswg STAINLESS STEEL MC311 CARGO TANK SEMI-TRAILER FROM ONE FOOT DROP ON TO THE LANDING GEAR LEGS

APPENDIX

4-A
DELETED

4-B
DYNAMIC FINITE ELEMENT ANALYSIS
USING DINAMIKA-3 COMPUTER CODE

DINAMIKA-3 COMPUTER CODE VERIFICATION
Sarov Open Computing Center
Sarov, Russia

4-C
STATIC NONLINEAR FINITE ELEMENT ANALYSIS
USING ANSYS COMPUTER CODE
ADAPCO
Melville, New York

4-D
REFERENCES

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NUCLEAR FUEL SERVICES

Part 4

DAMAGE EFFECTS TO 3700 uswg STAINLESS STEEL MC311 CARGO TANK SEMI-TRAILER FROM ONE FOOT DROP ON TO THE LANDING GEAR LEGS

ABSTRACT

Definitions and criteria for survivability of a stainless steel cargo tank semi-trailer in hazardous material service are established to judge the damage effects of an accidental one foot drop on to unextended landing gear legs. Plastic deformation of tank reinforcing rings is the primary means of energy transfer and absorption into the tank. The criteria for survivability is lading retention integrity of the tank vessel, which is seen to be equivalent to non-rupture of tank and rings. This criteria is further refined to a comparison of ring collapse strain energy with potential energy of a one foot drop.

A classical analysis was attempted and rejected because of it's difficulty and requirement of excessively conservative and simplistic assumptions. However, a preliminary nonlinear static overload finite analysis successfully tested the hypothesis underlying the abandoned classical analysis. The data used was conservatively approximate due to the preliminary "trial" nature of the analysis, The results verified and strengthened the original hypothesis underlying the classical attempt, despite conservative and unrealistic assumptions that the tank and rings absorb all drop energy. Despite it's limitation of scope, this static method is included for reference. The results provided justification for the employment of more sophisticated and powerful computer models.

A state of the art dynamic finite analysis method was employed that included all tank components along the load path from the landing gear legs up into the liquid filled tank. The method accounts for absorption of drop energy into all components in the model. The results of this analysis indicate deformations of noncritical components in the load path absorb most of the drop energy. The lading retention integrity of the tank remains unaffected by a wide margin of safety.

The two FEA methods give different numerical results, due to differences of assumptions and differences of scope and calculation prowess of the computer codes. The results are in fundamental agreement regarding outcome of the accident. Ring and/or tank rupture is not to be anticipated as a result of such an accident. However repair of extensive deformations should be expected.

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Part 4
DAMAGE EFFECTS TO 3700 uswg STAINLESS STEEL MC311 CARGO TANK SEMI-TRAILER FROM ONE FOOT DROP ON TO THE LANDING GEAR LEGS

COMMENTARY & CONCLUSIONS

Summary of Methods and Conclusions

A finite element model of the tank trailer was used to analyze the dynamic effects of an accidental drop of the loaded tank trailer onto unextended landing gear legs. The computer program used was developed for solutions of problems requiring accommodation of large nonlinear deformations, such as would occur in a car crash, or the analysis of the effect of ordinance impacts on targets.[1]

All components in the path of impact load, beginning with the upper (outer) landing gear tube, upward into the liquid filled shell were analyzed in the model. The failure criteria for the survival of the tank in this accident analysis is defined as failure to meet either or both of two conditions:

1) The primary failure criteria for any given component is defined as deformation beyond the acceptable strain limits for the material of that component. Strain beyond acceptable limits of the tank shell or "hat section" shell reinforcing rings would indicate a tearing or rupture that would compromise the lading retention integrity of the tank. Rupture is indicative that the tank trailer "failed to survive" the accident.

2) A second failure criteria condition requires that summation of the strain energy of the components of the entire unit (more narrowly limited to all components in the load path) must be equal to the drop energy. Analysis of this "failure" condition can be done two ways:

a. The components can be dynamically analyzed as a whole on the assumption the drop energy is dissipated (the energy equality condition is inherent in the program). Component failures occur when strain exceeds allowable values for the material of construction.

b) Every component can be postulated to "fail" at the limits of strain. The summation of strain energy must exceed the drop energy to avoid "unit failure".

---

[1] The dynamic FEA model was constructed using computer code DINAMIKA-3 by SAROV OPEN COMPUTING CENTER in Sarov, Russia. The analysis is attached a Appendix Part 4-B. Also attached in that appendix is a verification for DINAMIKA-3, This program employs state of the art dynamic impact analysis methods and is capable of dealing with large plastic deformations. This code,developed in Russia, performs tasks similar to those computer codes developed for the same purposes in the USA at Lawrence Livermore Labs.

Method 2)-a. was used. It conforms more closely with "reality", but powerful programs are required.

The results of this FEA analysis indicate that the tank's lading retention integrity is not threatened in such an accident, because most of the energy of the drop is absorbed in landing gear support structure deformation. A minimum stiffening of this structure is desirable to avoid it's complete failure. The transfer of energy to the tank, resulting from such modest and reasonable improvement, will not be sufficient to affect the lading retention integrity of the tank in such an accident.

An initial attempt to employ method 2)-b by classical methods proved to be extremely difficult and was rejected. An ANSYS[2] nonlinear static program was employed to provide a solution using this failure criteria. This analysis was limited to static overload of the tank shell, rings and bolsters. The summation of strain energy exceeded the drop energy, indicating "survival" of those components. The results of this program indicated that expense of the more realistic and robust DINAMIKA-3 program was justified.

---

[2] The ANSYS program used by ADAPCO of Melville, NY, is nonlinear. The particular version used cannot accommodate strains that change the geometry of the model.

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COMMENTARY & CONCLUSIONS

Commentary on Analysis Methods Used and Results

The most likely circumstance under which a cargo tank trailer will sustain damage from being dropped occurs when a truck driver absentmindedly drives out from under a parked trailer without extending the landing gear legs. This seldom occurs, but when it does, some part of the trailer will be damaged. Anecdotal evidence indicates that the tank will not be ruptured.

Since no analytical method can precisely predict damage in an accident, it is necessary to establish failure and survivability criteria for analysis to be meaningful. In this case, since tank trailer damage is a foregone conclusion, the criteria for "survivability" can be reasonably be defined as "maintenance of the lading retention integrity of the tank vessel", i.e. the tank is considered to have survived the accident if it does not rupture and allow the contents to leak out.

The structural integrity of the NFS cargo tank trailer is dependent upon reinforcing rings that are integrally attached to the outside of the tank shell. Under all conditions of support, the tank acts as the structural equivalent of a simple beam. [It is either supported by the rear tandem wheels and the front fifth wheel connection while being pulled by a truck, or it is supported by the same rear tandem wheels and the landing gear legs while unattached to the truck.] At all supported points, the weight of the loaded tank is transferred to the support structures through the tank reinforcing rings. Collapse of these rings is tantamount to collapse of the tank, since the tank, after losing it's "roundness", has no cross sectional stiffness and cannot function as a beam. Lading integrity of the vessel is considered to be equivalent to non-collapse of the rings.

The maximum energy that must be absorbed is the drop energy.[3] The first attempt to analyze the damage consisted of postulating a collapse of the rings from a static overload and determining whether the resulting strain energy exceeded the drop energy. Elastic analysis using curved beam formulae (see reference, *Roark*) were employed to determine points of maximum moment. It was postulated that these points, if subjected to increasing static loads to beyond the yield point, would behave as plastic hinges until a "collapse mechanism" occurred. [This method is far from reality, since no dynamic effects are considered.] An attempt was then made to estimate the plastic strain energy expended just prior to failure of the "collapse mechanism". If summation of strain energy ring and bolster collapse exceeded drop energy, it could be concluded that the rings would survive without rupture. This analysis method postulates failure to start with, sums strain energy as a test and then rejects the failure hypothesis if strain energy exceeds drop energy by a reasonable margin.

---

[3] Drop energy is defined as static weight in pounds on the landing gear, when parked without a truck, multiplied by the drop distance.

This "classical" method was found to be impractical for several reasons:
  a) The complexity of the changing geometry while each component was undergoing plastic deformation;
  b) The existence of "parallel" load paths and the complexity and doubts of the accuracy of assumptions required to apportion strain energy amongst these load paths at any stage of deformation:
  c) The difficulty of establishing analytical formulae for "area under the stress-strain curve"
  d) The repetition of calculations required to re-compute the altered geometry after each increment of deflection;

This method was abandoned because it was extremely difficult and time consuming. Simplifying assumptions necessary to accomplish these calculations proved to be so conservative as to make the results irrelevant. (The calculations and conclusions for this method were originally designated as 4-A in the appendix when the report was formatted, but the analysis was withdrawn from the Part 4 Appendix and does not appear there.)

A preliminary FEA analysis (Part 4 Appendix 4-C, attached) was employed as an alternative method to test the hypothesis underlying the "classical" attempt just described. The model of the trailer along the load path into the tank was analyzed. This load path is summarized as follows and includes:

1) the bottom end of each outer landing gear tube
2) the top end of each landing gear tube and the attachment bolts
3) each landing gear structure longitudinal beam to each end of each beam,
4) through several parallel load paths into the tank rings, these load paths as listed:
    a) side-cradle "bolsters" into the tank through continuous weldments to reinforcing rings, (these connections located at "four o'clock and seven o'clock", in a view of the tank cross section).
    b) plate cradle under each ring, welded continuously to bottom one-third circumference of each ring, thence to the side cradle bolsters described above,
5) tank rings integrally welded to the tank shell. [The tank shell and "hat shaped" rings together are considered to act as a single shape regarding stiffness up to a width of 40 times the tank shell thickness; see ASME and 49CFR]
6) tank shell, as containment
7) liquid load (lading)

The FEA program employed for this alternative analysis (ANSYS see footnote 2) is non-linear, thus allowing the analysis to proceed beyond yield. However the computer code used does not allow for the significant deformations required for failure at the strain limits of ductile failure. (21% for carbon steel to 40+% for stainless steel). Failure criteria for this method is imposed by the computer code and can be characterized as "excessive deformations". [The ANSYS deformations proceeded to a point just short of disqualifying the model's geometric integrity.] In addition to the obvious advantages of this FEA method, it's usefulness and advantage over the abandoned classical approach is the apportionment of loads along parallel load paths. The abandoned "classical" attempt and the alternative ANSYS program each utilize the same fundamental assumption, namely:

All energy absorption is confined to strain energy in items connected directly to tank and rings [items 4)a), 5) & 6)] listed above, except the ANSYS model also included item 4)b) above.

The "classical" attempt accomplished this limitation of load path components by just ignoring everything else; the ANSYS model included every load path item listed, but the "non-included" components were set to infinite stiffness, rendering them incapable of exhibiting strain energy.

Results of the "classical" analysis barely yielded enough strain energy to demonstrate the hypothesis. In contrast, the ANSYS model results indicated that even small nonlinear deformations, were, when summed, sufficient to produce strain energy exceeding the drop energy. Although the use of the ANSYS model verified the classical assumption, it was flawed to the extent that the material properties of the model did not correspond exactly to the properties actually used in construction according to blueprints furnished by the owner. Since these discrepancies erred on the conservative side, it was determined that this flaw, in what started out to be a "quick and dirty" inquiry, was a "defect of form over substance". [The ANSYS model analysis and results are included as an attachment labeled Appendix 4-C.]

Rather than spend further resources on a method limited to solution of a wholly artificial problem unconnected to actual events, it was determined that a dynamic nonlinear model, having capacity for extreme deformations, would yield answers from questions closer to "reality". Furthermore, use of a better (more powerful computer code) model would make practical the inclusion of consideration of the practicality of including all items along the load path.

The entire load path was modeled, using a program called DINAMIKA-3 (see footnote 1). The conclusion of this analysis is that the greater part of the drop energy is absorbed in the deformation of the landing gear support structure. The lading retention components do not indicate excessive strain. This analysis and the results appear in attached appendix 4-B, along with verification documentation. [Corrected data was used as well as actual measurements of certain landing gear support structure components that were unclear in definition or omitted on the trailer drawing.] See Fig. 1, 2, 3A, 3B, and 4. BY NSS.

Examination of the results of the DINAMIKA-3 analysis results indicates that certain landing gear support structure components are on the brink of failure, i.e. ancillary braces and connections show strains exceeding the accepted values for the particular material used. However there are several points to be made regarding these items.

1. The endangered components are not essential to the support of the trailer; this is not their primary purpose.

2. Even the "realistic" DINAMIKA-3 model analysis is conservative in consideration of energy absorbing components; several significant components and systems were omitted for reasons of simplicity and/or cost. Notable items excluded from the analysis are
    a. the entire energy absorbing system of the rear tires and springs [Although these items are simplistically assumed to be "immovable", they in fact do move short distances when a trailer is dropped onto it's landing gear, even when the drop is "normal", i.e. less than one foot. Springs, tires, and associated equipment are in fact, either energy absorbing, or energy distributing and/or delaying devices.]
    b. lower (inner) landing gear tubes and landing gear wheels, bearings and pins.
3. It is a simple matter to reinforce or replace the landing gear support structure longitudinal beams. Such an alteration could be regarded as almost a trivial consideration in comparison to the costs of replacement of the unit.

Any stiffening of the landing gear support structure will transfer some extra energy drop energy to the tank. The support structure now acts as a dynamic and structural "fuse" that absorbs nearly all the drop energy and more than adequately protects the tank. Common sense consideration of the DINAMIKA-3 results indicates that the tank's lading retention integrity is not even remotely threatened and that a modest strain energy transfer to the tank is acceptable. If a quantitative justification is required, it is noted that the survival of the alternative ANSYS model (at a postulated wholly unrealistic level of 100% strain energy absorption of the tank and rings, at modest deformations), is a limiting condition of unfavorable energy transfer. The ANSYS model analysis results indicate the tank is more than capable of assuming a somewhat larger share of strain energy without undesirable consequences.

CONCLUSION

There is a high probability that the loaded trailer will survive an accidental one foot drop onto unextended landing gear legs with no loss of lading due to tank rupture.

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Further Commentary on the Finite Element Methods Employed and Significance of Results.

Following completion of the two methods, it was discovered that data used in the models differed from the construction of the actual trailer.

1) So called compression struts are referred to on the drawing but not explicitly shown. After conferring with LBT Fruehauf, it was learned that these members stiffen the trailer longitudinally; they make no contribution to the ring stiffness.

2) The finite element models were run under the assumption of type 316 stainless steel for the shell and rings. The rings are in fact constructed of type 201 stainless steel and the shell is type 304. Type 201 has properties of yield strength, ultimate strength, and elongation that exceed both type 316 and type 304, both of which are listed to have equal properties of yield strength, ultimate strength, and elongation. Having discussed these discrepancies with engineers from ADAPCO, it was concluded that the differences are not significant in relation to the models or the outcomes, and err on the conservative side.

The material specifications corrections were changed before the final SAROV FEA calculations were performed, but were not made for the ADAPCO report since that analysis was a trial analysis, is limited in scope and is regarded as not affecting the final results and conclusions.

(

FIGURE WITHHELD UNDER 10 CFR 2.390

Fig. 1
TANK REPRESENTED AS A SIMPLY SUPPORTED BEAM

tank as beam rvsd ***yl*** *1003*

FIGURE WITHHELD UNDER 10 CFR 2.390

Fig. 2
HAT RING SECTION
continuously welded to shell

(

Y

FIGURE WITHHELD UNDER 10 CFR 2.390

Fig. 3 A
LANDING GEAR SUPPORT STRUCTURE
and BOX BOLSTER (side cradle) DETAIL

FIGURE WITHHELD UNDER 10 CFR 2.390

t.•

t.

.ANDING GEAR SUPPORT STRUCTURE

neasurements by NFS October 2001

FIGURE WITHHELD UNDER 10 CFR 2.390

NOTES
tube (1) area=3.387sqin
total area both legs 6.67sqin
material properties as follows:
carbon steel tube *
   50,000psi yield
   70,000psi ultimate tensile
   22% elongation
bolts: A490 1/2 in,
   130,000psi yield
   150,000psi ultimate tensile
outer tube stabilized by inner tube
[* data by landing gear manufacturer I

Fig. 4
OUTER **LANDING TUBE**

NFS landing gear tube rvsd yl 1004

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ANALYSIS
Fruehauf MC 311 Insulated Stainless Cargo Tank Semi-Trailer

Part 4
Determination of the effects of a one foot drop in the upright position

Appendix 4-D

REFERENCES

American Institute of Steel Construction, *Manual of Steel Construction*- *Load & Resistance Factor Design* *Volume II Connections*, Second Edition, Chicago

The American Society of Mechanical Engineers, *ASME Boiler and Pressure Vessel Code, Part A Ferrous Material Specifications* and *Part D, Properties* , *1988 Code, 1999 Addenda*, New York

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FRUEHAUF CORPORATION (now owned by LBT), *Drawing UNF 2232 - Rev. A*, Omaha, NB

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[NOTE: Although this version of 49CFR is not the official set of volumes, it consolidates the highway hazardous regulations into one volume, Furthermore. since it is periodically republished in new editions, one can refer to older versions to discover the history and previous wording of any regulation. a service not furnished by DOT. This volume is reported to be the "preferred reference" of DOT field inspectors who find the single book to be convenient.]

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United States Department of Transportation *49 CFR*, Washington,DC

AGREEMENT NO OCC-002/2001

REPORT NO 7
NOVEMBER 2, 2001

DYNAMIC ANALYSIS OF NUCLER FUEL SERVICES 3700 uswg STANLESS STEEL INSULATED CARGO TANK SUBJECTED BY ONE FOOT DROP

PREPARED FOR:
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Sarov, Russia
2001

TABLE OF CONTENTS

1.0 INTRODUCTION

The numerical simulations results of truck trailer tank subjected to one foot drop onto a hard surface are presented in this Report. The results of previous simulations are presented in [2]. New data of geometry and material properties were supplied by **adapco** e-mail of October 9, 2001. Simulations were curried out by Russian explicit dynamics computer code DINAMIKA-3, which has a Certificate of Russian GOSSTANDART No POCC RU.ME20.H00338, No 0090655 [1].

A truck trailer tank is being used to transport radioactive waste. The waste is dissolved in nitric acid. Thus the tank is 95% full of nitric acid. The postulated failure mechanism is that the driver of the truck unhooks the "fifth wheel" and drives off without rolling down the landing gear. As he drives off, the fully loaded tank then suddenly drops one foot (12 inches or 305 mm). The analysis objective is to determine if this event will cause the tank to rupture.

The overview of the structure shows that there are several elements in the system, which can absorb the kinetic energy during the interaction of construction with hard surface. They all should be included in computer models. They are:

- Tank's cylindrical shell stiffening by rings:
- Support frame structure;
- Outer landing gear legs;
- Bolts, which fix the legs to the frame.

There are two features of the problem: a plane of symmetry in the lateral direction and the liquid in the tank. The liquid can be included in the model directly but in this case it takes much more time to get a solution, because the time step of numerical integration becomes very short. The preliminary numerical investigations of the problem have demonstrated that it is quite acceptable only to include the weight of the liquid in analysis.

New model of the tank have been made on the basis of Model 3 from [2], by changing the dimensions of:

1) Hat cross section rings;
2) Box bolsters;
3) Support frame,

and adding following new elements:

1) Compression strut on the top of the tank
2) Cross brace channel in the support structure
3) Leg braces

In the model only two rings are attached to the landing gear assembly were included in the models, because they determinate the local stresses in tank skin. All other rings affect only on stress level caused by action of liquid inside the tank. The level of this stress is very small [2], so by this reason other stiffening rings were included only in the weight of the system.

All models are built of 8-nodes solid finite elements with special high accuracy approximations of strains and stress inside elements.

The weight of the fluid was included in the density of the bottom half of the tank's shell.

The bolts were not preloaded, because the level of preloading was unknown. There is no any energy being dissipated by damping.

2.0 ANALYSIS OVERVIEW

This section provides a description of problem data (dimensions, weights, etc) and material properties.

2.1 Data of Structure

The construction of a truck trailer tank consists of parts with dimensions (See Figures) as follows:

- Tank's Cylindrical Shell

Diameter $D = 1273$ mm (50.5 in);

Length $L = 10693$ mm (421 in);

Thickness $t_o = 4.76$ mm (3/16 in);

- Stiffening Ring

Width $a_1 = 76.2$ mm (3.0 in);

Height $a_2 = 66.675$ mm (2 5/8 in);

Thickness $t_1 = 3.57$ mm (0.1406 in);

- Support Frame Structure

Width $H = 914$ mm (36 in);

Length $L = 1244$ mm (49 in);

Thickness of inner cross member $t_2 = 4.76$ mm (3/16in);

Thickness of beam wall $t_3 = 4.76$ mm (3/16 in);

Thickness of walls $t_5 = 4.76$ mm (3/16 in);

Outer Landing Leg

Tube cross section sizes $b_1 \times b_2 = 127$ mm x 127 mm (5in x 5in)

Thickness of the outer tube $t_4 = 4.57$ mm (0.180 in)

Length of the outer tube	b_3 = 762 mm (30 in)

- Bolts (six at each leg)

Diameter d = 12.7 mm (½ in)

The weight parameters of system components are as follows:

The total weight of loaded trailer	G_0 = 29060 Kg (64,000 lbs)
The weight without tandem	G_1 = 26830 Kg (59,100 lbs)
The density of the liquid inside	p = 1497.8 Kg/m^3 (12.5 lb/gal)
The density of steel structures	r = 7800 Kg/ m^3

2.2 Material Properties

For all materials in this analysis the elastic-plastic model of deformation with bi-linear isotropic hardening was used based on following data:

- Tank's Cylindrical Shell is made of stainless steel **304SS**:

Young's Modulus	20,000 Kg/mm2
Tangent Modulus	81.03 Kg/mm2
Poisson's Ratio	0.3
Yield Strength	23.25 Kg/mm2
Ultimate Strength	55.66 Kg/mm2
Elongation	40%

- Hat Cross Section Rings and Compressed Strut are made of steel **201SS**:

Young's Modulus	20,000 Kg/mm2
Tangent Modulus	76.86 Kg/mm2
Poisson's Ratio	0.3
Yield Strength	38.75 Kg/mm2
Ultimate Strength	81.02 Kg/mm2
Elongation	55%

- Cradle Plates and Box Bolsters are made of **H.T.S.**:

Young's Modulus	20,000 Kg/mm2
Tangent Modulus	46.4 Kg/mm2
Poisson's Ratio	0.3
Yield Strength	37.70 Kg/mm2
Ultimate Strength	49.30 Kg/mm2
Elongation	25%

- Support Frame Structure are made of steel **A36**:

Young's Modulus	20,000 Kg/mm2
Tangent Modulus	77.5 Kg/mm2
Poisson's Ratio	0.3
Yield Strength	25.36 Kg/mm2
Ultimate Strength	42.24 Kg/mm2
Elongation	21.5%

- Outer landing tube are made of **Carbon Steel (C.S)**

Young's Modulus	20,000 Kg/mm2
Tangent Modulus	64.09 Kg/mm2
Poisson's Ratio	0.3
Yield Strength	35.20 Kg/mm2
Ultimate Strength	49.30 Kg/mm2
Elongation	22.0%

- Bolts (six at each leg) are made of **carbon steel**:

Young's Modulus	20,000 Kg/mm2
Tangent Modulus	70.05 Kg/mm2
Poisson's Ratio	0.3
Yield Strength	91.5 Kg/mm2
Ultimate Strength	105.6 Kg/mm2

Computer model of the structure and its parts with marked materials and dimensions of elements are presented on Figures 1-7

3.0 RESULTS AND DISCUSSION

This section provides the results of numerical simulations of the developed model. They are presented on Figures 8 … 26

Fig. 8 … 11 – Distributions of maximum plastic strain intensity in support structure;

Fig. 12 – Plastic strain intensity vs. time in points 1, 2, 3 4, 5, 6. The point's positions are presented on Fig. 6

Fig. 13 - Distribution of maximum plastic strain intensity in tank's shell;

Fig. 14– Distributions of maximum von Mises stress in the tank's structure (Kg/mm^2);

Fig. 15 - Distributions of maximum von Mises stress in tank's shell (Kg/mm^2);

Fig. 16 …18 - Distributions of maximum von Mises stress (Kg/mm^2);

Fig. 19 - Kinetic energy of the model vs. time;

Fig. 20 - Energies of the tank vs. time; [(1) – strain energy, (2) – kinetic energy, (3) – sum of (1) and (2)];

Fig. 21 – Vertical force on leg in the model vs. time;

Fig. 22 - Vertical force on leg in the model vs. time (in the time range from 0 to 30ms) ;

Fig. 23 - Vertical force time history of model at the tandem support point.

Fig. 24 – Vertical displacement of the tank's end vs. time;

Fig. 25 – Vertical displacement of the support structure's end (point A on Fig 6) vs. time;

Fig. 26 – Vertical displacement of the support structure's end (point B on Fig 6) vs. time;

An analysis of numerical simulation results shows that the duration of deceleration process is about 150 – 160 ms. In this period of time the kinetic energy of the system drops down from K =27.12 KJ to zero. Analysis of Fig. 20 shows that tank absorbs only 5 %, but support structure absorbs 95% of the kinetic energy of the system.

The reaction force in the leg reaches its maximum $F_{max} = 325 - 338$ KN in the time of about t = 18-19 ms and then slowly falls down to the near constant level of $F_s = 270$ KN, which keeps itself up to t = 150ms and then drops fast when the structure rebounds. The nominal static force of one leg is about $F_{st} = 93.4$ KN, so the maximum dynamic loading factor is $K_d = F_{max} / F_{st} = 338/93.4 = 3.6$. The nature of oscillations of reaction force on the leg during the first 1...1.5 ms (Figure 22) can be explained the same way as it have been done in [2]. Analysis of leg's displacements shows that during whole time interval 0...1.5 ms there is no gap between leg and rigid surface.

From Fig. 23 it can be seen that the trailer does not "bounce" off of the ground at the tandem location. From the results we can conclude that there is a quasi-static behavior of the system. The maximum vertical displacement of the end of the tank reaches a maximum value of 267mm (Fig 24). The ends of the support structure beam go down to 194 mm at point A and to 154 mm at point B (Fig. 6, 25, 26). It can be observed from the results of calculations that there are some buckling phenomena in the deformed frame structure and leg brace.

The most stressed element of the structure is horizontal beam of the support structure and leg brace (Fig. 8 ...11), where intensity of plastic stains reaches maximum values at points 1,2,3,4,5,6 (fig. 6). As it can be seen from Fig. 12 the maximum intensity of plastic strains in point 1 reaches the calculated level of 38.8 % which is much more then elongation of the steel A36 elongation (21.5%). It means that the connection between brace and leg will be lost during the accident. In points 2 and 6 the level of plastic strain intensity is about 21% and in points 3 and 5 it is about 15%. So we can conclude that in this points the local cracks can appear in the structure. The maximum level of the von Mises stress in most loaded bolt is 97.2 Kg/mm^2 (Fig. 17). It is more than Yield Stress –

91.5 Kg/mm^2, but it less that Ultimate Stress – 105.6 Kg/mm^2 of Carbon Steel from which one the bolts are made of. So there is no rupture of the bolts in this case.

The maximum intensity of plastic strains in tank's skin is no more than 0.4% (Fig. 13). So there is no rupture of the tank shell in this accident.

4.0 CONCLUSIONS AND RECOMMENDATIONS

1. The duration of the active impact continues about 160 ms. During the period of time vertical reaction force in outer landing gear leg increases up to the level of 325-338 KN (dynamic loading factor K_d =3.6). It causes large deformations in horizontal beam of support frame structure and kinetic energy of the system absorbs mostly by this element.
2. Maximum levels of plastic deformations intensity in element of the structure are as follows:

- Shell of the tank:	0.4% < 40% (304SS);
- Leg brace	38.8%> 21.5% (Steel A36)
- Horizontal beam	21% ~ 21.5 % (Steel A36)
- Outer landing gear leg:	1.4% (local) <22% (C.S.).

The maximum levels of calculated plastic deformations in leg brace exceed the elongation of material (A36) and it means that the connection between brace and leg will be lost in this accident. There are several points in horizontal beam where the maximum levels of plastic strain intensity near reach elongation of Steel A36 and may cause the appearance of local cracks in the elements of the structure, which would increase the danger of the accident.

On the basis of obtained results it can be recommended to increase the resistance of the horizontal beams of the support frame structure by adding new more strong elements to the existing structures.

5.0 REFERENCES

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2. NUCLER FUEL SERVICES 3,700 uswg STANLESS STEEL INSULATED CARGO TANK SEMI-TRAILER [MC311] EFFECT OF ONE FOOT DROP [ON LANDING GEAR] Report of SOCC, May 31, 2001

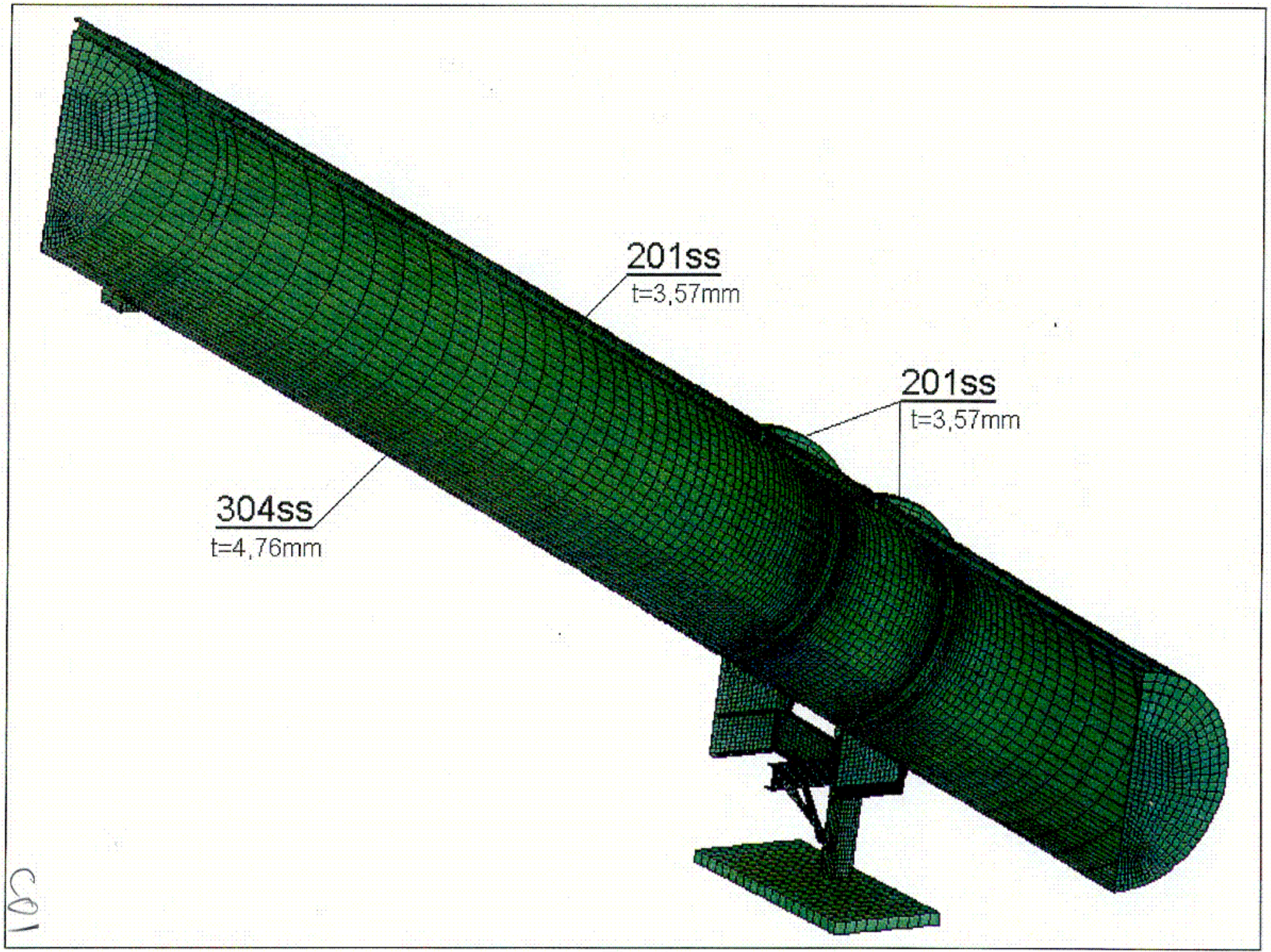

Fig. 1

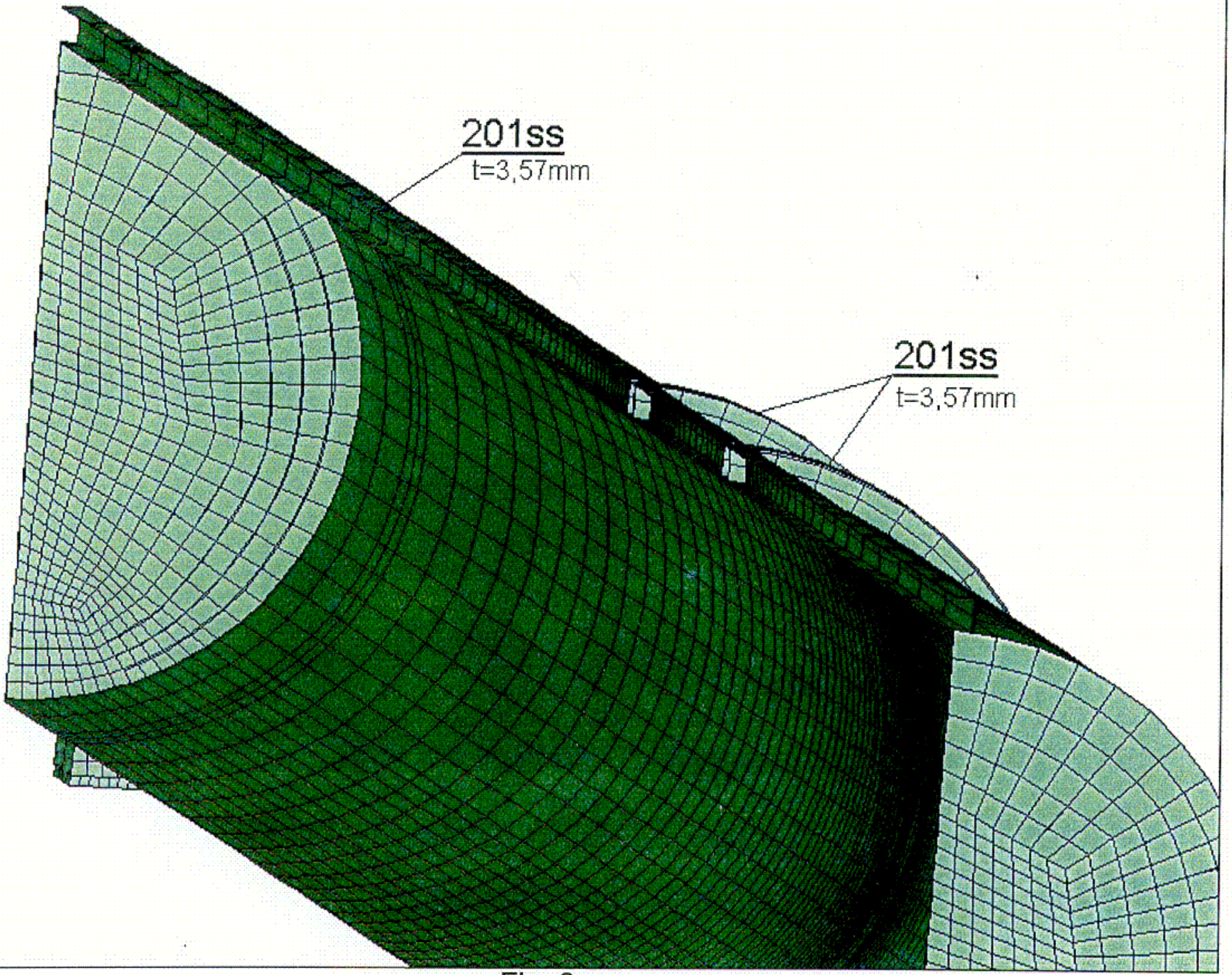

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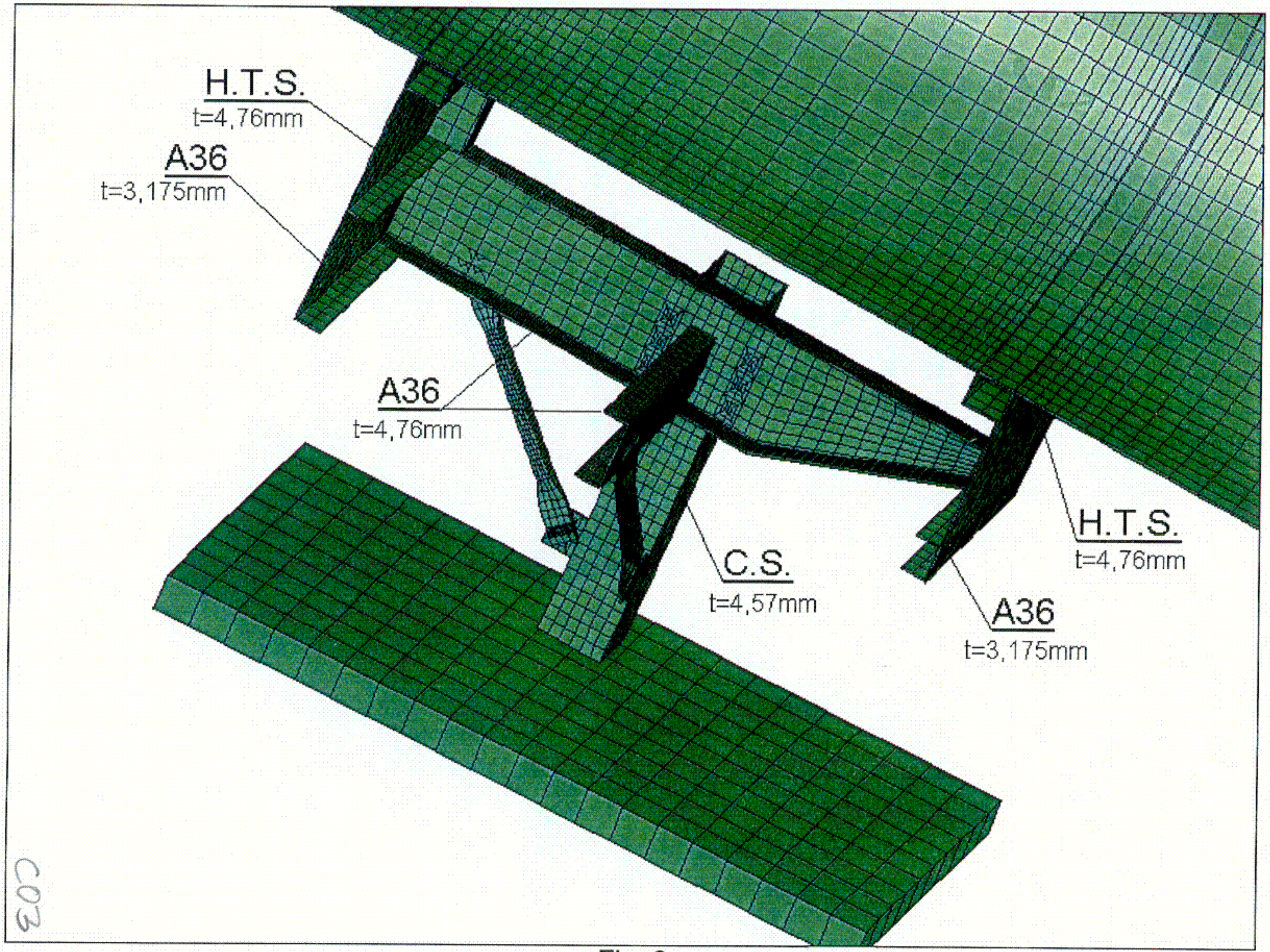
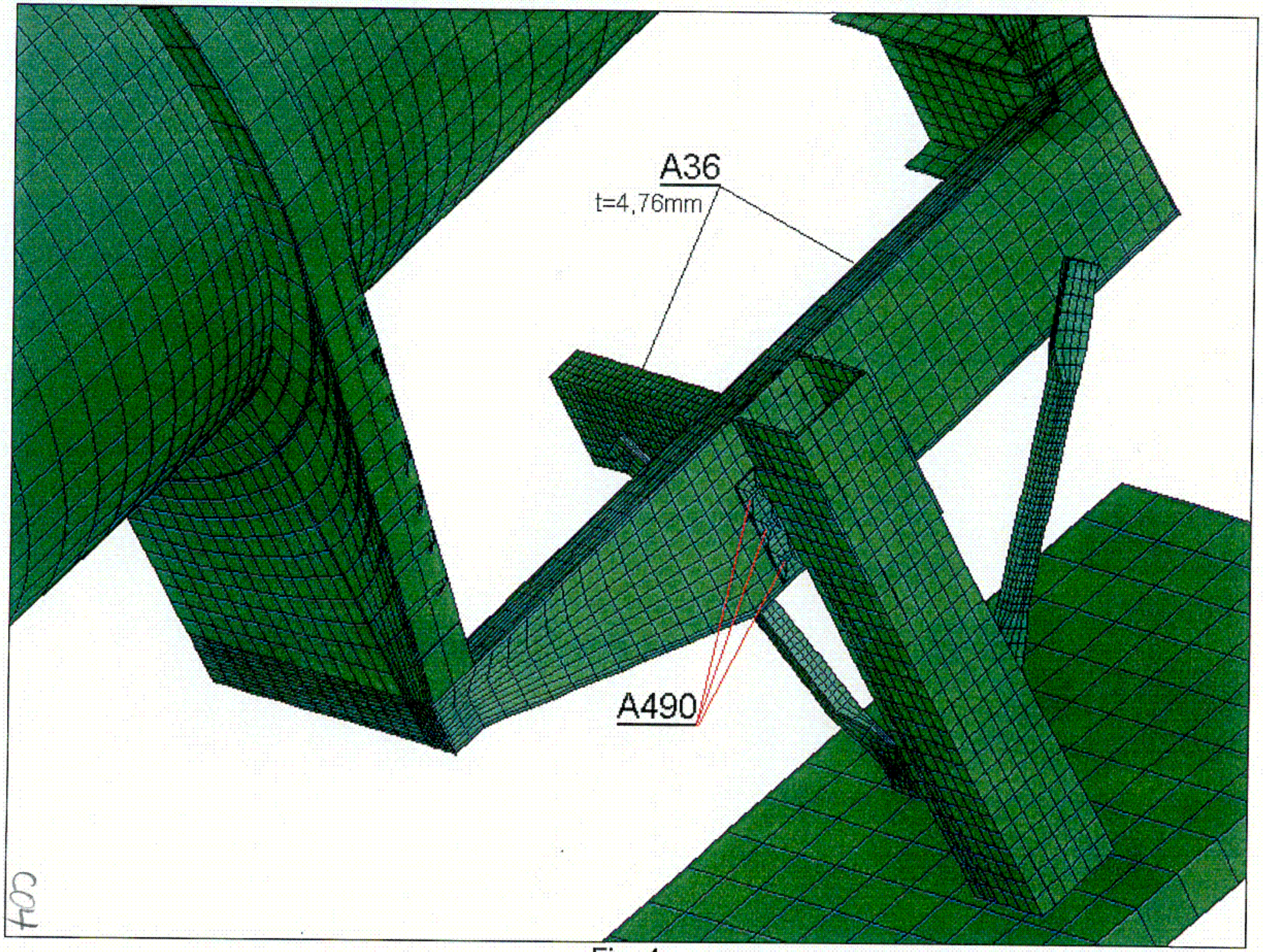
Fig. 2


H.T.S.
t=4,76mm
A36
t=3,175mm
A36
t=4,76mm
C.S.
t=4,57mm
H.T.S.
t=4,76mm
A36
t=3,175mm

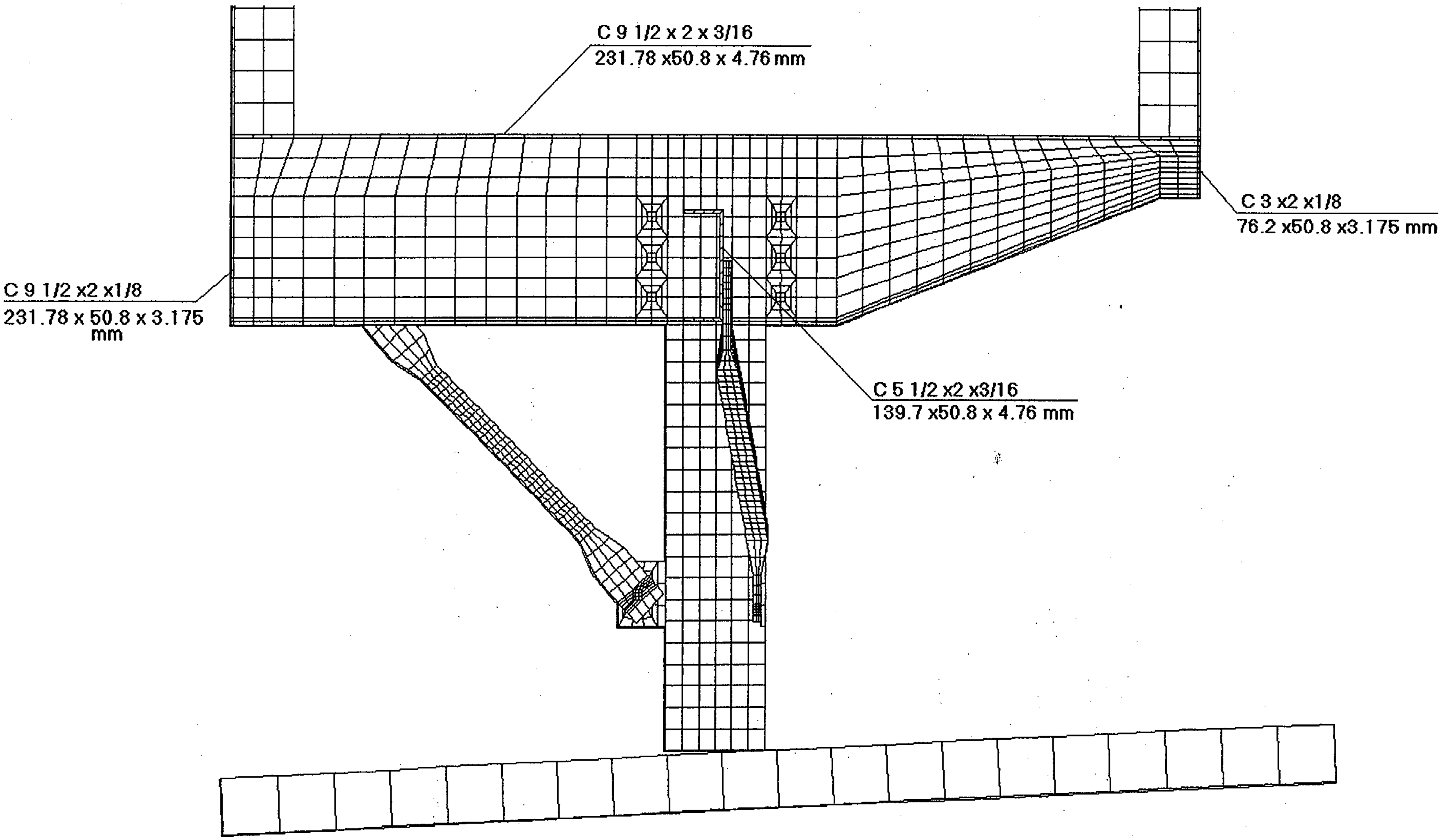
C03

Fig. 3



C04

Fig. 4



C 9 1/2 x 2 x 3/16
231.78 x50.8 x 4.76 mm
C 3 x2 x1/8
76.2 x50.8 x3.175 mm
C 9 1/2 x2 x1/8
231.78 x 50.8 x 3.175 mm
C 5 1/2 x2 x3/16
139.7 x50.8 x 4.76 mm

Fig. 5

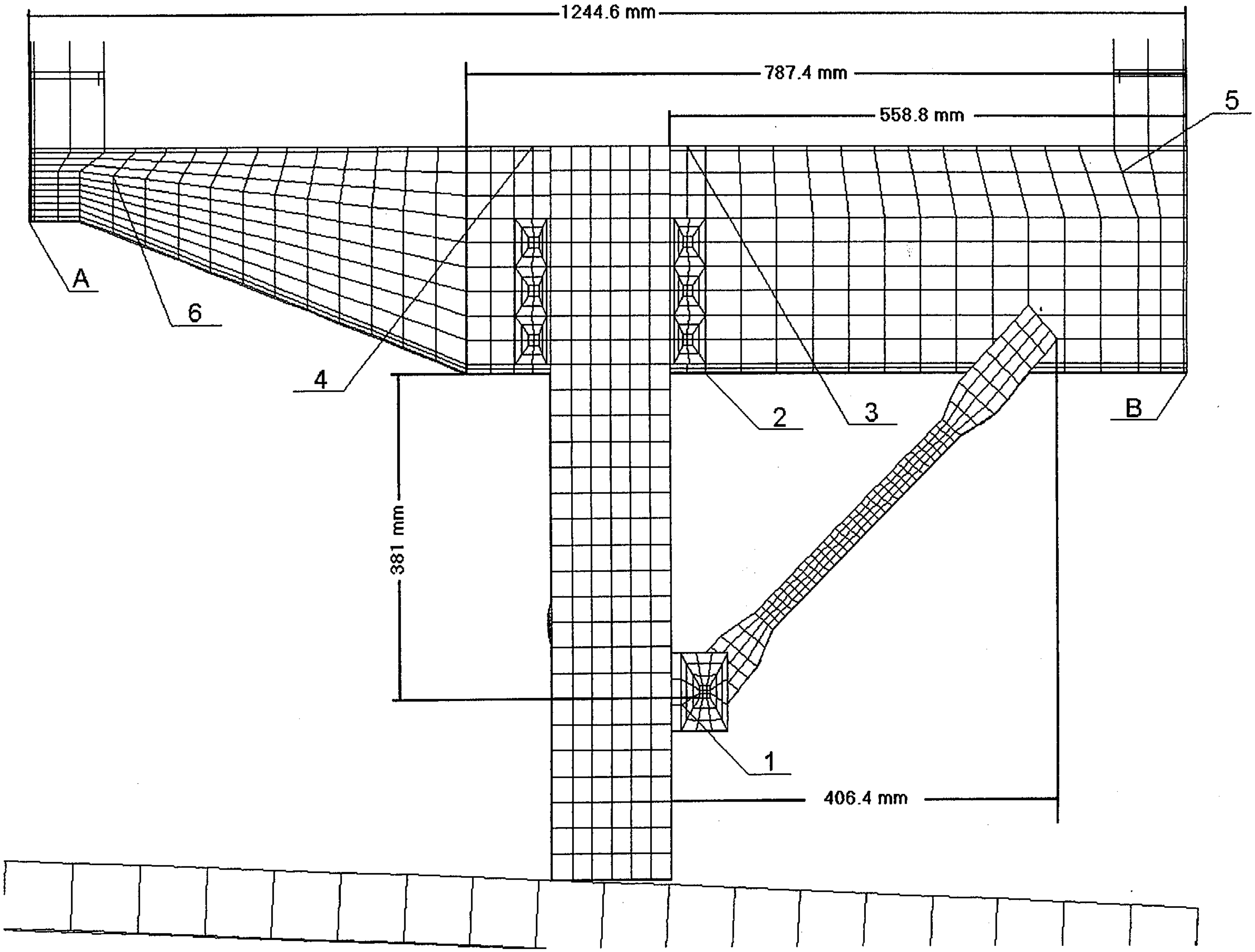


Fig. 6

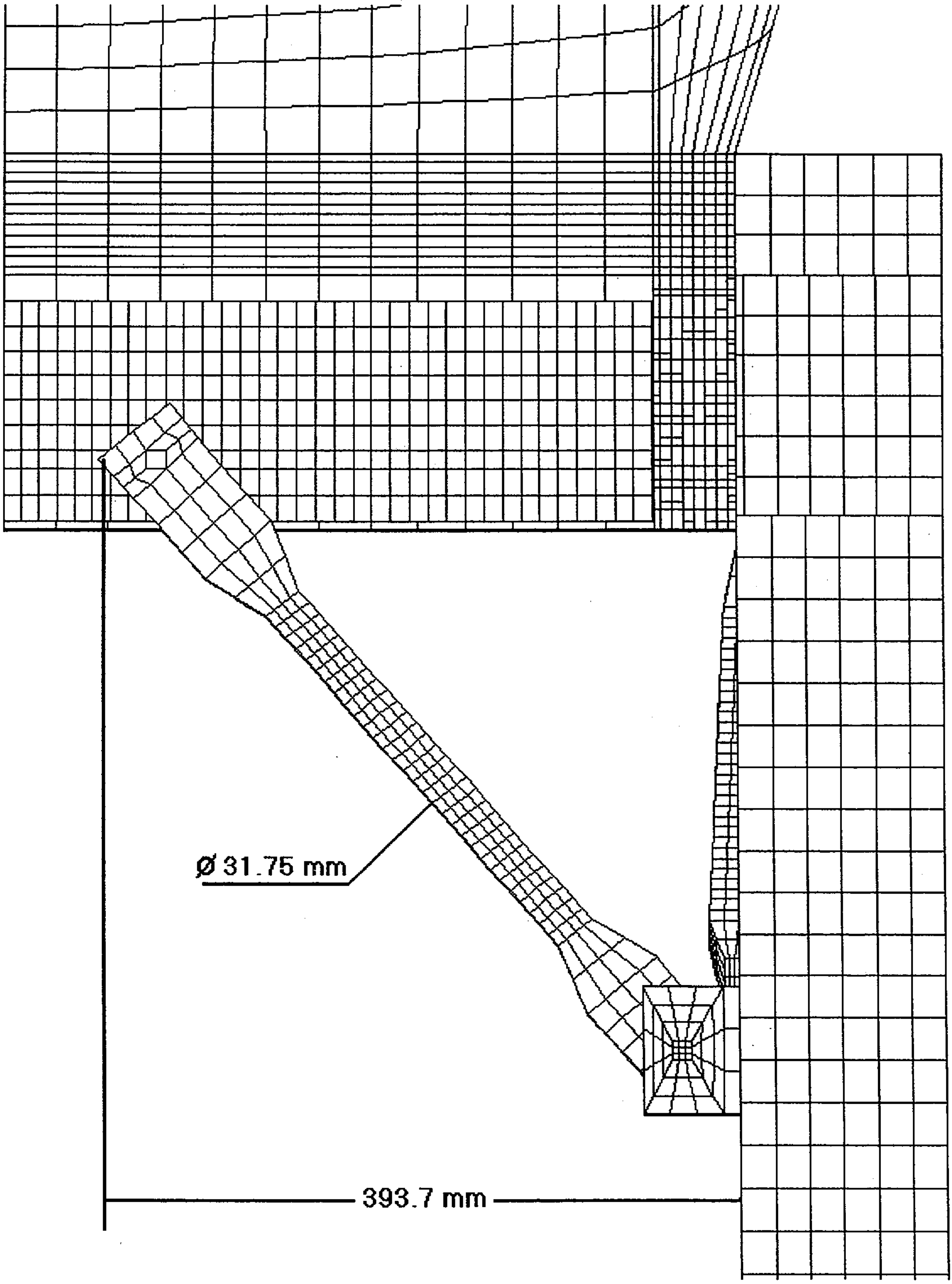
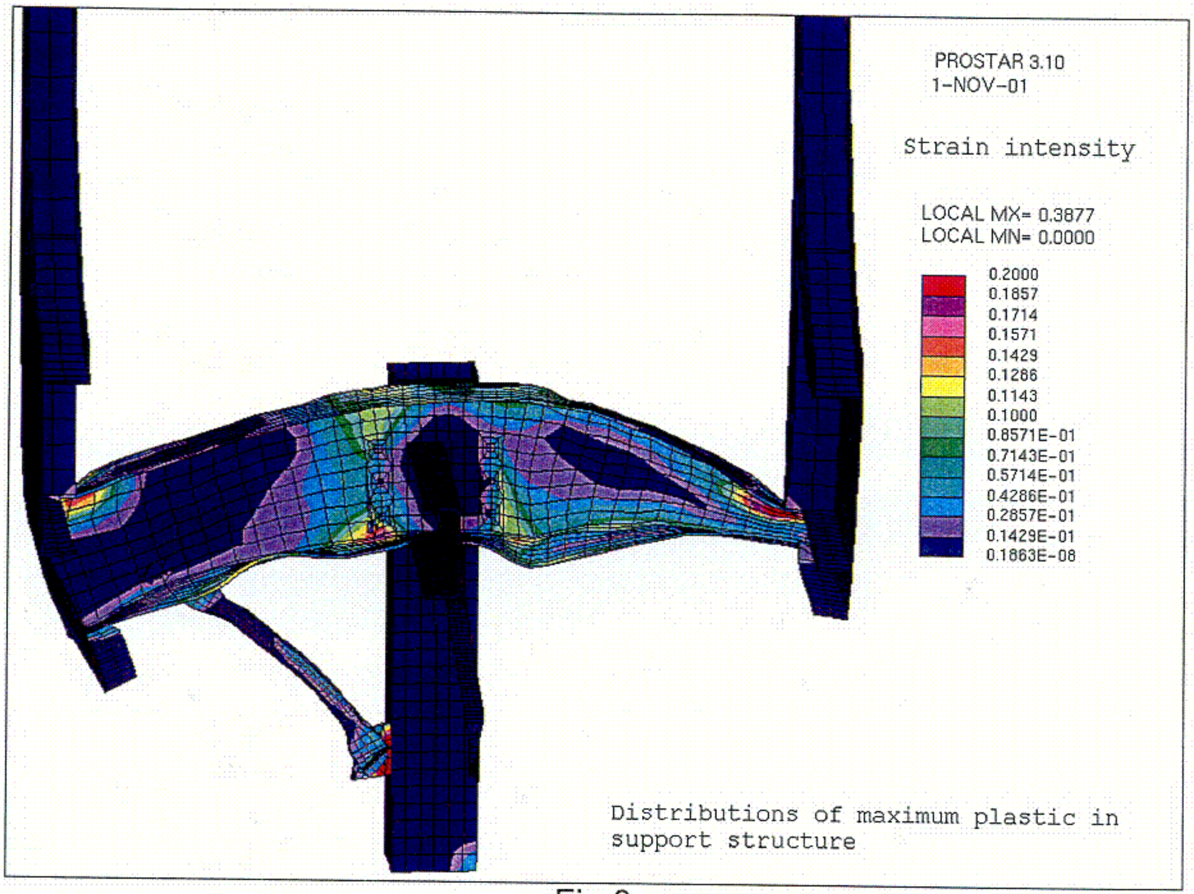
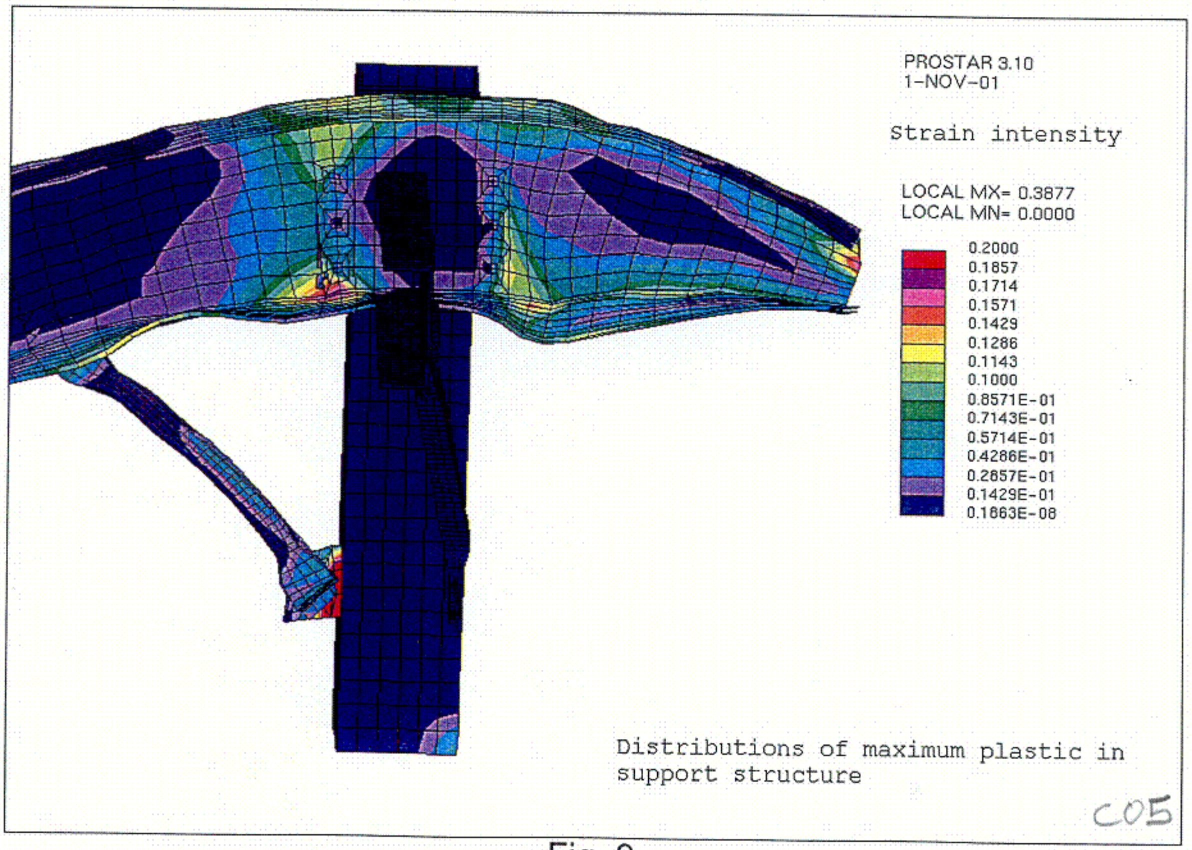


Fig. 7



Distributions of maximum plastic in support structure

Fig.8



Distributions of maximum plastic in support structure

C05

Fig. 9

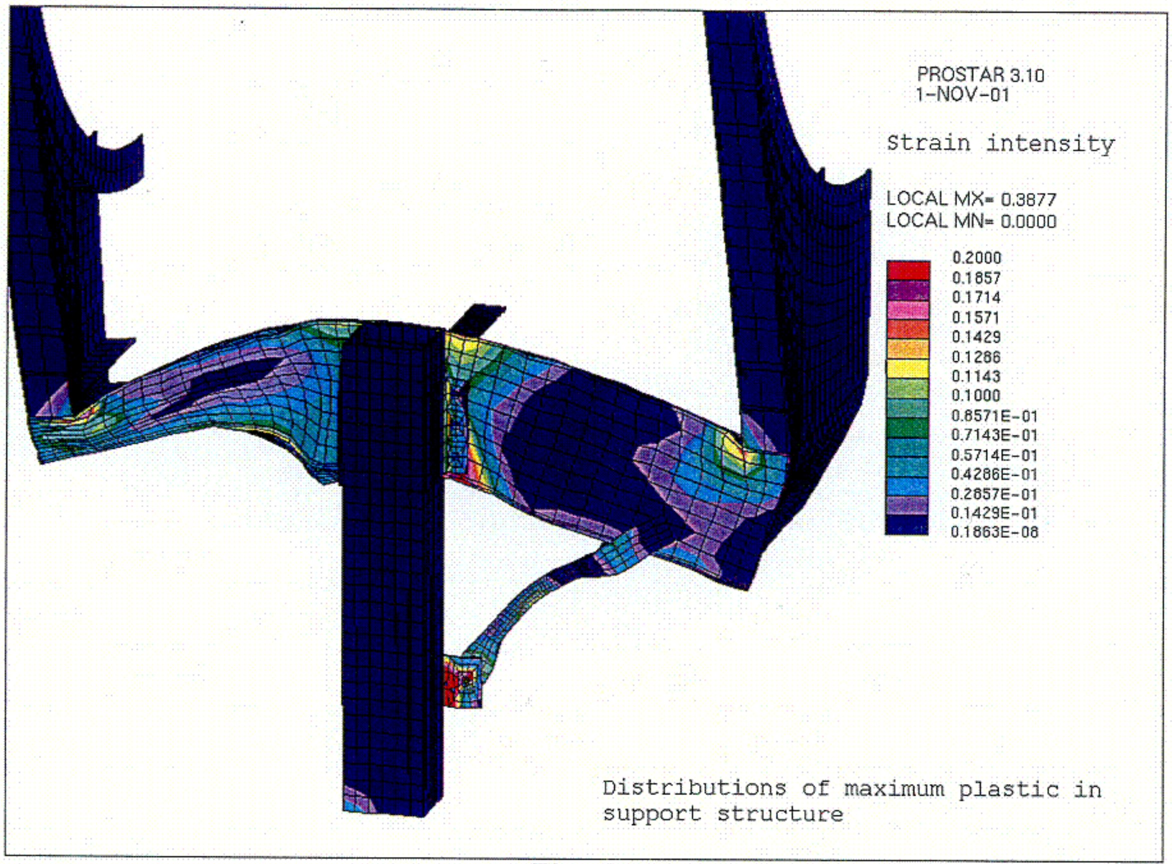
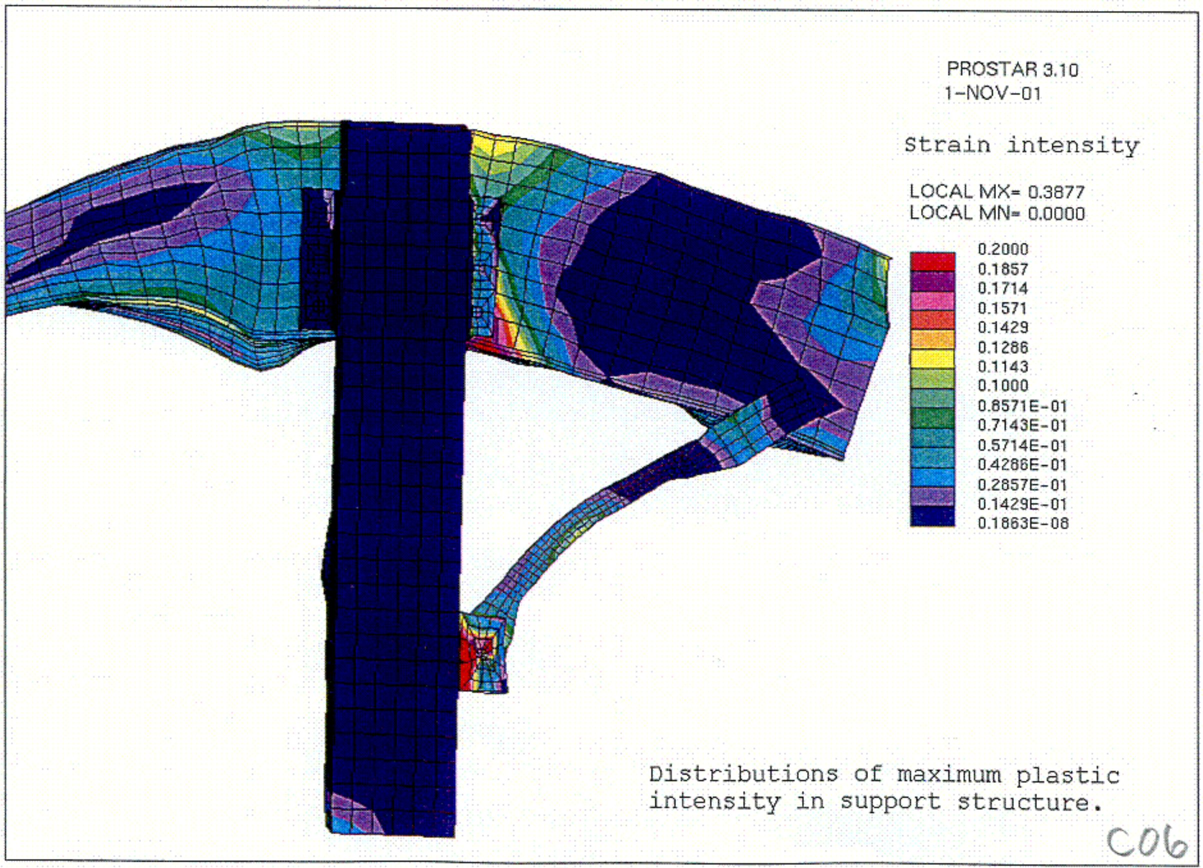


Fig. 10



C06

Fig. 11

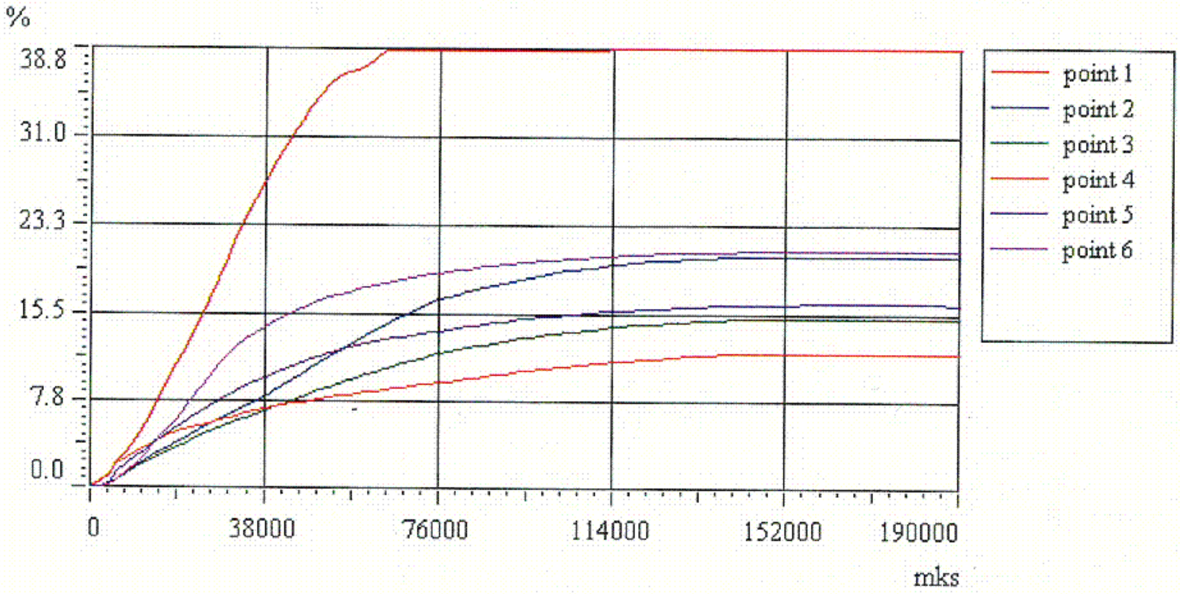


Fig. 12

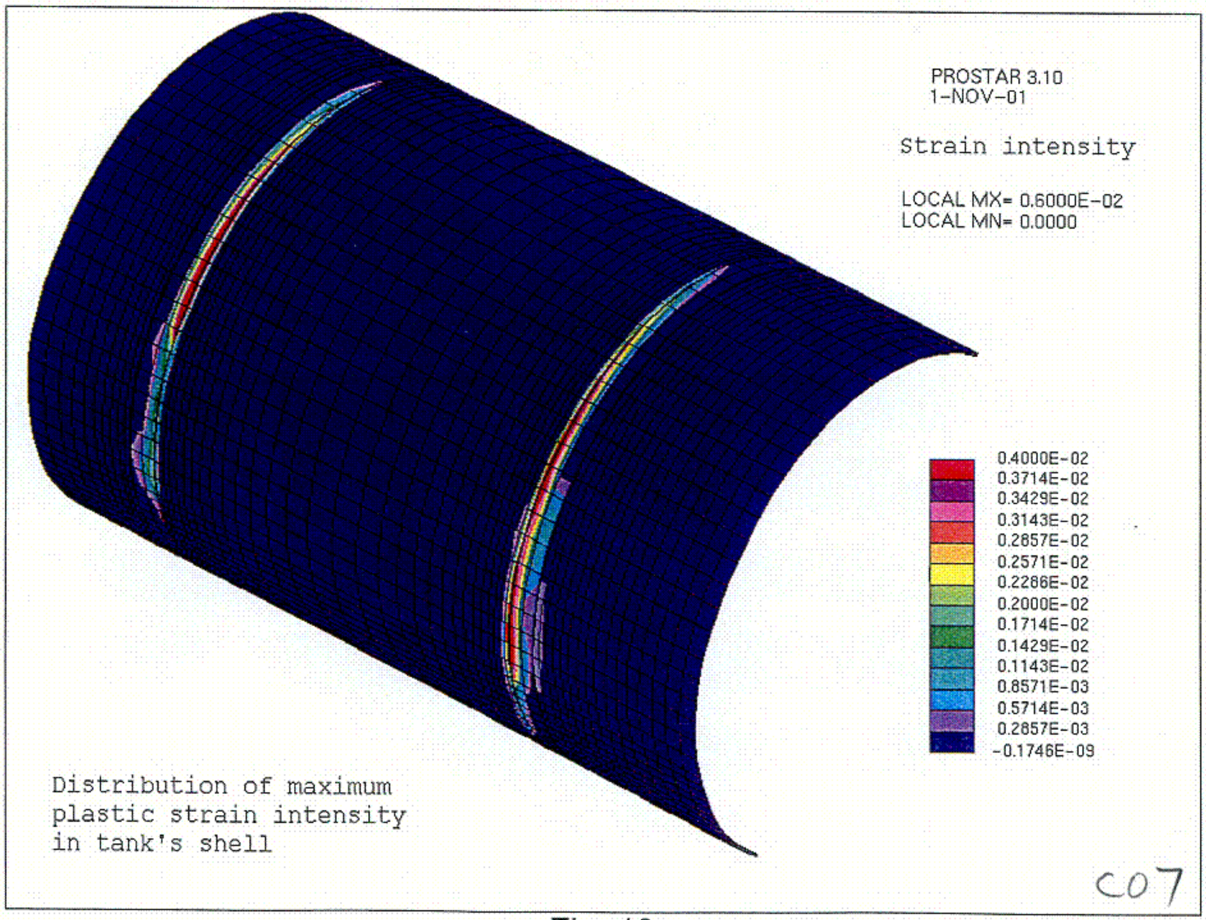


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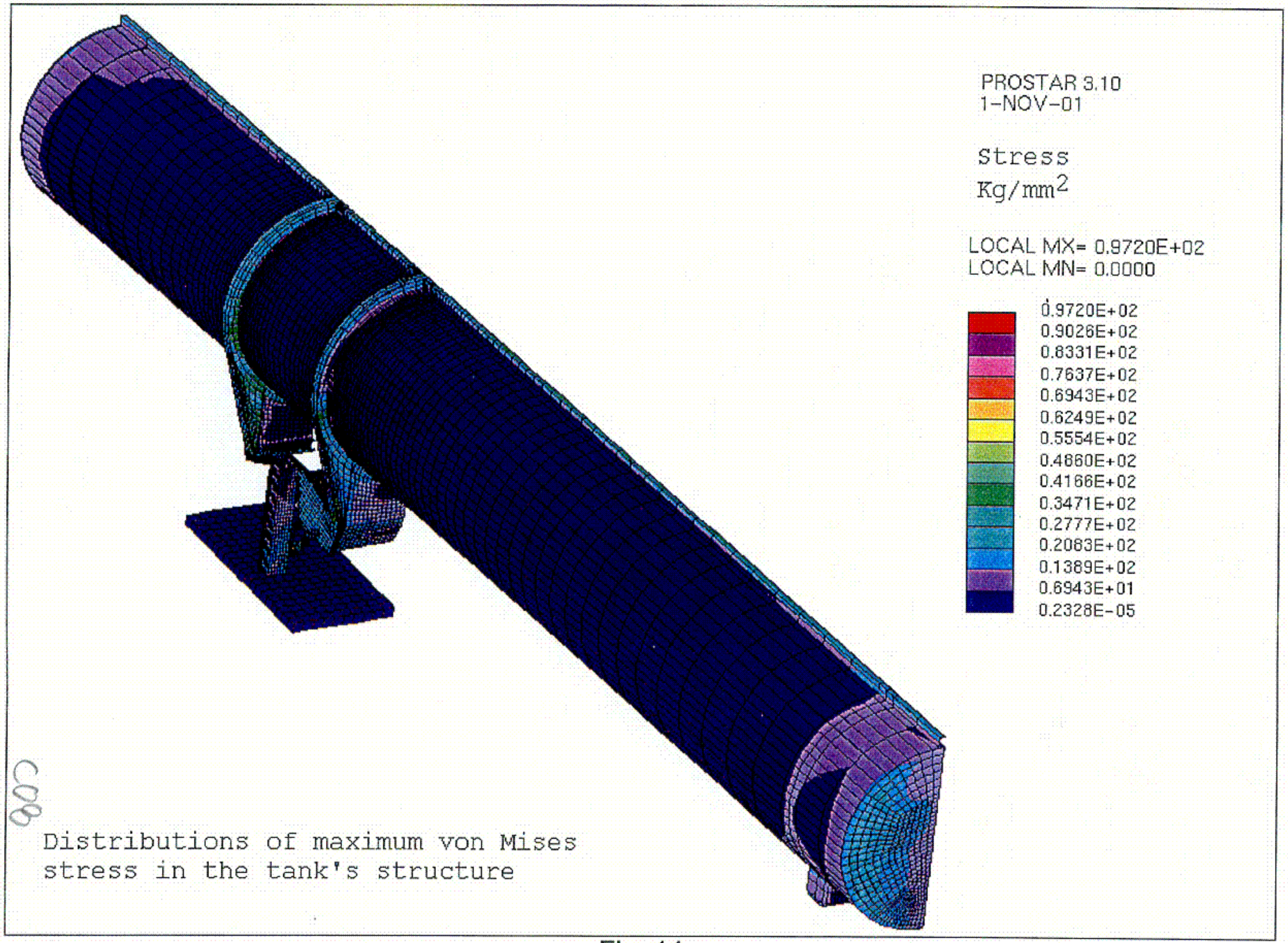
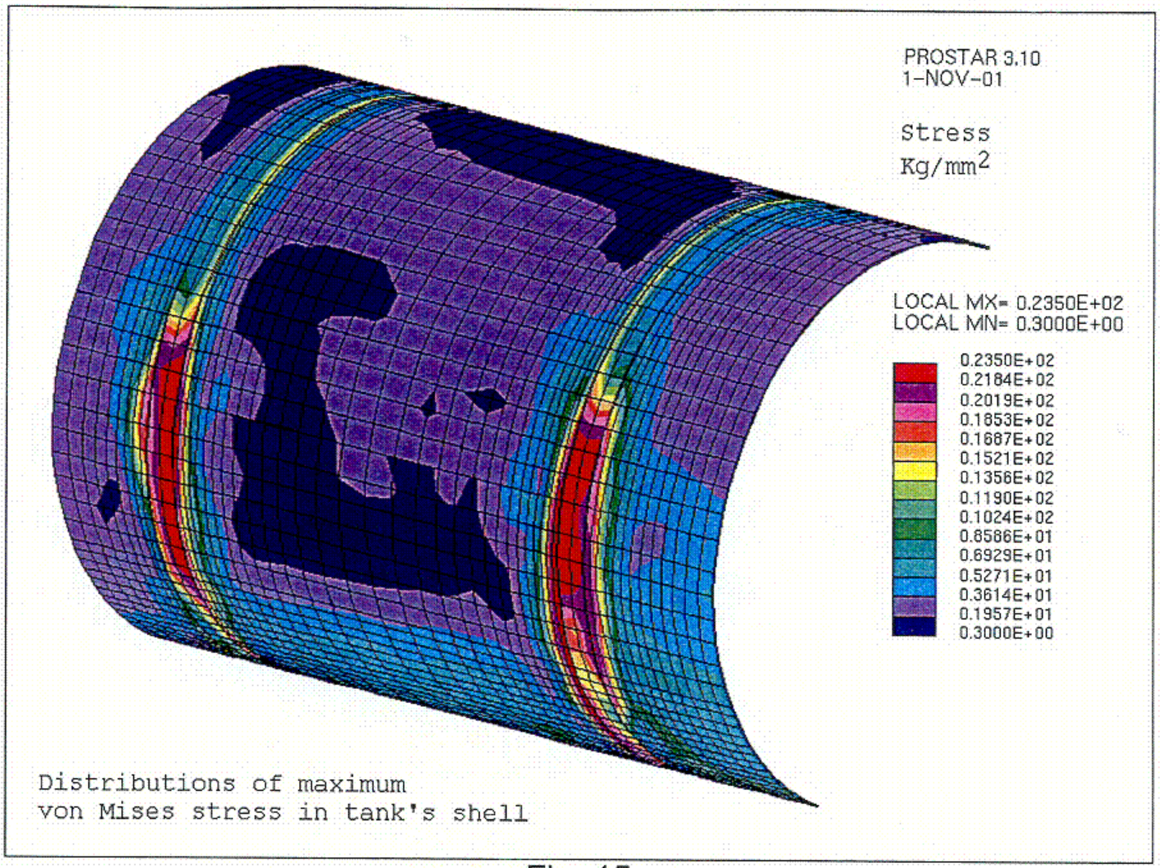
Fig. 13



C08

Distributions of maximum von Mises stress in the tank's structure

Fig. 14



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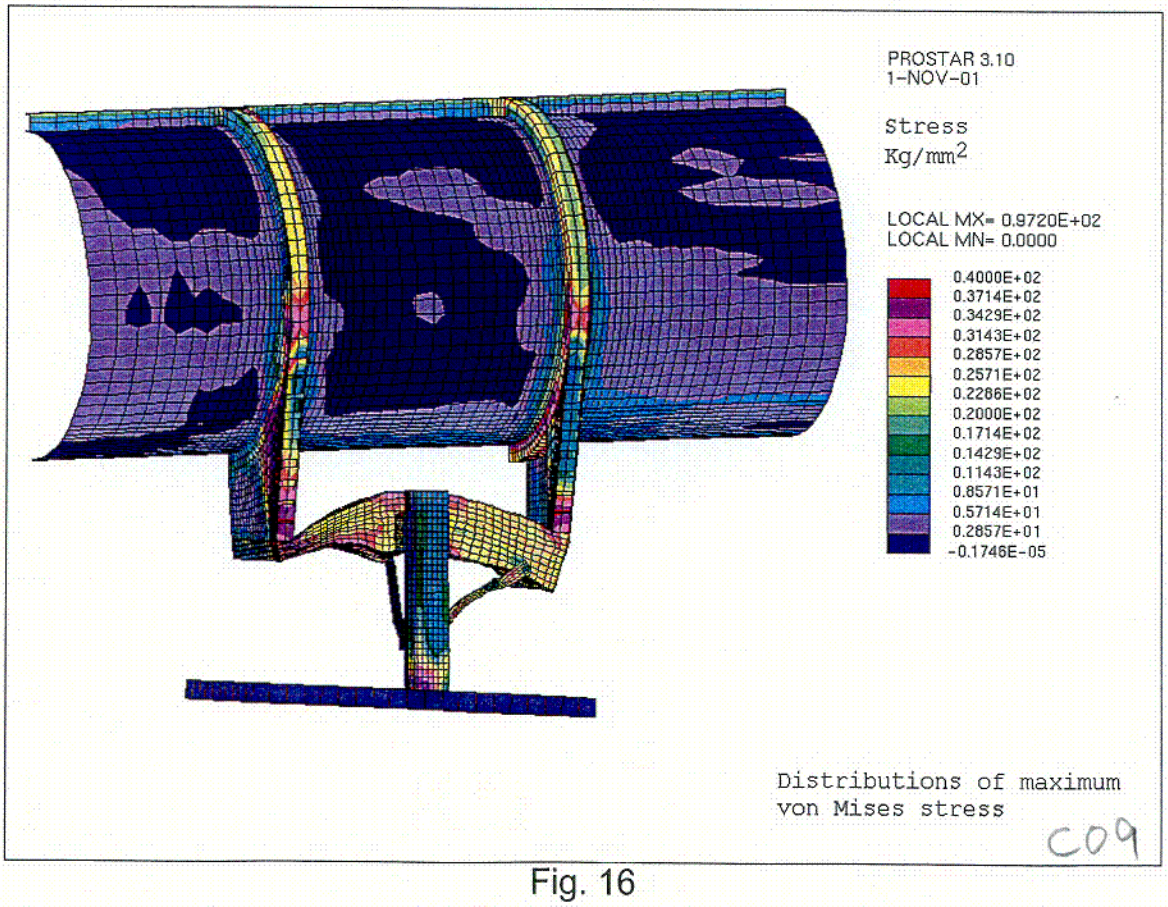
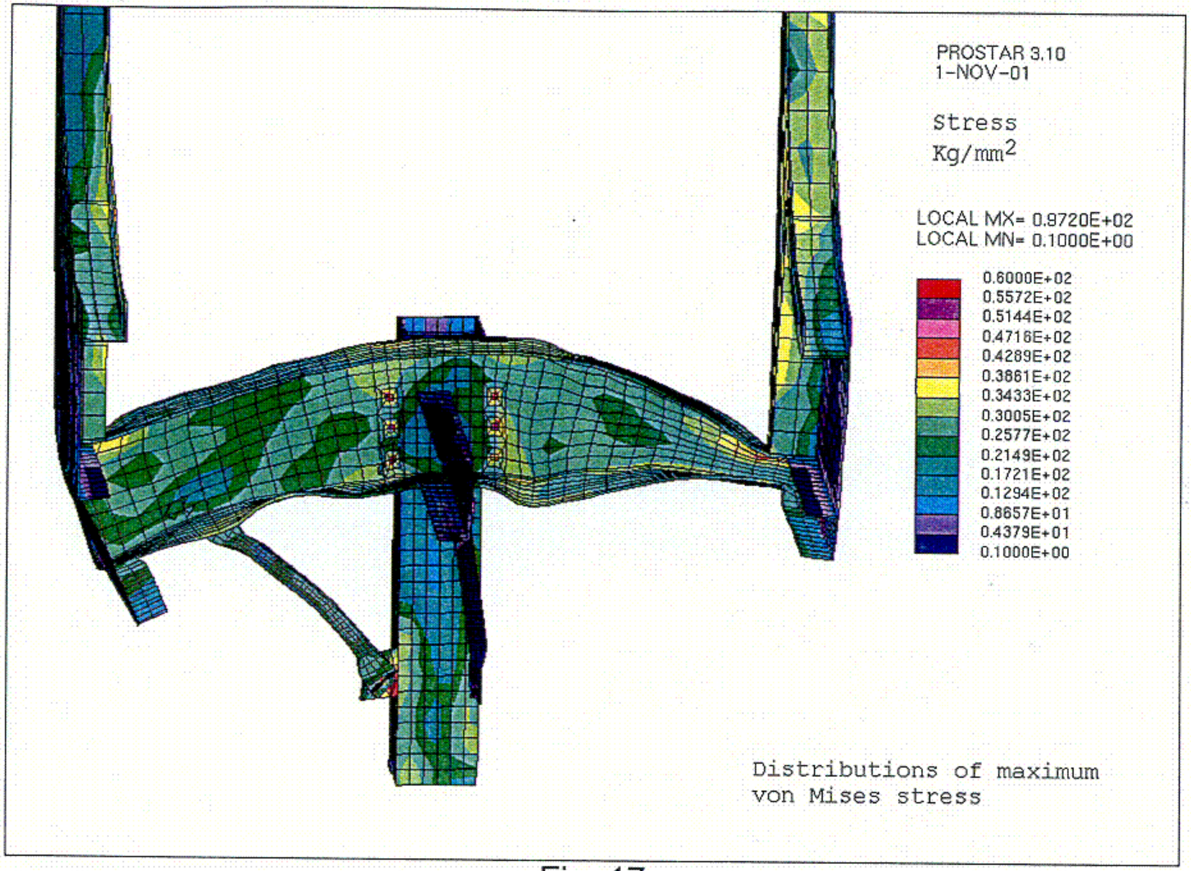


C09

Fig. 16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0.4289E+02
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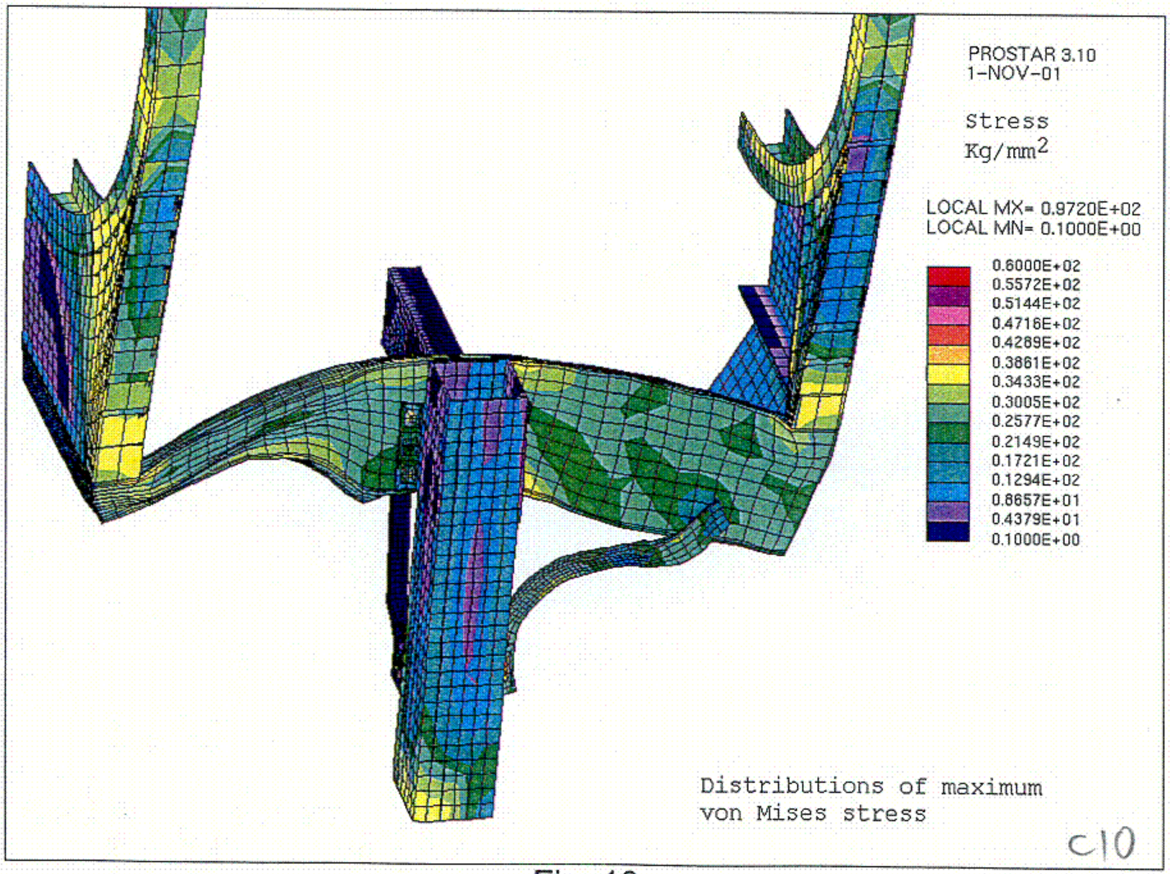
Distributions of maximum
von Mises stress

Fig. 17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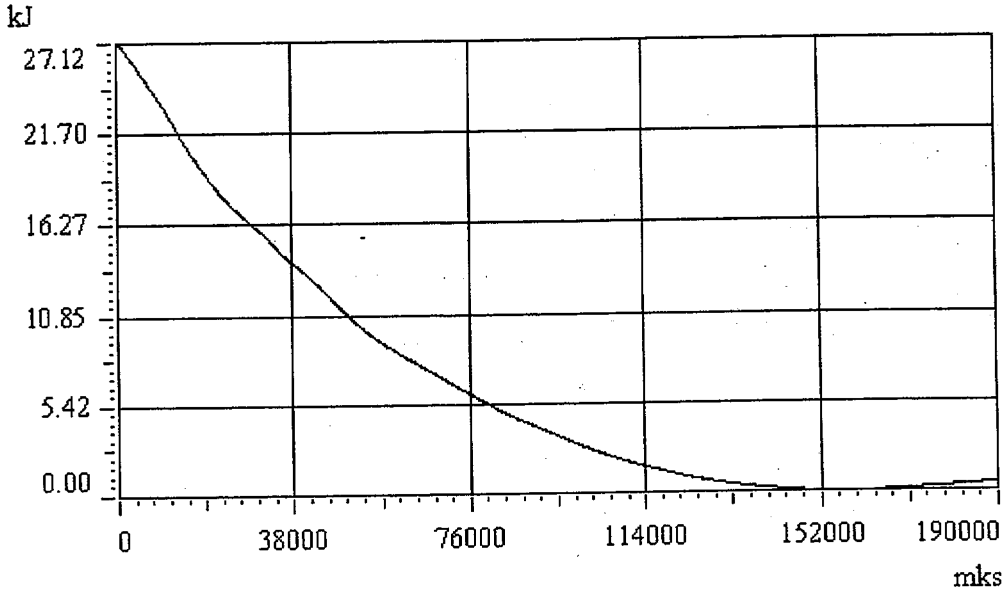
Distributions of maximum
von Mises stress

Fig. 18



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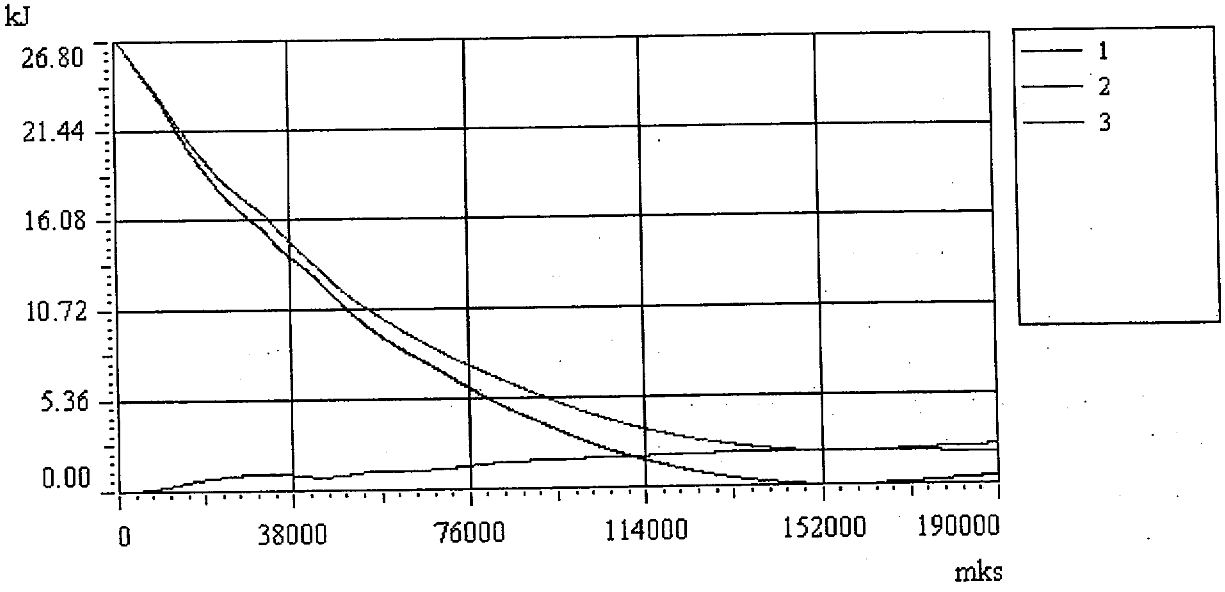
76000
114000
152000
190000
mks

Fig. 19



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190000
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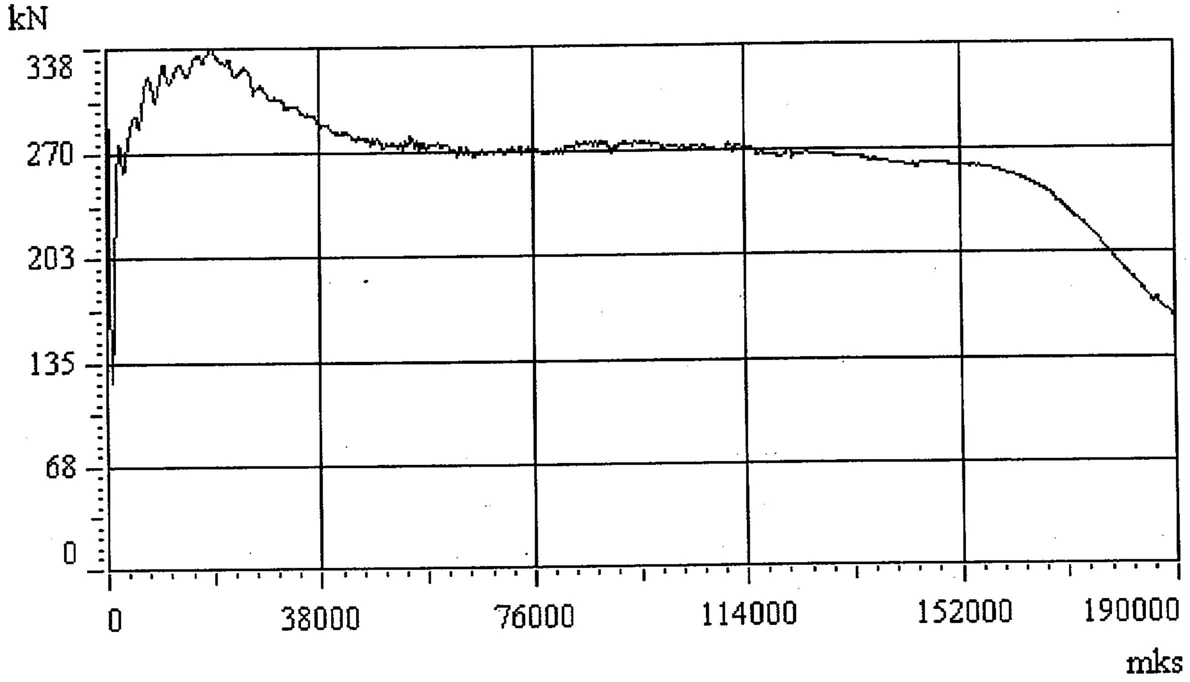
1
2
3

Fig. 20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190000
mks

Fig. 21

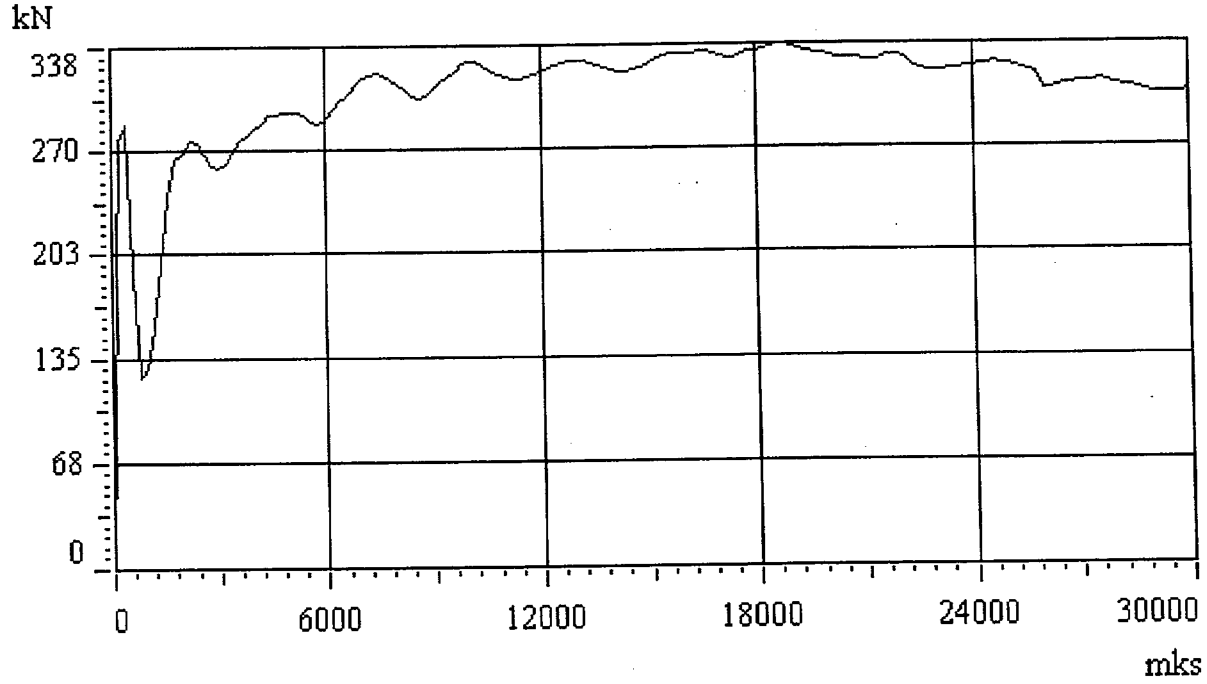
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Fig. 22

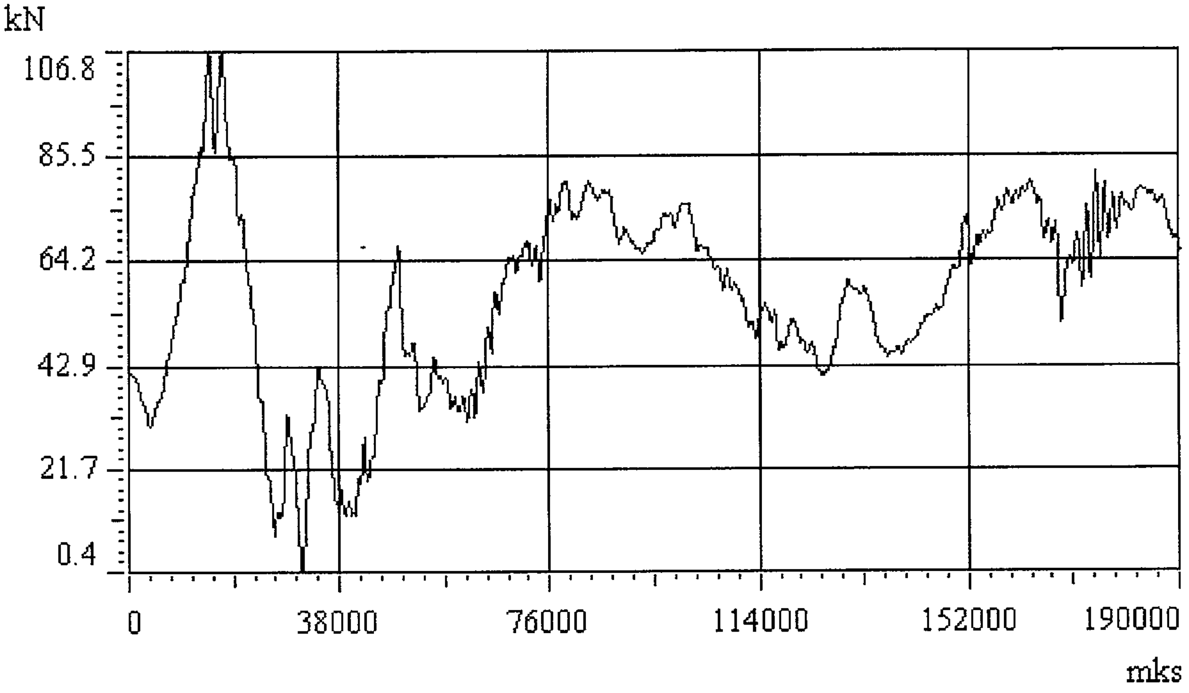

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Fig. 23

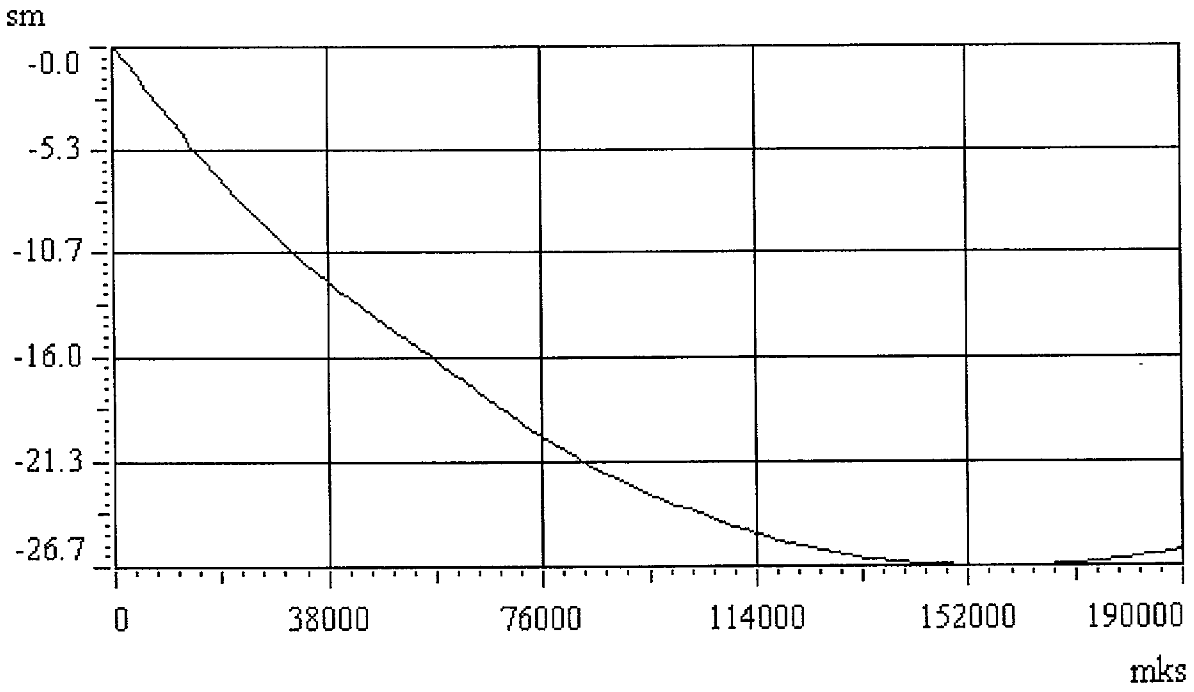

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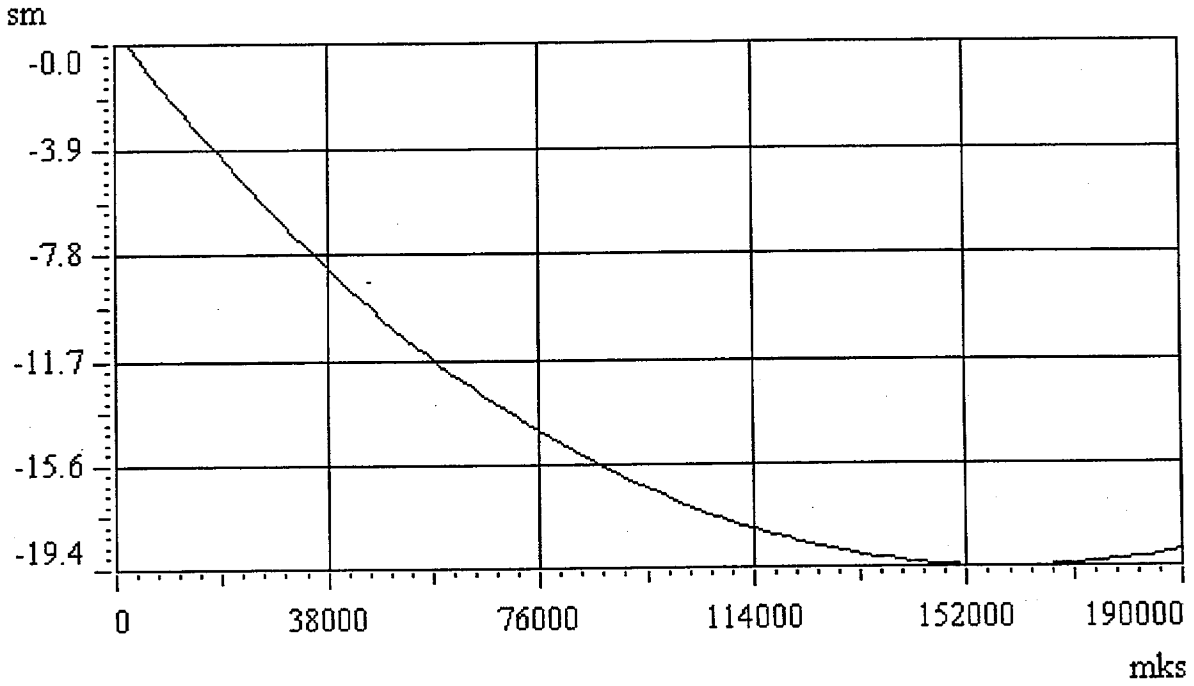
Fig. 24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76000
114000
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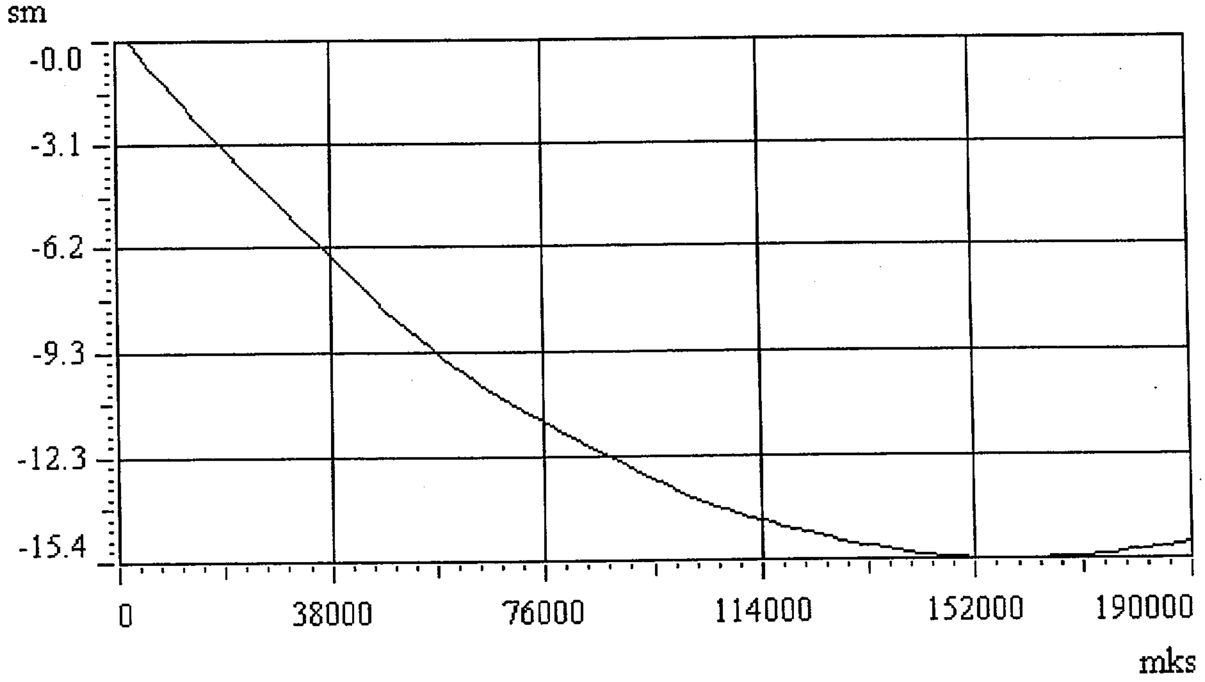
190000
mks

Fig. 25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38000
76000
114000
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190000
mks

Fig. 26

AGREEMENT NO OCC-002/2001

REPORT NO 1-C
MAY 31, 2001

NUCLER FUEL SERVICES 3,700 uswg STANLESS STEEL INSULATED CARGO TANK SEMI-TRAILER [MC311] EFFECT OF ONE FOOT DROP [ON LANDING GEAR]

PREPARED FOR:
ANALYSIS AND DESIGN APPLICATION COMPANY, LTD

PREPARED BY:
SAROV OPEN COMPUTING CENTER

APPROVED BY: ALEXANDER A. RYABOV

Sarov, Russia
2001

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FIGURES

1.0 INTRODUCTION

This report presents numerical simulations of dynamical behavior of truck trailer tank subjected to one foot drop onto a hard (non-yielding) surface. Task Order was prepared by Bill Wheeler and supplied to SOCC by e-mail of April 1. The additional information requested on the Task was sent to SOCC by e-mails of April 9 and April 10. Simulations were curried out by Russian explicit dynamics computer code DINAMIKA-3, which has a Certificate of Russian GOSSTANDART No POCC RU.ME20.H00338, No 0090655 [1].

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- Tank's cylindrical shell stiffening by rings:
- Support frame structure;
- Outer landing gear legs;
- Bolts, which fix the legs to the frame.

There are two features of the problem: a plane of symmetry in the lateral direction and the liquid in the tank. The liquid can be included in the model directly but in this case it takes much more time to get a solution, because the time step of numerical integration becomes very short. The preliminary numerical investigations of the problem have demonstrated that it is quite acceptable only to include the weight of the liquid in analysis.

Preliminary brief analysis of the system has shown that it is twice nonlinear problem (elastic-plastic behavior of materials and large displacements of elements). So it was decided to investigate the system consequently by creating more complicated models. As a result of this approach three computer models of the system were developed and simulated:

Model 1 has two planes of symmetry, simulates dropping without rotation from the height of 305 mm (one foot or 12 inches). The length of the model in lateral direction from the plane of symmetry is 3320 mm (130.5 inches). This length corresponds to the distance from the leg's vertical axis to the forward end of the tank. The objective of this model was to estimate two methods of liquid simulations (direct approach and weight only). It was found out that there is small difference in the results of calculations for two cases. In next models (2 and 3) only the weight of the liquid was used.

Model 2 has only one plane of symmetry in lateral direction, simulates dynamical behavior of whole system during dropping from the height of 305 mm on an un-extended landing gear with rotation of unmoving point at the tandem supporting structures. This model has demonstrated that the most critical region (or place) in the structure is horizontal beam of the support frame near the leg attachment bolts. So it was decided to modify this model 2, by including the bolts.

Model 3 is the modified model 2 by adding the bolts, which connect the outer landing gear legs to the beam of the supporting frame.

Only the two rings that are attached to the landing gear assembly were included in the models, because they determinate the local stresses in tank skin. All other rings affect only on stress level caused by action of liquid inside the tank. The level of this stress can be estimated as follows:

$S_m = K_d \cdot \rho \cdot (2 \cdot R^2/h) = 3 \cdot 1.5 \cdot 10^{-3} \cdot (2 \cdot 64^2/0.48) \approx 0.8$ [Kg/mm^2]

$S_m = 0.8 << S_y = 21.5\ Kg/mm^2$

Where: $K_d = 3$ – dynamical factor;

$\rho \approx 1.5 \cdot 10^{-3}\ Kg/cm^3$ – liquid density;

R = 64 cm – radios of cylindrical shell;

H = 0.48 cm – thickness of the shell.

Because the level of S_m is much less than S_y - Yield Strength, other stiffening rings were included only in the weight of the system.

All models are built of 8-nodes solid finite elements with special high accuracy approximations of strains and stress inside elements.

The weight of the fluid was included in the density of the bottom half of the tank's shell.

The bolts were not preloaded, because the level of preloading was unknown.

There is no any energy being dissipated by damping.

2.0 ANALYSIS OVERVIEW

This section provides a description of problem data (dimensions, weights, etc) and material properties.

2.1 Data of Structure

The construction of a truck trailer tank consists of parts with dimensions (See Figures) as follows:

- Tank's Cylindrical Shell

Diameter	D = 1273 mm (50.5 in);
Length	L = 10693 mm (421 in);
Thickness	t_o = 4.76 mm (3/16 in);

- Stiffening Ring (Figure A)

Width	a_1 = 57.15 mm (2.25 in);
Height	a_2 = 50.40 mm (2.00 in);
Thickness	t_1 = 3.57 mm (0.1406 in);

- Support Frame Structure (Figure A)

Width	H = 965 mm (38 in);
Length	L = 1200 mm (48 in);
Thickness of inner cross member	t_2 = 4.76 mm (3/16in);
Thickness of beam wall	t_3 = 4.76 mm (3/16 in);
Thickness of walls	t_5 = 4.76 mm (3/16 in);
Width of side cradles	$c_1 = c_2$ = 57 mm (2.25 in);
Width of shelf	d_1 = 57 mm (2.25 in);

Outer Landing Leg (Figure A)

Tube cross section sizes	$b_1 \times b_2$ = 127 mm x 127 mm (5in x 5in)
Thickness of the outer tube	t_4 = 4.57 mm (0.180 in)
Length of the outer tube	b_3 = 762 mm (30 in)

- Bolts (six at each leg)

Diameter d = 12.7 mm (½ in)

The weight parameters of system components are as follows:

The total weight of loaded trailer	G_0 = 29060 Kg (64,000 lbs)
The weight without tandem	G_1 = 26830 Kg (59,100 lbs)
The density of the liquid inside	p = 1497.8 Kg/m^3 (12.5 lb/gal)
The density of steel structures	r = 7800 Kg/ m^3

2.2 Material Properties

For all materials in this analysis the elastic-plastic model of deformation with linear isotropic hardening was used based on following data:

- Tank's Cylindrical Shell and Stiffening Rings are made of stainless steel **316SS**:

Young's Modulus	20,000 Kg/mm2
Poisson's Ratio	0.3
Yield Strength	21.1 Kg/mm2
Ultimate Strength	52.76 Kg/mm2
Elongation	40%

- Support Frame Structure and Landing Gear are made of steel **A36**:

Young's Modulus	20,000 Kg/mm2
Poisson's Ratio	0.3
Yield Strength	31.68 Kg/mm2
Ultimate Strength	42.24 Kg/mm2
Elongation	21.5%

- Bolts (six at each leg) are made of **carbon steel**:

Young's Modulus	20,000 Kg/mm2
Poisson's Ratio	0.3
Yield Strength	35.20 Kg/mm2

3.0 RESULTS AND DISCUSSION

This section provides the results of numerical calculations for developed models 1-3.

3.1 Model 1

Finite Element Model and its part are presented in Figures 1 and 2. Results of numerical simulations obtained by this model are demonstrated in Figures 3-9.

Fig. 3, 4 – Distributions of maximum plastic strain intensity;

Fig. 5, 6 – Distributions of maximum von Mises stress (Kg/mm^2);

Fig. 7- Kinetic energy of the model vs. time;

Fig. 8 – Vertical force on leg in the model vs. time;

Fig. 9 – Vertical displacement of the horizontal beam's end vs. time

Results of calculations show that the duration of the impact is about 70 – 75 ms. In this period of time the kinetic energy of the system drops down from K =9.93 KJ to zero and vertical reaction force of one leg reached its maximum value of F_{max} = 160 x 2 = 320 KN at about time t = 14 ms and then slowly falls down. It appears to be a quasi-static process. The most stressed region of the structure is horizontal beam of the frame attached to the leg (Fig. 3, 4). The maximum intensity of plastic strains at this place is about 12-13.8 %. The maximum intensity of plastic strains in tank's skin is no more than 0.9%. Maximum vertical displacement of the end of the horizontal beam is 92.6 mm.

3.2 Model 2

Finite Element Model 2 is presented in Figures 10 and 11. In this model the weight of tank equipment and insulation was not included. The weight of the model was 21400 Kg (in accordance with e-mail of April 10 the weight of the system is 26830 Kg). Numerical results of are shown in Figures 12-19.

Fig. 12, 13,14 – Distributions of maximum plastic strain intensity;

Fig. 15, 16 – Distributions of maximum von Mises stress (Kg/mm^2);

Fig. 17- Kinetic energy of the model vs. time;

Fig. 18 – Vertical force on leg in the model vs. time;

Fig. 19 – Vertical displacement of the horizontal beam's end vs. time

For this model the duration of the de-acceleration process is about 75 – 80 ms. During this time the kinetic energy of the system drops down from K =21.31 KJ to zero. The reaction force of one leg reaches a maximum of F_{max} = 299 KN at about time t = 4 ms and then after some hesitations slowly falls down. This case is also a quasi-static process. The maximum intensity of plastic strains in the beam is about 12-13.7 %. The maximum intensity of plastic strains in tank's skin is no more than 1.2%. The Maximum vertical displacement of the end of the horizontal beam is 134.1 mm. The Static vertical reaction force of one leg is F_{st} = 75.2 KN, and the dynamics loading factor in this accident is about $K_d = F_{max} / F_{st} = 299/75.2 = 4.0$.

3.3 Model 3

Finite Element Model 3 is shown Figures 20 and 21. The Model 3 is modified Model 2, and includes the weight of tank equipment and insulation. The weight of Model 3 was 26830 Kg). Model 3 also includes the bolts and additional elements at the bolt installation location on the leg. The contact interaction between leg and horizontal beam was also included in the Model 3. Results of are shown in Figures 22-39.

Fig. 22 ...26 – Distributions of maximum plastic strain intensity;

Fig. 27 – Distribution of maximum plastic strain intensity in stiffening rings;

Fig. 28 - Distribution of maximum plastic strain intensity in tank's shell;

Fig. 29,30,31 – Distributions of maximum von Mises stress (Kg/mm^2);

Fig. 32 - Distributions of maximum von Mises stress in rings (Kg/mm^2);

Fig. 33 - Distributions of maximum von Mises stress in tank's shell (Kg/mm^2);

Fig. 34 - Kinetic energy of the model vs. time;

Fig. 35 – Energies of the tank vs. time; [(1) – strain energy, (2) – kinetic energy, (3) –sum of (1) and (2)];

Fig. 36 – Vertical force on leg in the model vs. time;

Fig. 37 - Vertical force on leg in the model vs. time (in the time range from 0 to 20ms) ;

Fig. 38 – Vertical displacement of the tank's end vs. time;

Fig. 39 - Vertical force time history of model at the tandem support point.

In this case the period of active impact is about 260 ms. In this period the kinetic energy of the system (K= 27.01 KJ) is transformed into deformation energy. Analysis of Fig.35 shows that tank absorbs only 5 %, but support structure absorbs 95% of the kinetic energy of the system. The reaction force in the leg reaches its maximum $F_{max} = 250 - 260$ KN and then slowly falls down to the end of active process (t =250-260 ms) and then drops fast when the structure rebounds. In this model, the nominal static force of one leg is $F_{st} = 93.4$ KN, so the maximum dynamic loading factor is about $K_d = F_{max} / F_{st} = 255/93.4 = 2.7$. The oscillations of reaction force on the leg during the first 2...3 ms (Figure 37) can be explained as follows:

0...100mks - load raises from 0 to 200kN, horizontal beam transmits inertia force from the tank by elastic and small plastic deformations, leg is elastic compressed;

100...300mks - load increases up to its max of 260kN and in horizontal beam plastic deformations cover whole cross-section and it drops the bending stiffness of the beam and decreases its ability to transmit load from the tank to the leg, in this interval leg is else deformed elastically;

300...800mks - because of transfer collapse of the beam caused by plastic deformation, the leg elastically relaxes itself as compressed spring and it causes the decreasing of compression contact force in the leg (it confirms by analysis of displacements in different points of the leg);

800..1700mks- motion of the tank continues and the leg is again compressed. Analysis of leg's displacements shows that during whole time interval 0...2 ms there is no gap between leg and rigid surface.

From Fig. 39 one can see that the trailer does not "bounce" off of the ground at the tandem location. For Model 3 we can observe the same quasi-static behavior of the system. The maximum vertical displacement of the end of the tank reaches a maximum value of 425mm. There are some buckling phenomena in the deformed frame structure. The maximum intensity of plastic strains in the bottom part of the beam is about 32.3 %. But they are mainly compressive components and can't cause the rupture of the wall. In the top part of the beam maximum tensile deformations reach the level of 13-14% and is less than elongation of the steel A36 (21.5%). The maximum intensity of plastic strain in most loaded bolt is about 8-9 %. The maximum plastic strain in the shell of the tank is no more than 0.9 %.

4.0 CONCLUSIONS AND RECOMMENDATIONS

1. In the accident of the tank dropping onto its landing gear legs from a height of 305 mm (12 in) to a rigid surface, the active impact continues for about 260 ms. During the period of time vertical reaction force in outer landing gear leg increases up to the level of 250-260 KN (dynamic loading factor K_d =2.7). It causes large deformations in horizontal beam of support frame structure and kinetic energy of the system absorbs mostly by this element.
2. Maximum levels of plastic deformations intensity in element of the structure are as follows:

- Shell of the tank:	0.9% < 40% (316SS);
- Horizontal beam of support frame structure:	32.3% (compressive)
	14% (tensile) < 21.5% (Steel A36):
- Maximum loaded bolt:	8 - 9% (Unknown Carbon Steel);
- Outer landing gear leg:	5 – 6% (local) <21.5% (Steel A36).

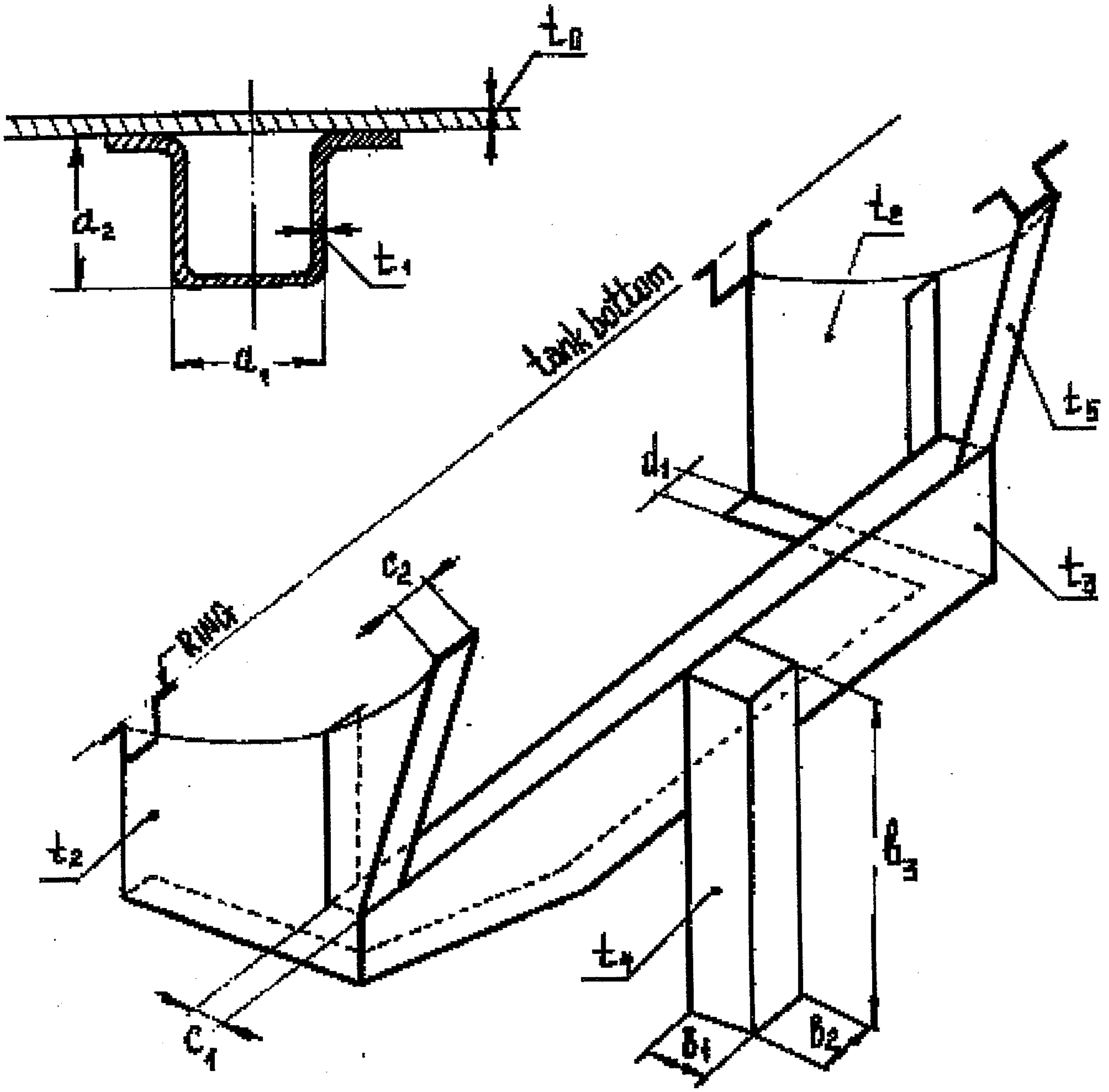
The maximum levels of calculated plastic deformations do not exceed the elongations of the correspondent steels. However, they have high levels and may cause the appearance of local cracks in the elements of the structure, which would increase the danger of the accident. Taking this into account, it is recommended that the resistance of the horizontal beams of the support frame structure and leg bolts should be increased by welding some additional elements to the existing structures.

6.0 REFERENCES

1. DINAMIKA-3, Пакет прикладных программ ДИНАМИКА-3, Сертификат Госстандарта России № РОСС RU.ME20.HOO338, NO 0090655, 2000г.

April 3 2001

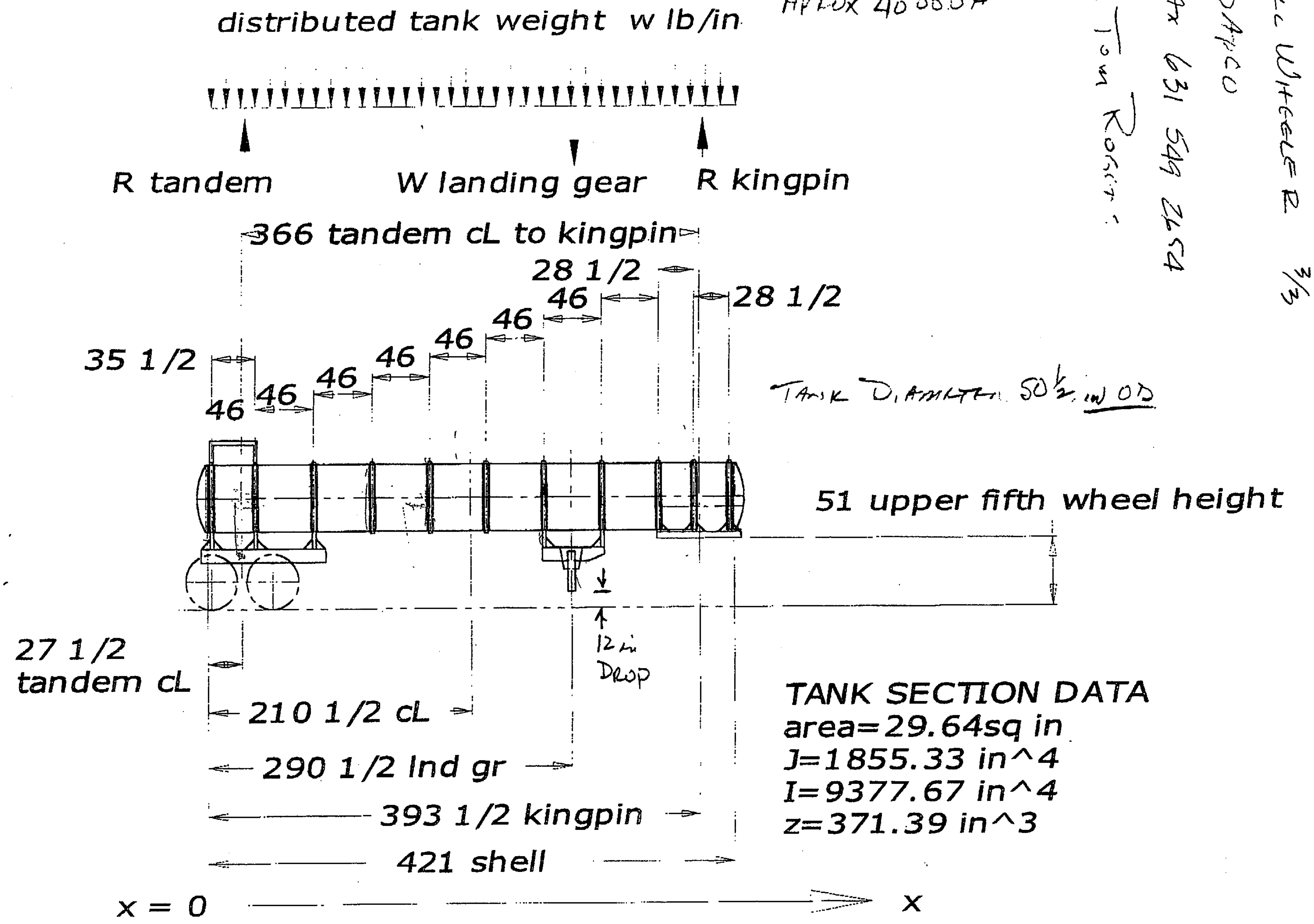
Task 1. Dropping of the tank with radioactive waste
Support structure



t_0 – thickness of tank shell
t_1 – thickness of ring walls
t_2 – thickness of inner cross member
t_3 - thickness of beam wall
t_4 - thickness of bar walls
t_5 - thickness of walls

Bill Wheeler 3/3
IDAPCO
Fax 631 549 2654
From Tom Rogers?

DISTRIBUTED WT APPROX 60 30# 26 STATIC WT
APPROX 40000#



TANK SECTION DATA
area=29.64sq in
J=1855.33 in^4
I=9377.67 in^4
z=371.39 in^3

TANK REPRESENTED AS A SIMPLY SUPPORTED BEAM

HAT RING SECTION
continuously welded to shell

TYPE 316 STAINLESS STEEL DATA
ASME SECTION II 1998 EDITION

yield	30,000psi [206.85 N/mm^2]
ultimate	75,000psi [517.125N/mm'A2]
elongation	40%

NOTE:
hat section is connected to shell
ONLY at fillet welds at edge of section
7 1/2 in (190.5 mm) shell band width is considered
to be effectively combined with hat section as
part of circumferential shell reinforcement, (=40t)

FIGURE WITHHELD UNDER 10 CFR 2.390

shell 3/16in
4.763mm
type 316 ss

SECTION DATA
total area=2.256sqin
Ix=1.5571n^4
cl=1.573in
c2=0.61Sin

PLASTIC SECTION DATA
(top) areal=0.770sqin
yl=1.107in
(bottom) area2=1.7867sqin
y2= 0.477in

w

T

W

FIGURE WITHHELD UNDER 10 CFR 2.390

ok - Leonard A Abram NFS 11/15/65

M.S.	PURCH.	C.H.	INSL.	S.O.	ENG.	OMP.	FRP.	CUST.
✓		✓	✓	✓	✓			

METAL PRODUCTS
EHAUF TRAILER CO.
N, PA. PLANT

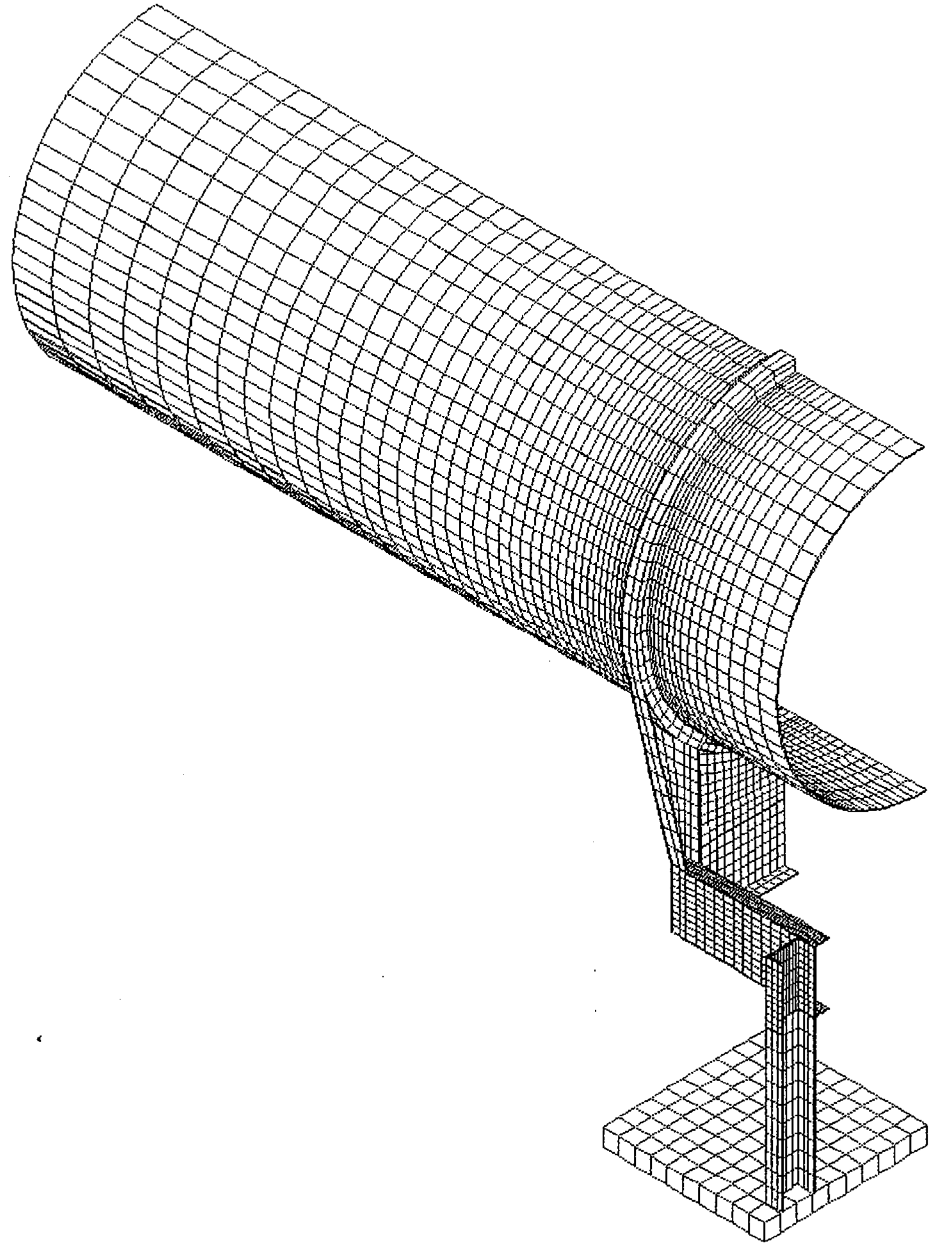
EL SERVICE INC."
E ASSEMBLY

MATERIAL	DTR.	CKR.	ENG.	MODELS	PART NO.
C.S. ☐ S.S. ☐ A.L. ☐ C.C.S. ☐	F A G		J T A.	UNF 223201-02	UNF 2232
SCALE 3/4" = 1'-0"	9 22 65				REV. A SHEET 1 OF

OUTER LANDING TUBE

SECTION AA

FIGURE WITHHELD UNDER 10 CFR 2.390



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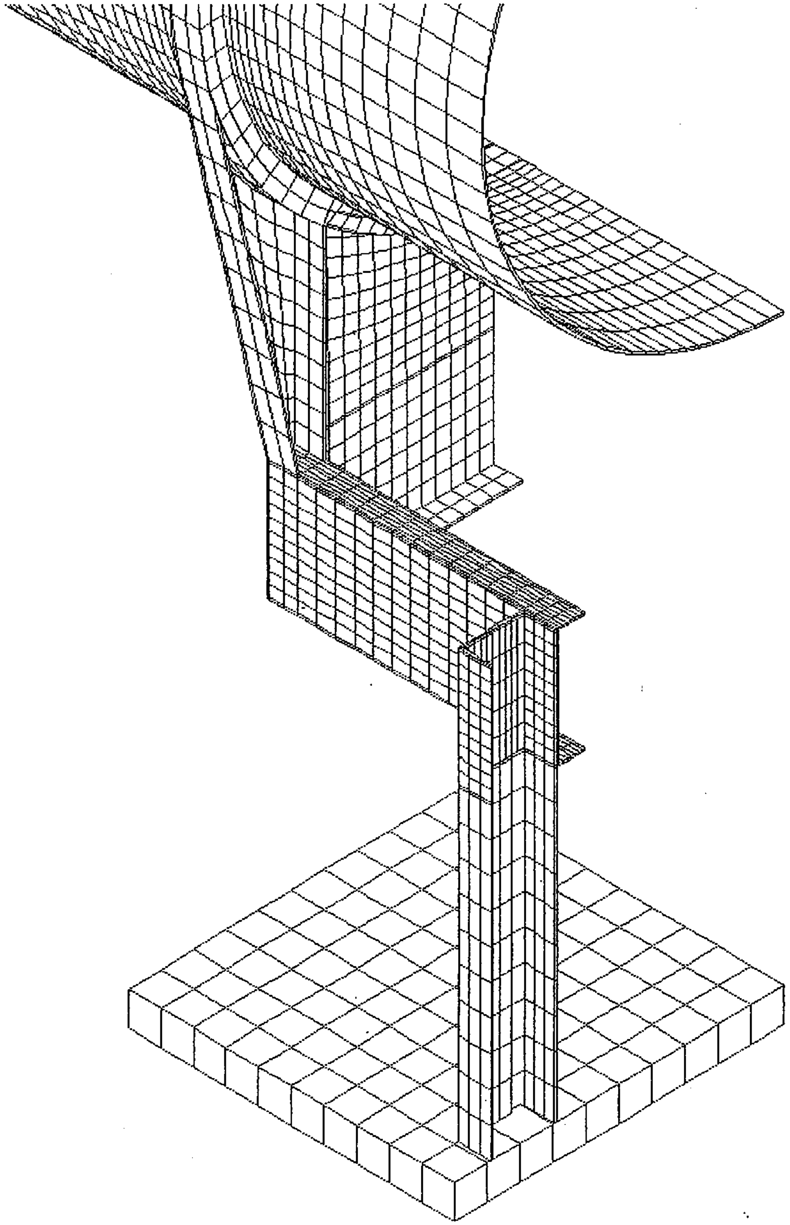

Fig. 2

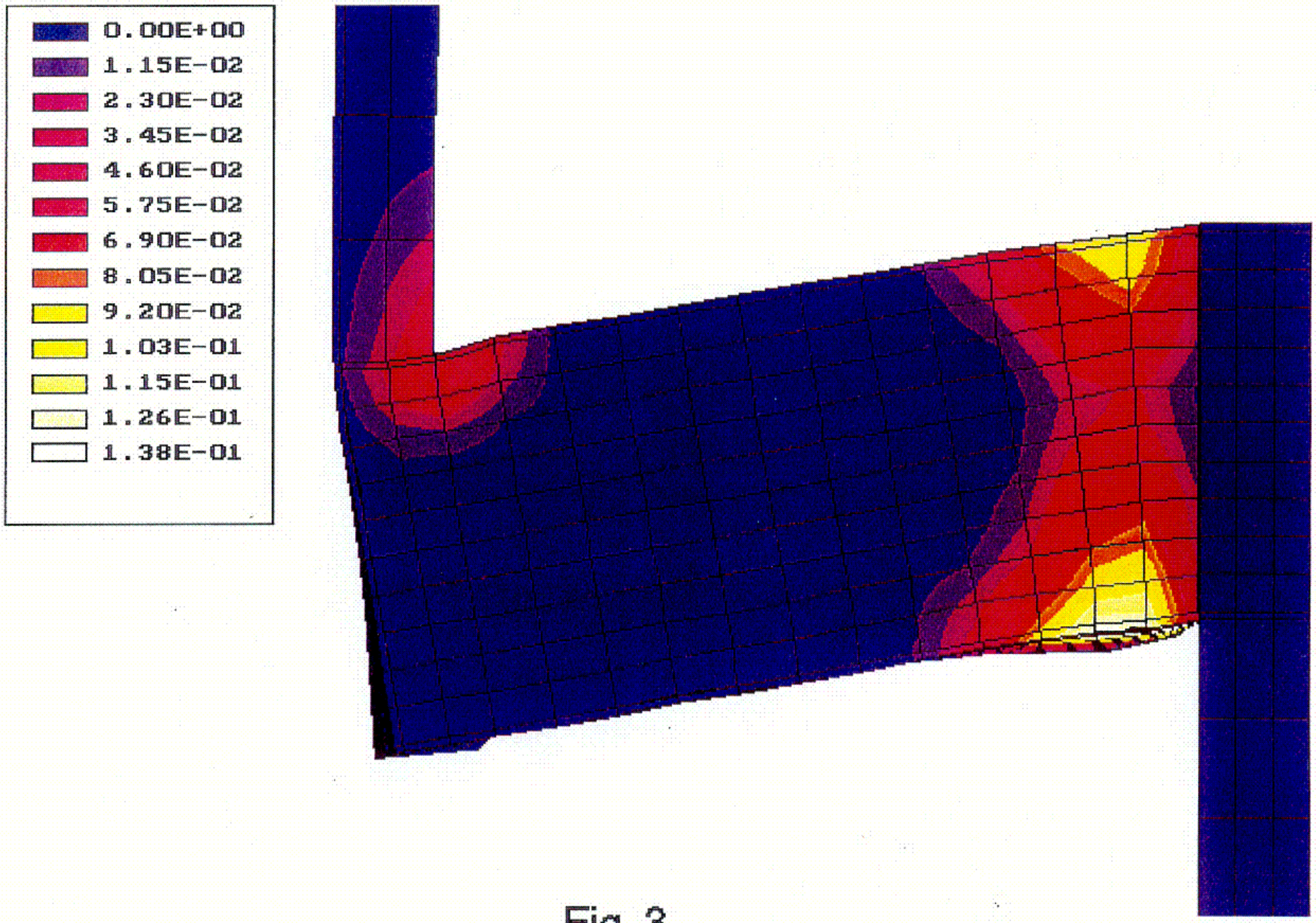
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Fig. 3

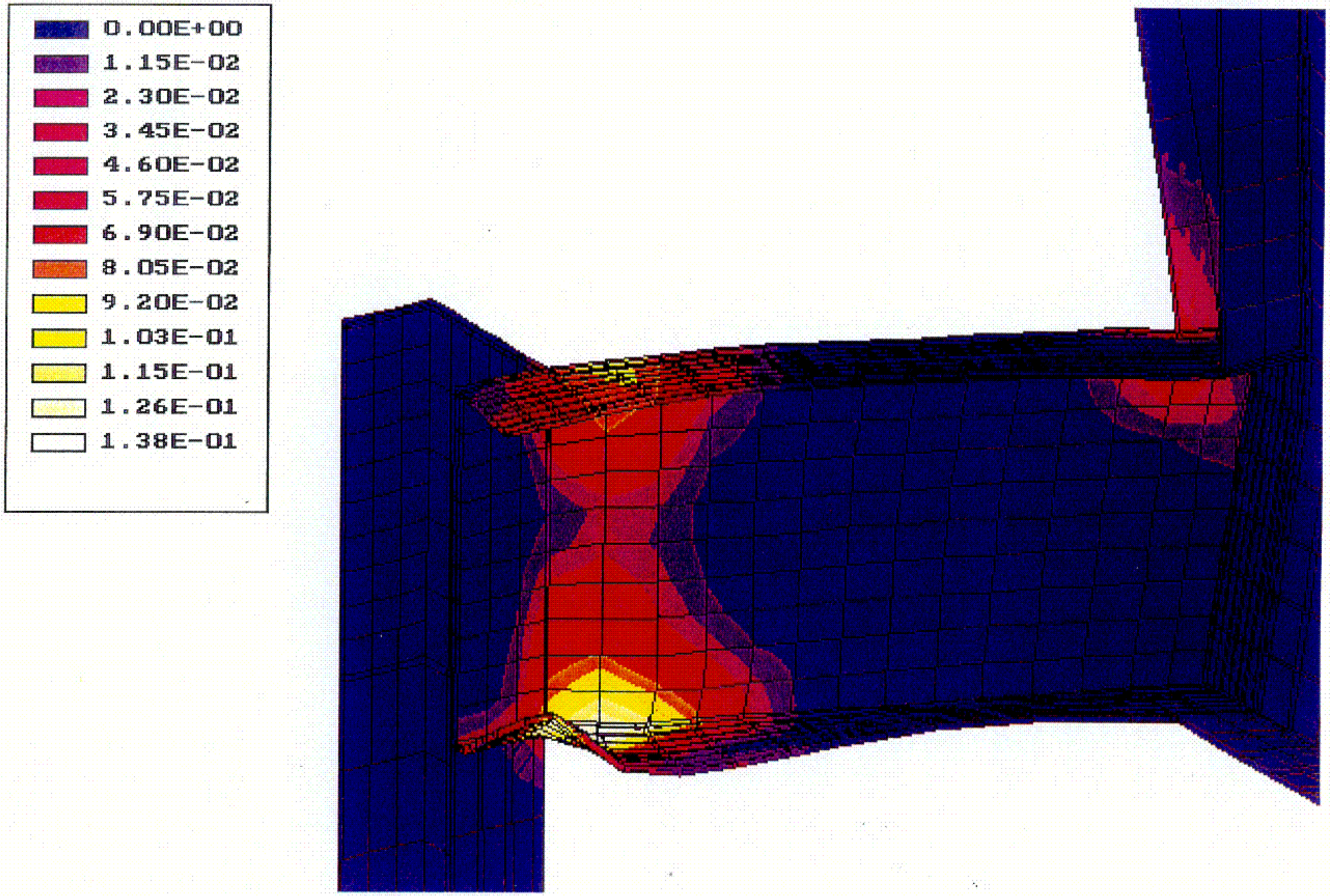
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Fig. 4

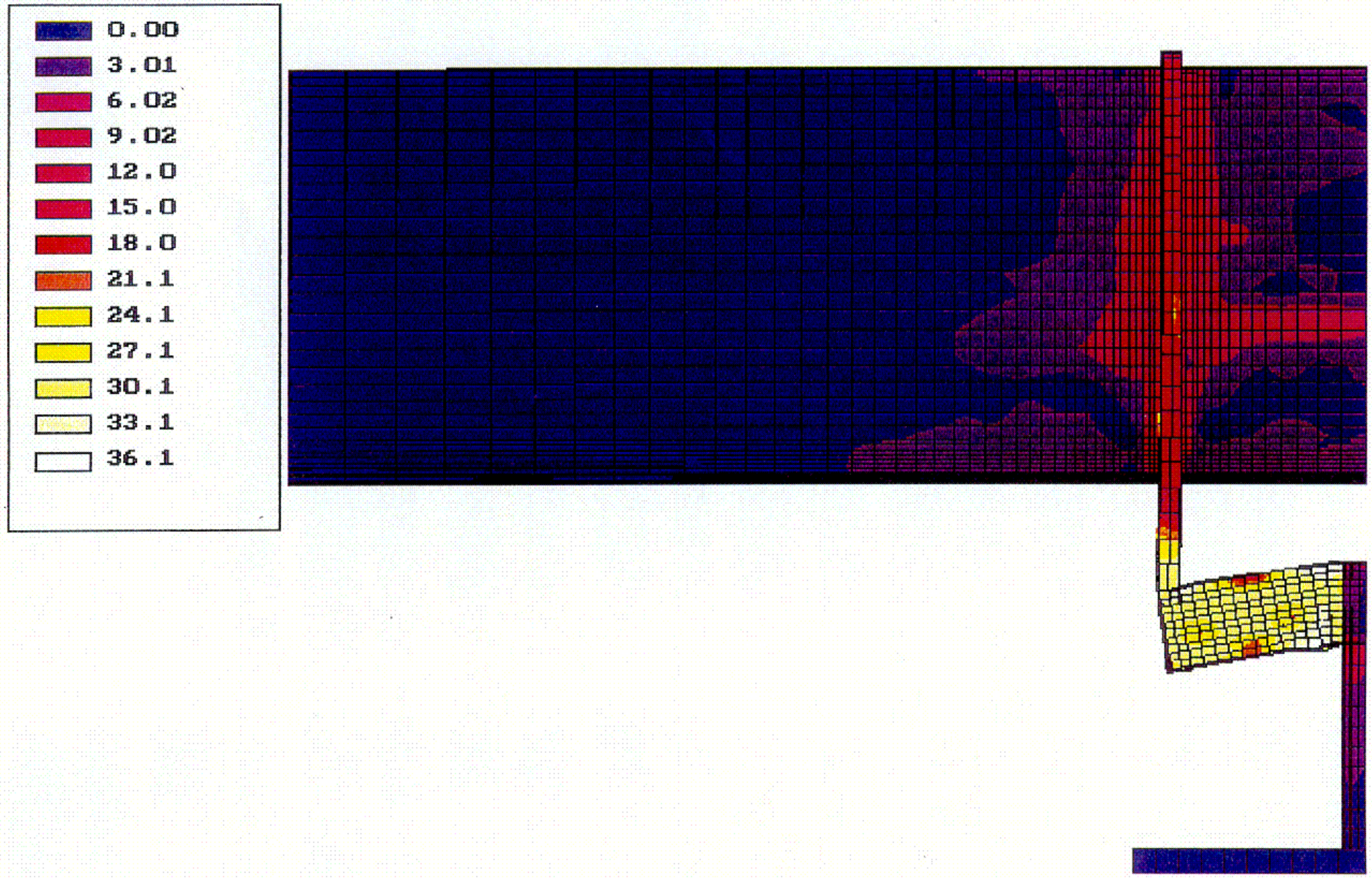


Fig. 5

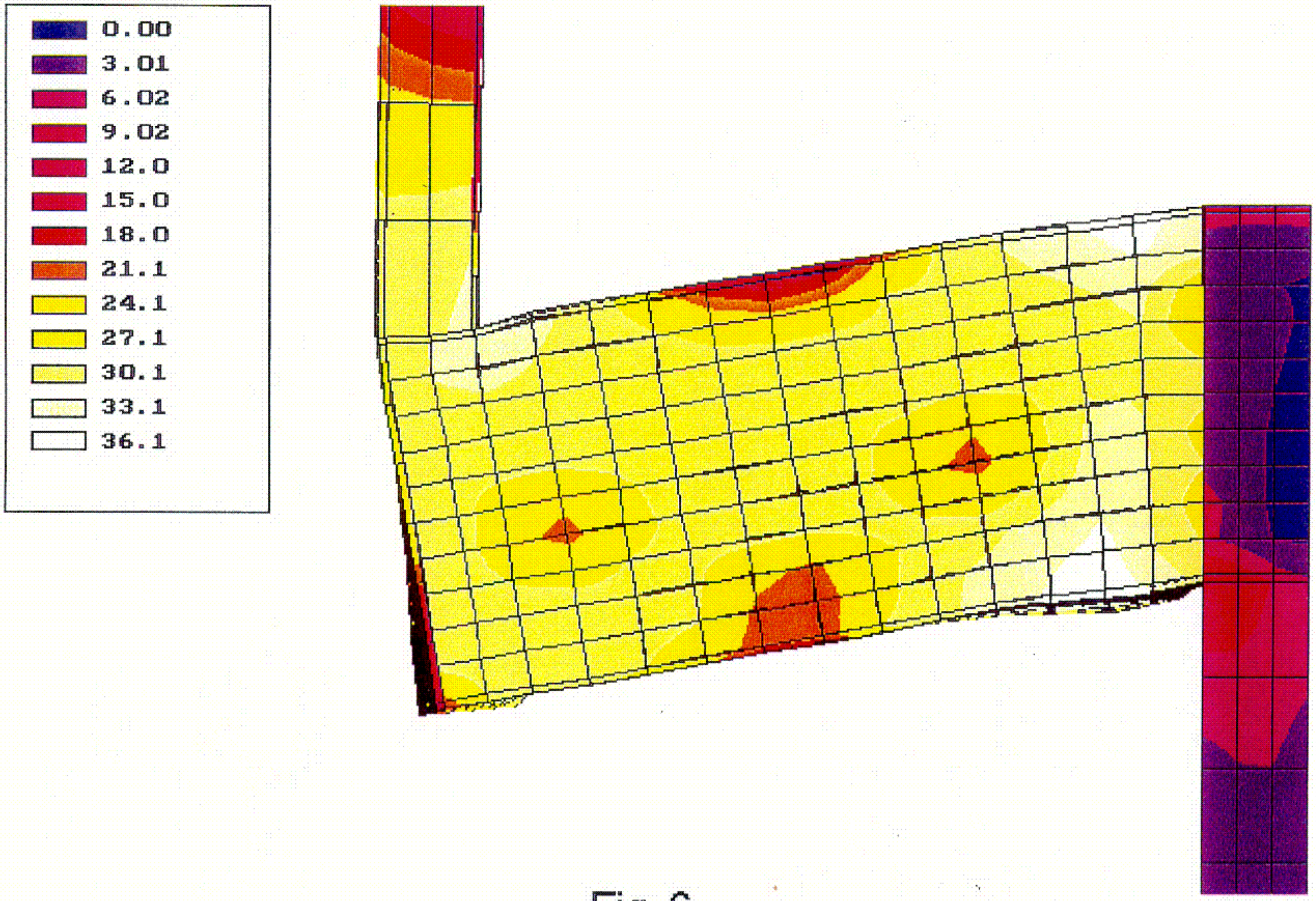
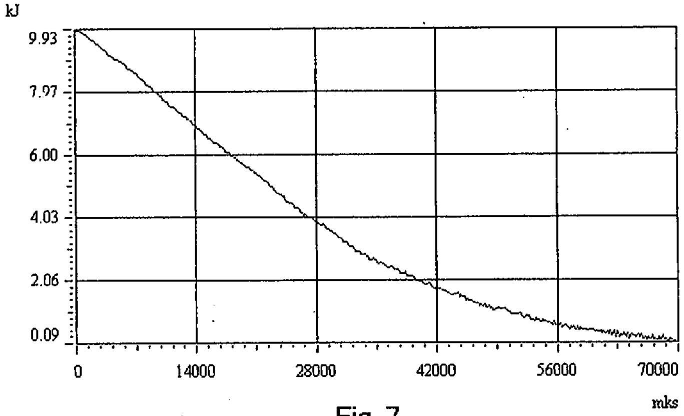


Fig. 6



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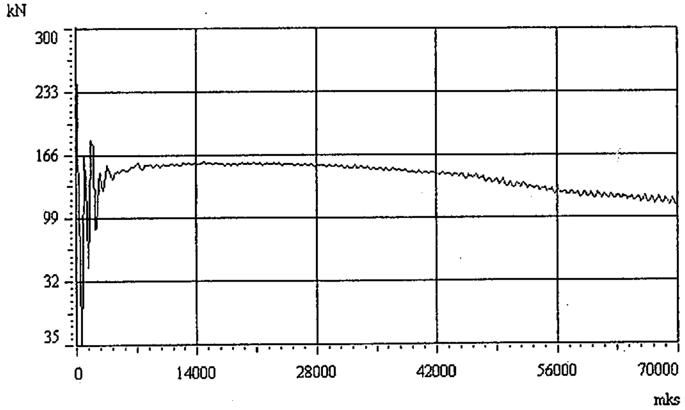

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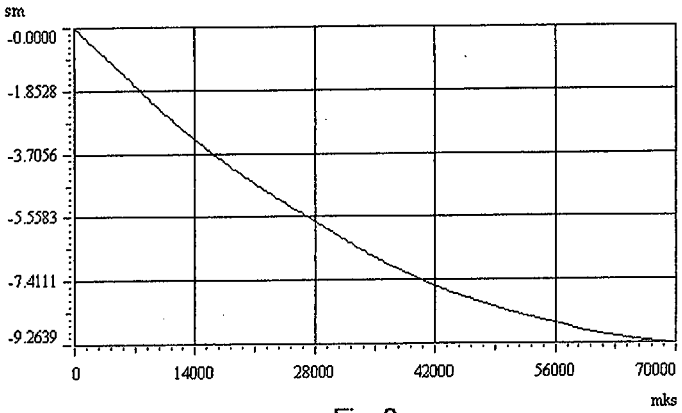

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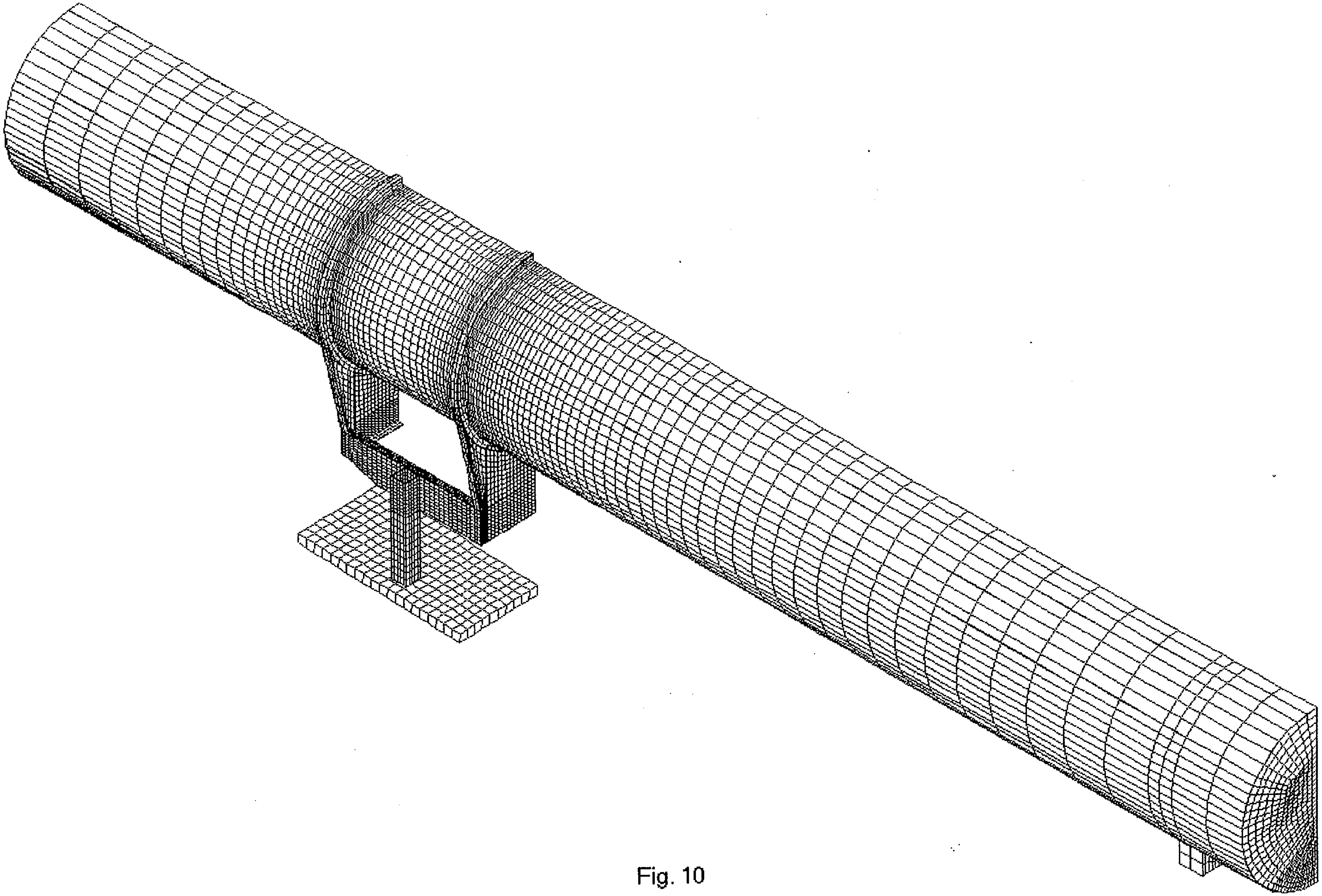

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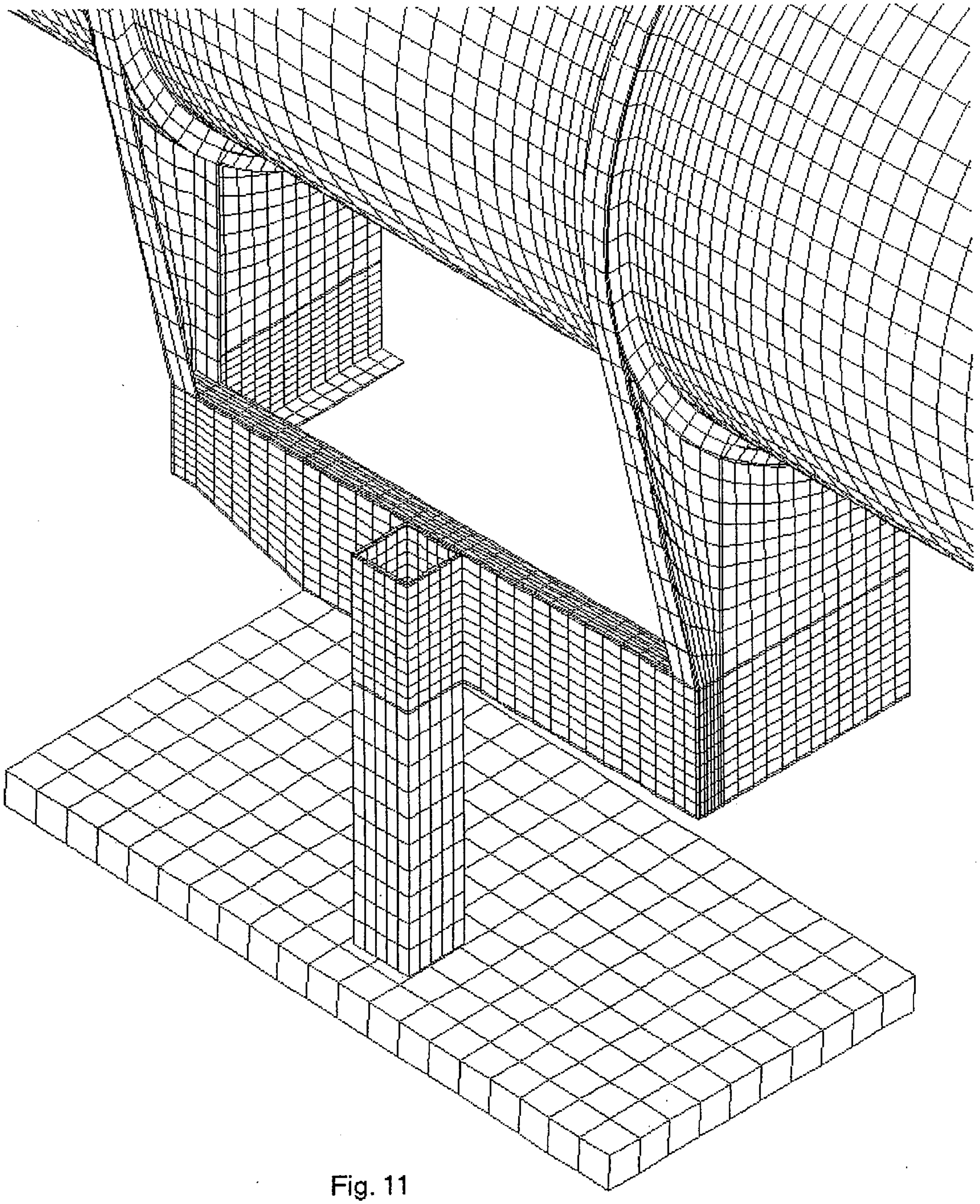

Fig. 11

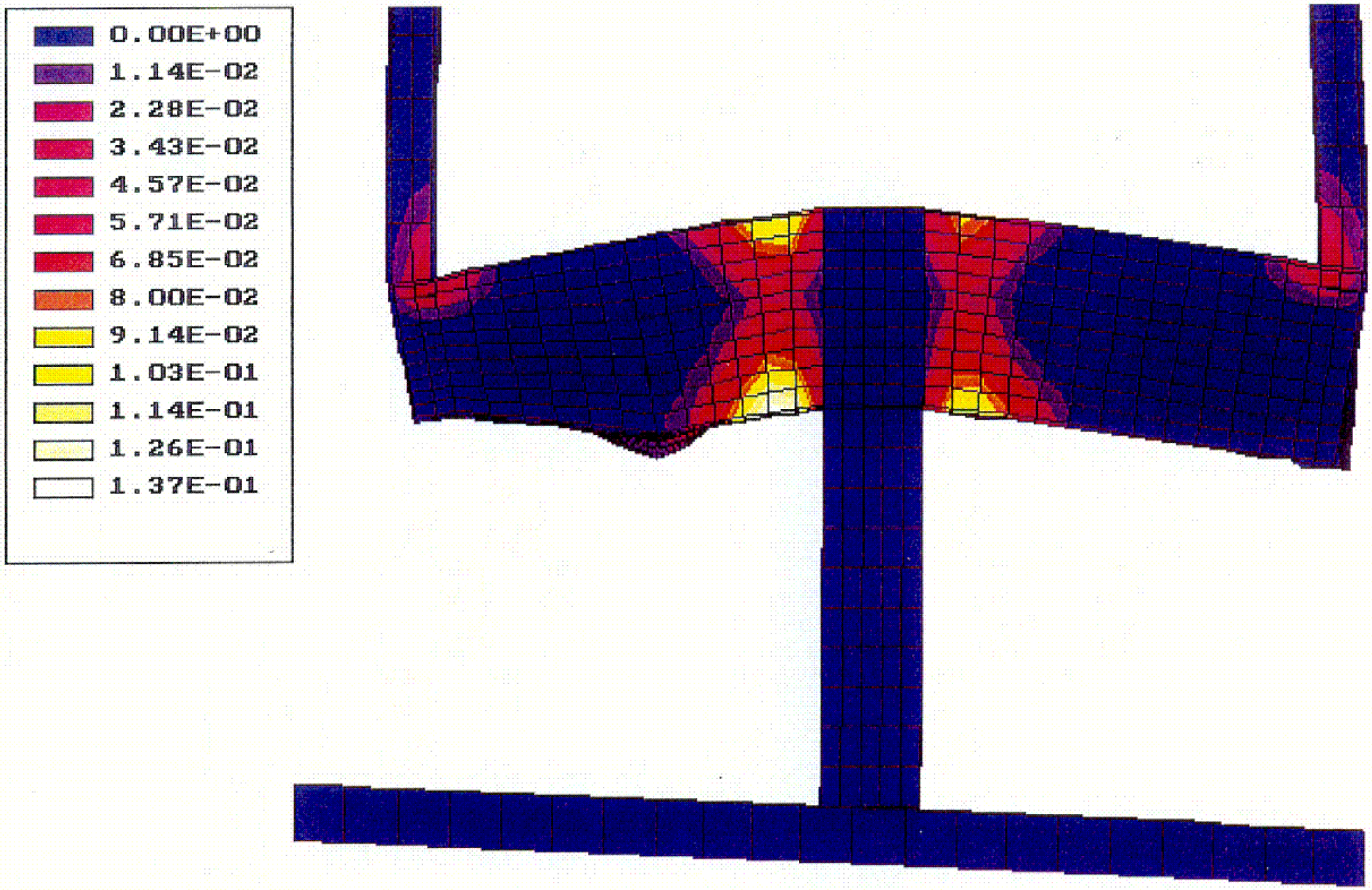
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Fig. 12

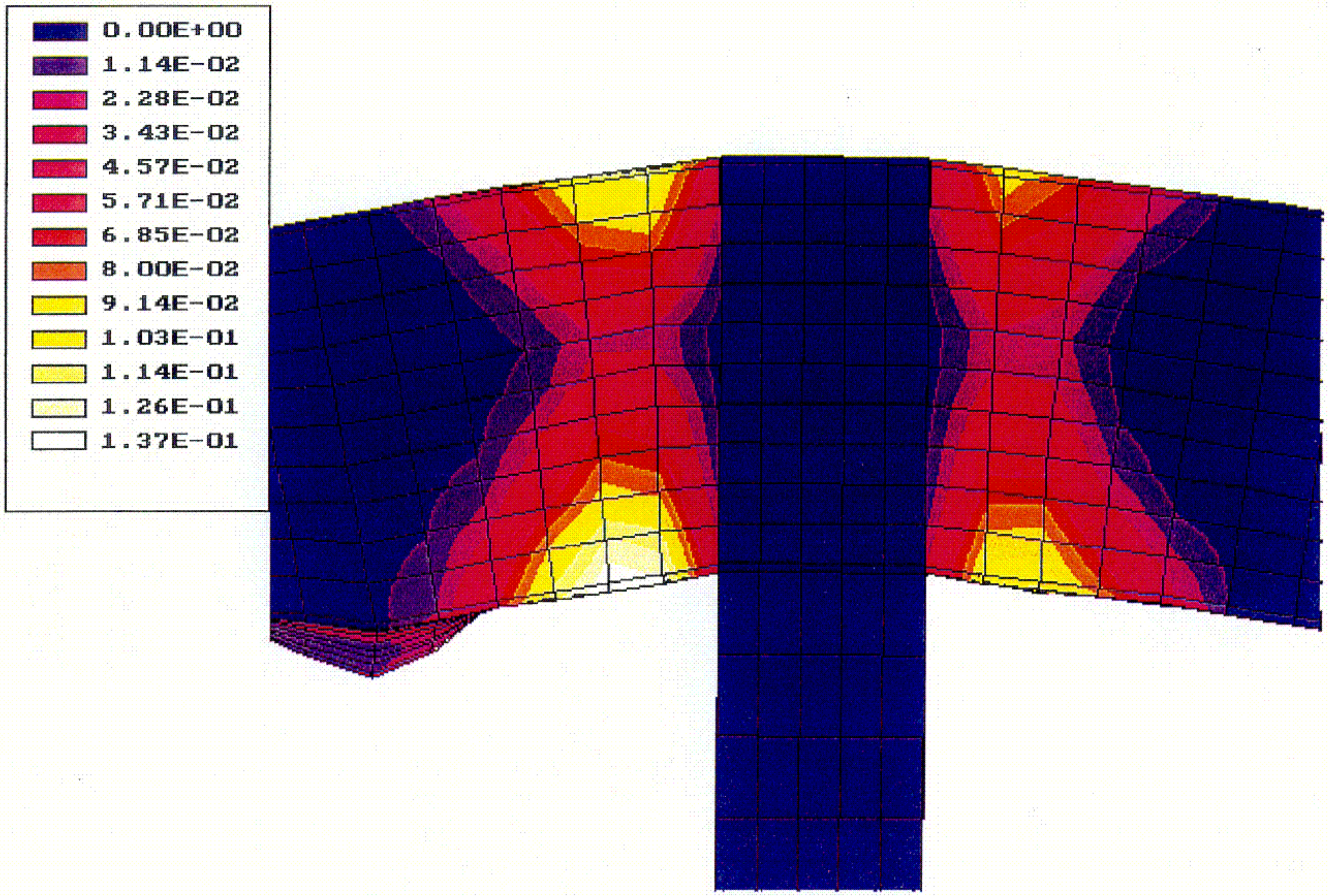
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Fig. 13

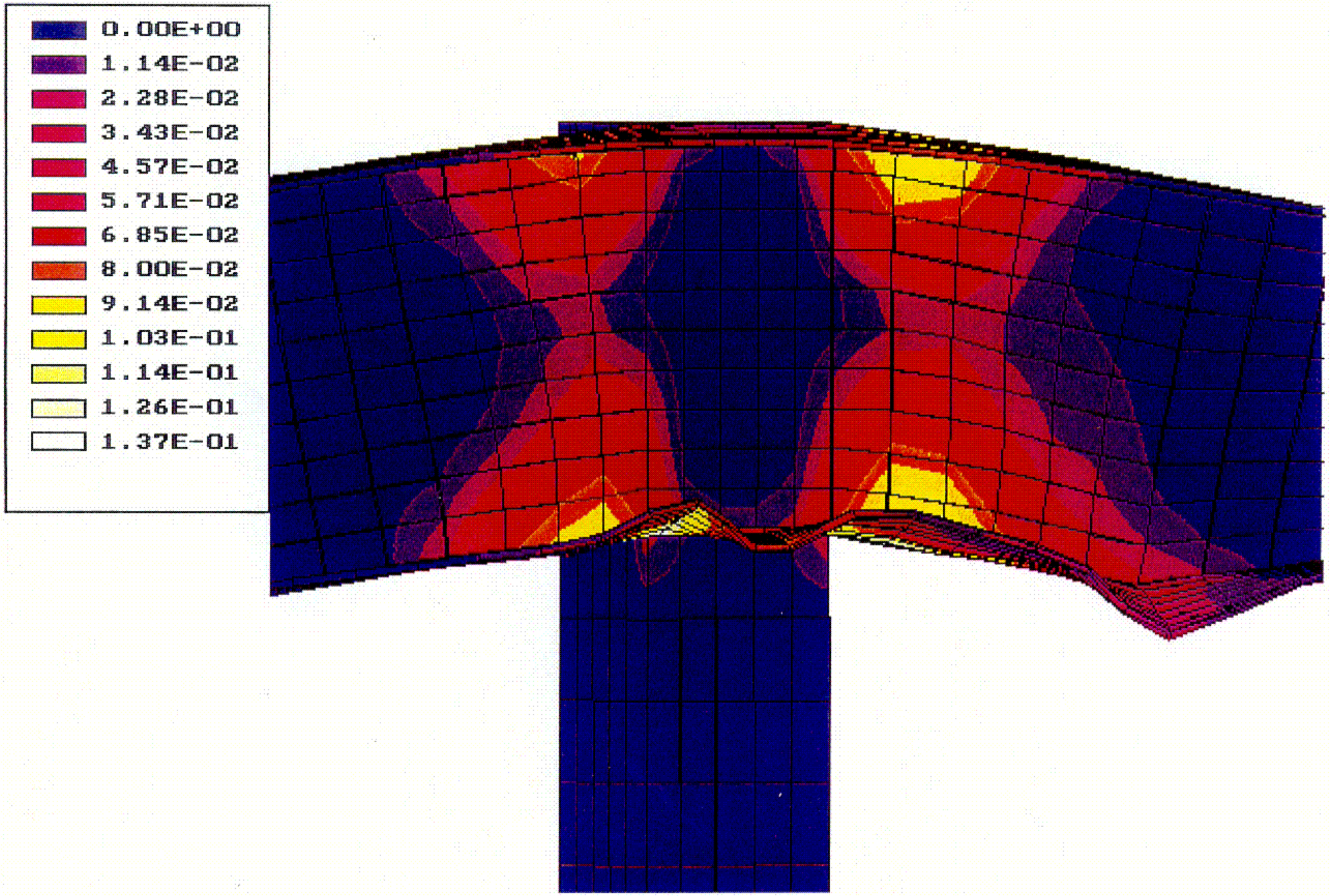
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Fig. 14

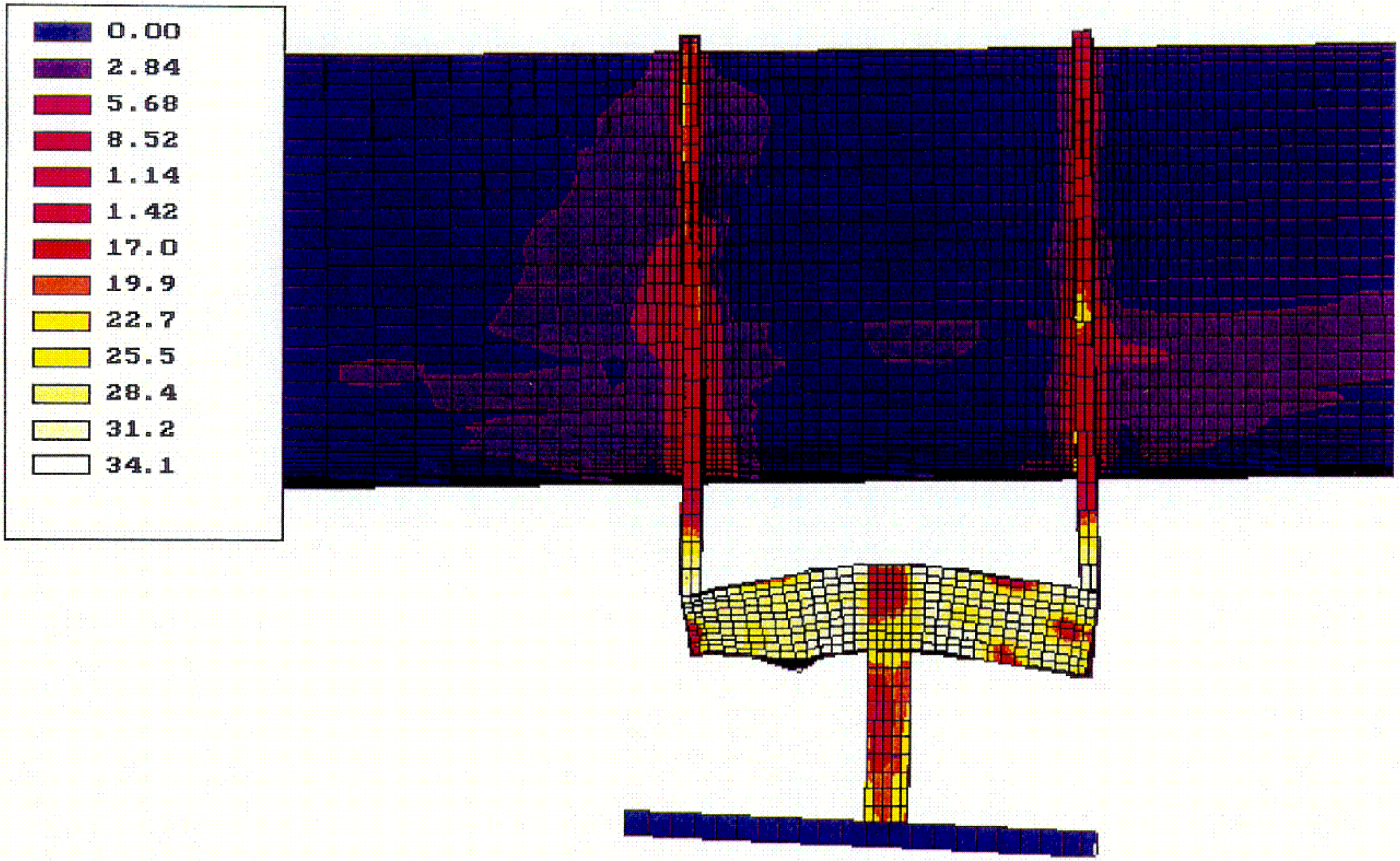


Fig. 15

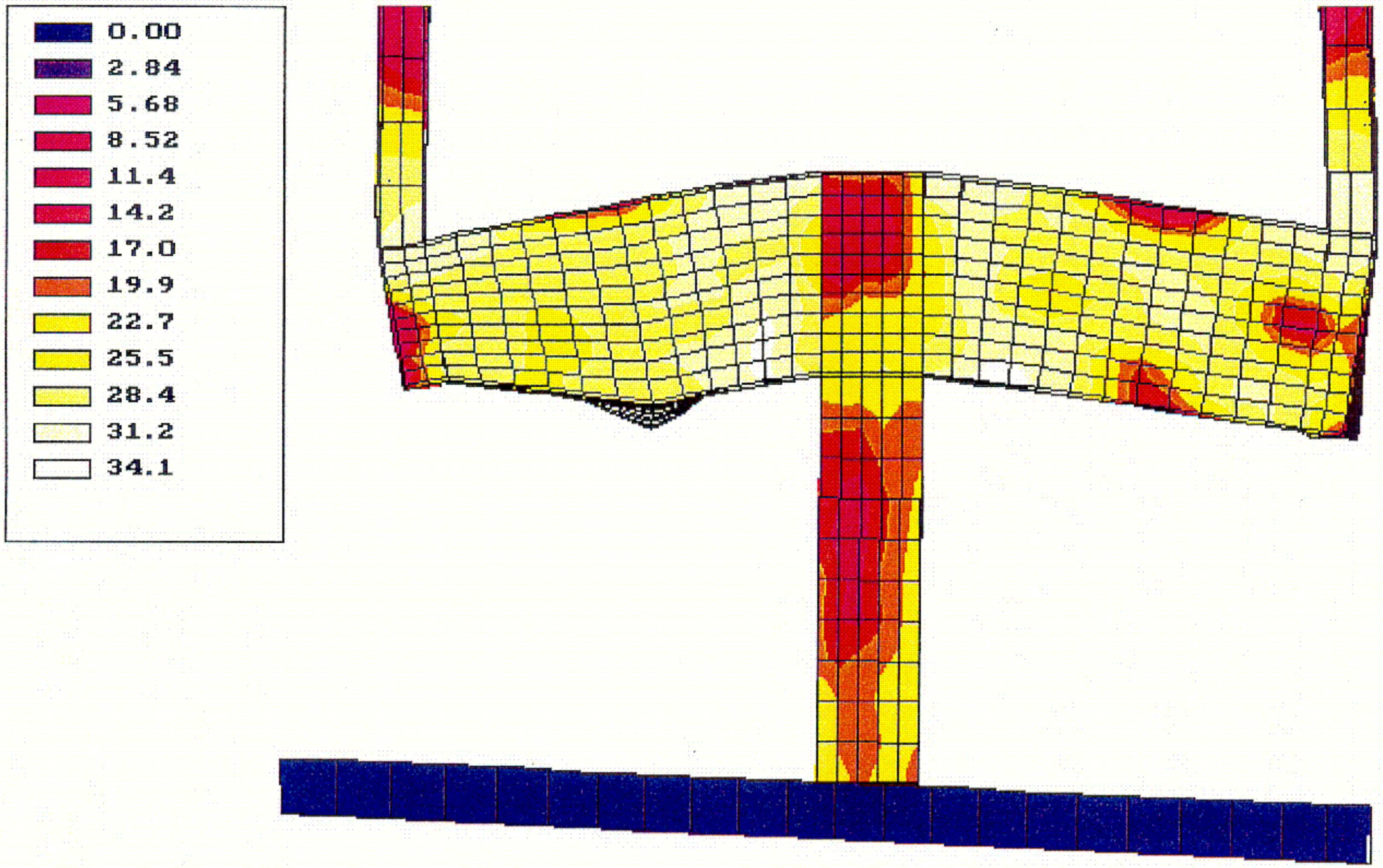


Fig. 16

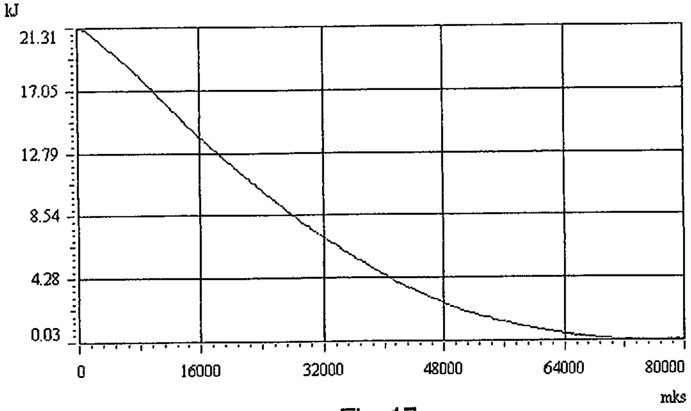


Fig. 17

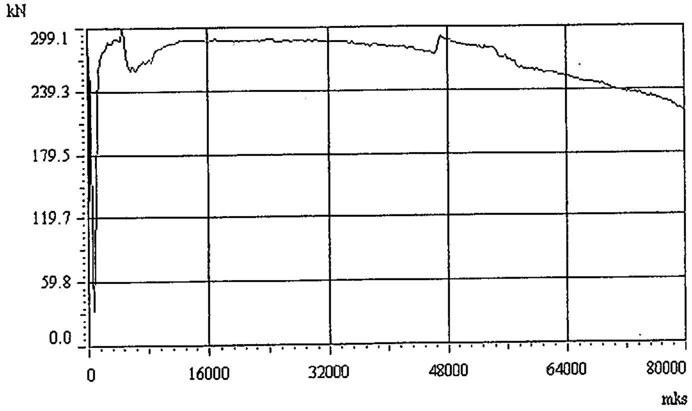
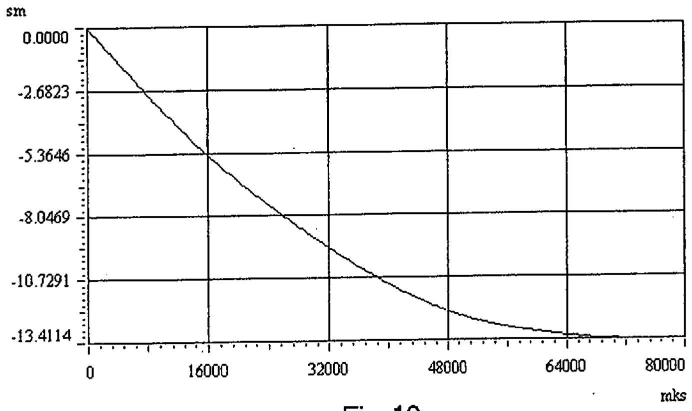


Fig. 18



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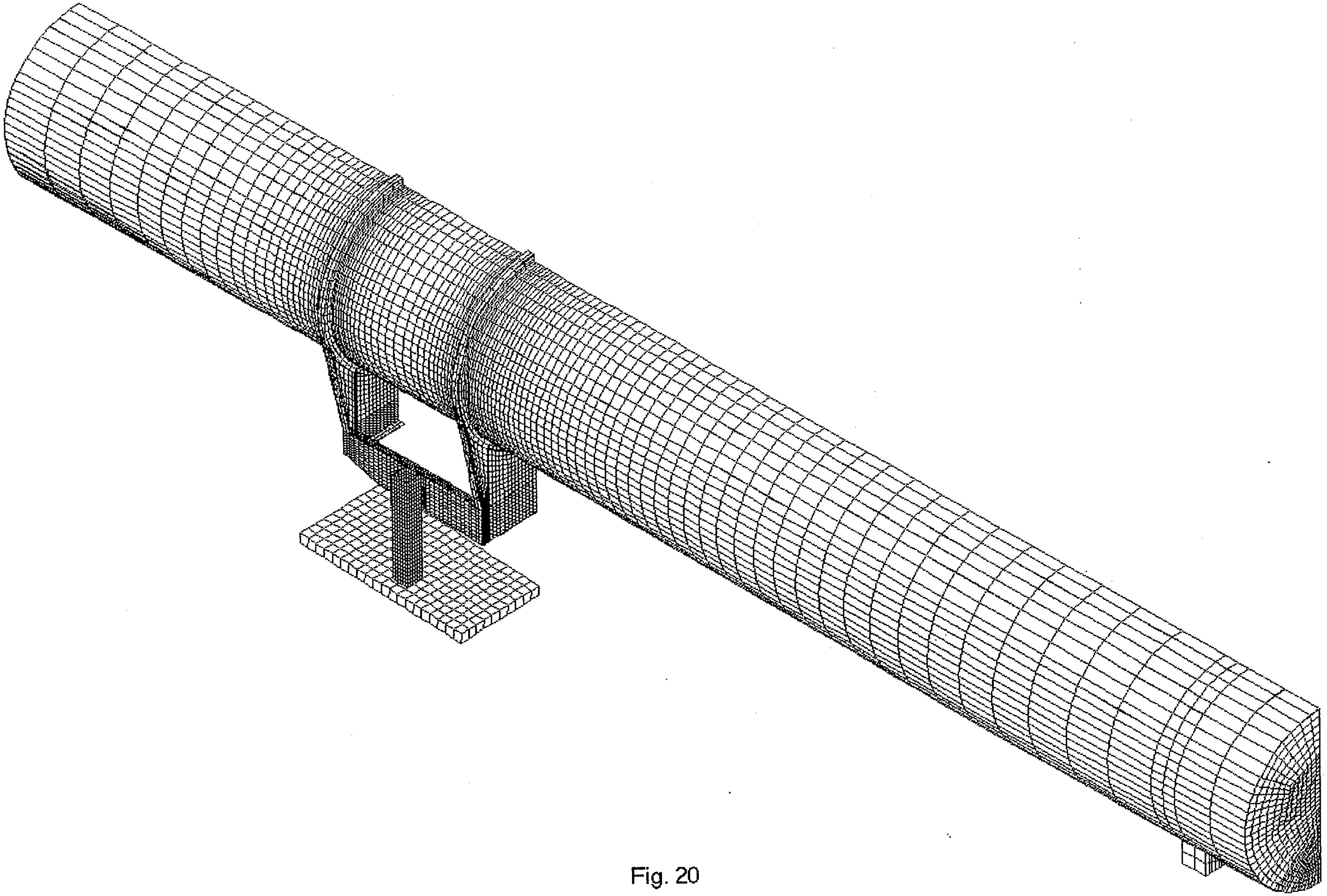

Fig. 20

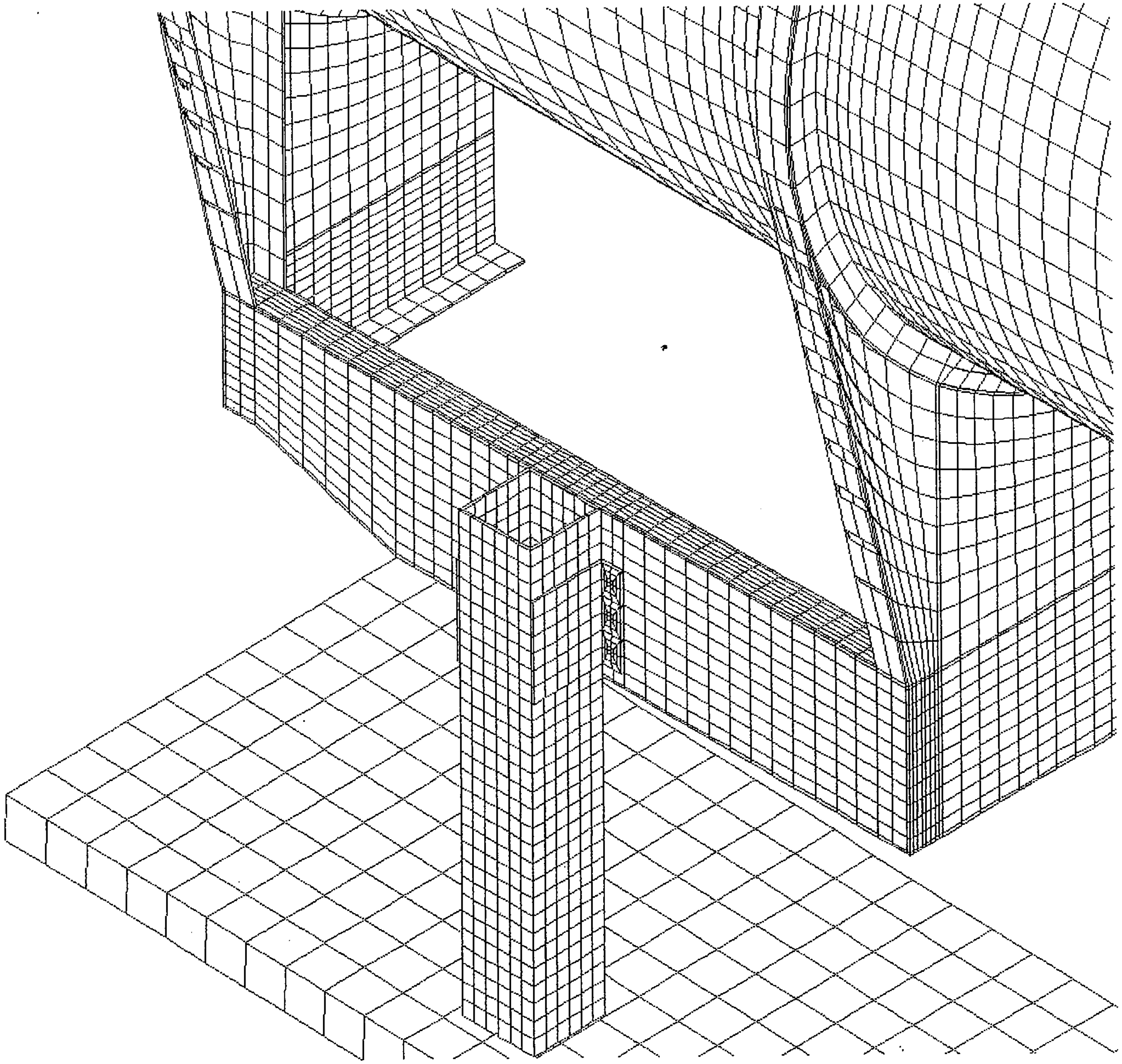

Fig. 21