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DESIGN REQUIREMENTS FOR A NEW AGS MARMAN CLAMP

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Collider Accelerator Department

Brookhaven National Laboratory

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BROOKHAVEN NATIONAL LABORATORY
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Upton, New York 11973

Accelerator Division
Technical Note

AGS/AD/Tech. Note No. 268

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J. Tuozzolo and W. Van Zwienen

November 20, 1986

Introduction

The present clamps used on the chamber-to-chamber Marman flanges in the AGS were designed and fabricated during the 1970 conversion for use with indium coated metal "C" rings. In 1981 aluminum diamond seals were introduced into the AGS which required a much higher sealing force. This resulted in the permanent deformation of some of the clamps when they were over torqued in an effort to attain a vacuum tight seal. Various design improvements were considered and a finite element stress program was used to find the weakness in the old clamp and to help in the design of a new Marman clamp.

The Problem

The present clamp (fig. 1) was designed as a quick disconnect device for the Marman flanges which were install during the AGS conversion. They were intended for use with indium coated metal "C" rings which required clamping forces of only 200 to 250 lbs/in to properly seal. (lbs/in refers to the force required per linear inch of seal. The 8 1/2 inch O.D. seal used on the Marman flange would require a total force of 5340 to 6675 lbs applied to the flange to seal). Unfortunately the indium coating has a low melting point and would leak whenever there was localized beam heating at a flange. In 1981 aluminum diamond seals were adapted from a CERN design for use in the AGS. While they are less susceptible to beam heating and cost about the same price, they require a much higher clamping force for leak tight operation. Estimates for proper force run as high as 600 to 700 lbs/in. which is beyond the design of the present clamp.

Clamps are routinely tightened until proper a seal is attained. This has resulted in clamps which are permanently deformed and are not reuseable. The damage occurs in two areas: the opening of the clamping arms and where the tightening bolt fits into the slot (fig 2). The wider opening moves the clamping point on the flange farther to the outside which makes a proper seal even more difficult to attain. Eventually, the clamp opens to the point where the outside diameter of the flange contacts the inside diameter of the clamp and further torqueing of the bolt does not provide additional sealing force. Further torqueing only results in deformation of the slot where the tightening nut pulls on the clamp.

During the summer 1986 shutdown, measurements were made on diamond seals which had been in service in the ring and seals which were installed but leaked. The thickness of the seal was measured to determine the extent to which the diamond had been crushed. The greater the crush the better the chance for a good seal since the sealing surface area would increase. The measurements were taken at the top, bottom and the both sides of the seal. The top coincides with the bolt location on the clamp, the bottom with the pin location and the sides with the midpoint of the clamp segments (fig. 3). The results are given below:

DIAMOND SEAL THICKNESS MEASUREMENTS

TOP	BOTTOM	RIGHT SIDE	LEFT SIDE	NOTES
.183	.183	.186	.186	
.180	.191	.200	.199	SPRAY SEALED
.195	.192	.194	.190	
.180	.185	.190	.196	SPRAY SEALED
.180	.180	.195	.200	
.190	.196	.200	.201	
.173	.173	.180	.175	SPRAY SEALED
.181	.186	.190	.193	LEAKER
.190	.178	.196	.198	LEAKER
.180	.187	.194	.190	LEAKER
.181	.180	.200	.195	LEAKER

The original purpose of the measurements was to determine the difference in clamping force from the pinned or hinged bottom side to the bolted top side of the clamp. While some differences were found, the largest difference in crush and therefore clamping force occurred at the midpoints of the segments. On six of the seals the crush at the sides was almost non-existent. Not surprisingly, 2 of those seals had residual spray sealant and another two leaked and were removed. These results indicate that the flanges which are machined flat to .005" are distorting under load because of insufficient support at the midpoints of the clamps. Fortunately there has been no indication of permanent deformation of the stainless steel flanges.

Design Considerations

The major design requirements for a new Marman clamp are:

1. It must have adequate strength to crush the diamond seal.

2. The overall dimensions of the new clamp should be as close as possible to the present clamp to avoid interferences and to allow installation 1f the RC network used between the vacuum chambers.

3. Continued use of aluminum for the clamp material is desirable because of its ease of fabrication, light weight, and low residual radiation.

The first major change considered was from the present single bolted design to a double bolted design (fig. 4) which is a common feature on commercial high strength clamps. By evenly torqueing both bolts on the new design, the clamping force should be applied more evenly at the top and bottom. With the old design the bottom pivot was located at the theoretical fully tightened position. As the bolt was tightened the clamping force was applied to the bottom of the flange before the top. Over or under tightening of the bolt would result in an uneven clamping force and distortion of the flange and the seal. Only three of the seals measured in the list above had an even crush on the top and the bottom.

The most important change considered is a revised flange cross section which would give the flange greater stiffness around its entire circumference and lower stresses to eliminate the permanent distortion of the clamp at high loads. Various cross section designs were considered and analyzed for the new clamp.

Analysis

Using the ANSYS stress analysis program, the forces on the clamp and their resultant stresses were analyzed. The cross section of the clamp was modeled as a solid axisymetrical part which was symetrical at its center line. Four different clamp cross sections were eventually considered and analyzed (fig. 5):

1. The present clamp design.

2. The present design without the scalloped sides.

3. The clamp proposed by Connectors Ltd.

4. The Connectors Ltd. clamp with a narrower width to allow compatibility with the RC network connection.

The elements for each clamp analysis and the associated node locations with applied pressure points and symmetry boundary conditions are shown in appendices 1 thru 4 respectively. Also given in the appendices are some of the stress plots generated as results of the analysis. Pressures which are equivalent to sealing forces of 300, 450, 600, and 700 lbs./in. were applied in separate load steps 1 through 4 respectively. The SIGE plots refer to equivalent stress and the SIG1 plots refer to tensile stress plots. A summary listing of the maximum stresses and deflections for all of the load cases are given below:

CLAMP DESIGN	CLAMP FORCE(LBS/IN) [STEP]	MAX TENSILE STRESS (PSI) SIG1	MAX EQUIVALENT STRESS (PSI) SIGE	MAX DEFLECTION (INCH)
1	[1] 300	23,694	19,759	.0033
1	[2] 450	35,530	29,629	.0050
1	[3] 600	47,388	39,518	.0066
1	[4] 700	55,285	46,104	N/A
2	[1] 300	22,847	19,273	.0030
2	[2] 450	33,722	28,447	.0045
2	[3] 600	44,564	37,593	.0059
2	[4] 700	51,986	43,854	.0069

CLAMP DESIGN	CLAMP FORCE(LBS/IN) [STEP]	MAX TENSILE STRESS (PSI) SIG1	MAX EQUIVALENT STRESS (PSI) SIGE	MAX DEFLECTION (INCH)
3	[1] 300	9,706	7,896	.0011
3	[2] 450	14,560	11,846	.0017
3	[3] 600	19,412	15,793	.0023
3	[4] 700	22,646	18,424	.0027
4	[1] 300	10,313	8,255	.0010
4	[2] 450	15,631	12,513	.0016
4	[3] 600	20,657	N/A	.0021
4	[4] 700	24,091	19,289	.0024

The present clamp design (Appendix 1) is highly stressed even at the lowest clamping force. The maximum tensile stress (from Mohrs circle) at 300 in/lbs (sig1) was 23,694 psi along the inner radius of the clamp. The 6061-T6 aluminum used to fabricate these flanges has a yield strength of only 40,000 psi leaving a safety factor of less than 2 to 1. Even if the equivalent stress is used as the governing stress for design considerations, the stress of 39,518 psi at 600 lbs/in load leaves no safety factor to prevent yielding and permanent distortion of the clamp. As shown in the stress plots the highest stresses occurred at the radius bend where the clamp's arms extended to hold the flange. The arms are thinnest in that area and the radius may be acting as a stress raiser. Originally the designers may have preferred flexibility in the clamp to allow even clamping of the C rings. Slots were added to the clamp which allowed even greater flexibility than this model would indicate. Unfortunately this does not work for the diamond seals. It should be noted here that the deflection results are for only half of the clamp segment as measured at the flange contact point. The total deflection of the clamp surfaces would be twice the numbers given above.

The first alternative design considered (Appendix 2) had the same overall dimensions without the scalloped sides or the slots of the present clamp. This design would add material to the area where the present clamp had its highest stresses; but, the results of the analysis indicated little change in the maximum stresses. A review of the stress plots indicated the area of high stress moved slightly around the radius toward the top of the clamp. Here the thickness of the clamp between its O.D. and I.D. is now thinner than the arm where material was added. Material would have to be added to the clamp's O.D. for improved performance which would change its overall dimensions.

Connectors Ltd. Inc. provides Marman clamps for CERN where diamond seals are used extensively without problem. They provided a clamp design proposal for the AGS which was analyzed. Its cross section, shown in Appendix 3, not only has thick arms but also a thick section between the I.D. and the O.D. The results of the analysis quickly indicated the advantages of this design. The highest stresses still occurred along the inside radius but they were almost 60% less than the stresses on the present design. The deflection of the clamp was also only a third of the present design. This clamp has the strength characteristics we require but may be too wide for compatibility with the AGS.

The last design considered (Appendix 4) was a modified version of the Connector's Ltd. design with a narrower cross section. This allows compatibility with the new RC network design and will fit in tight areas such as the rf cavities. The loss in arm thickness increased the maximum stress by only 6% and had no measurable affect on the deflection of the clamp. The maximum tensile stress at the highest clamping force (24,091 psi), is close to the maximum we now have on the present clamp at the lowest clamping force. With the rule now in place of using a torque wrench to tighten the clamps there should not be any chance of over torqueing and distorting these clamps.

Prototype Testing

Two prototypes were fabricated using design #2 as shown in figure 4. Both clamps were tested with diamond seals and were leak tight each time tested. The results below give the sealing torque on the clamp bolts and the measurements on the seals after they were removed:

DIAMOND SEAL THICKNESS MEASUREMENTS

TORQUE (IN.LBS)	TOP	BOTTOM	RIGHT SIDE	LEFT SIDE	NOTES
200	.182	.184	.196	.196	
250	.180	.182	.187	.191	REUSED SEAL
350	.173	.182	.188	.189	

As indicated by the computer analysis this clamp design has stress and deflection problems similar to the present clamp and the results above are very similar to those for the present clamp. The seals were well crushed at the top and bottom where the heavy lugs on the clamp reinforced the outside diameter. At the midpoints where the clamp cross section matched the analysis cross section, the amount of crush was marginal even though vacuum tight seals were achieved each time. Measurements were also taken of the width of the clamps while in place and fully torqued. At 200 in.lbs, the midpoints expanded almost .020" and the lug areas only expanded .010". (These measurements were taken at the outside of the arms and not at the flange to clamp contact points as given in the analysis). It was also noted that these clamps sealed at lower torques than the present clamp which are torqued to 300 in.lbs. normally and higher when a seal is not achieved. The seal which was torqued up to 350 in.lbs. was only torqued that high for test and measurement purposes.

The prototype clamp had much heavier bolt lug areas and was a solid machined fabrication rather than a weldment. At one point in the testing the clamp was torqued up to 600 in.lbs. Measurements made at the lug and bolt area indicated that there was little distortion under load and no permanent distortion after the load was removed. Permanent distortion of the clamping arms was found only at the midpoints. The arms were found to have moved out by about .006 of an inch. This again emphasizes the importance of the larger outside diameter in reducing the stresses on the clamp. These clamps are presently being used in the AGS ring as replacements for standard clamps which were distorted beyond use.

Conclusions

The new clamp for the AGS will be a double-bolted design with a cross section similar to design #4. Two clamp designs would be considered acceptable at this time:

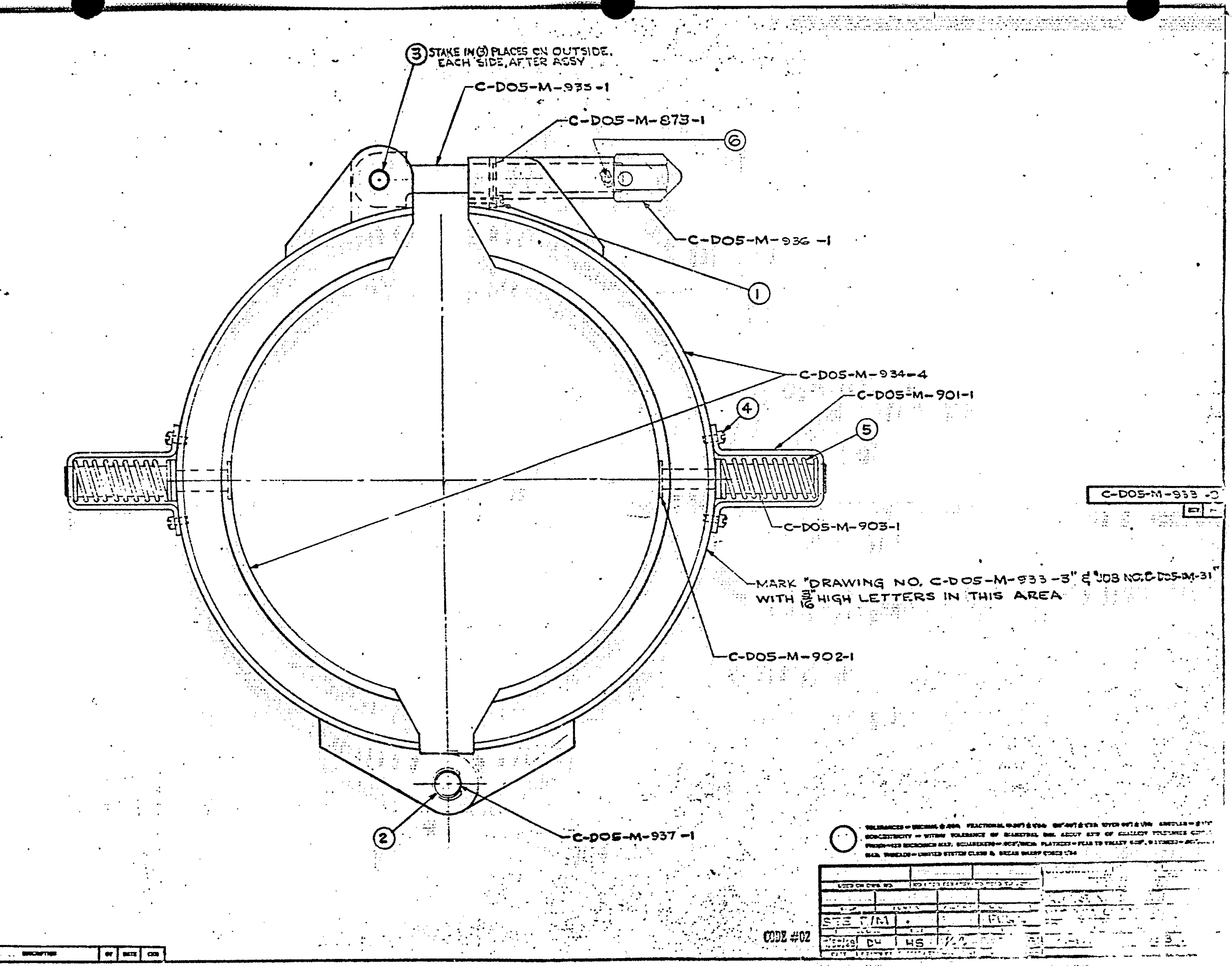
1. A segmented clamp as provided by Connectors Ltd. (figure 6).

2. The design used on the prototype clamps with an O.D. of 10-1/2 inches.

The connector's design which uses a stainless steel banding and four separate segments should be acceptable based on the experience of CERN with the clamp. The stainless steel band is .080 thick by 1.250 wide, but in the area where the bands pass over the lugs the load carrying cross section is reduced to .080 thick by 1 inch wide. Because of this, Connectors Ltd. recommends a maximum torque on the clamp bolts of 260 in. lbs. which they claim gives a sealing force of 840 lbs. per inch. According to our on calculations (Appendix 5), this load will result in a stress of 44,000 psi on the stainless steel strap giving a safety factor of better than 2:1. Also the greater elastic modulus of the stainless steel band serves as an extra backing for the aluminum clamping sections improving their stiffness. The alumuminum proposed for the clamp is a forging of hillininum which is equivalent to 2024-T4. Its yield stress (45,000 psi) and elongation (20%) is higher than the 6061-T6 aluminum (40,000 psi/ 12%) used in the present clamp.

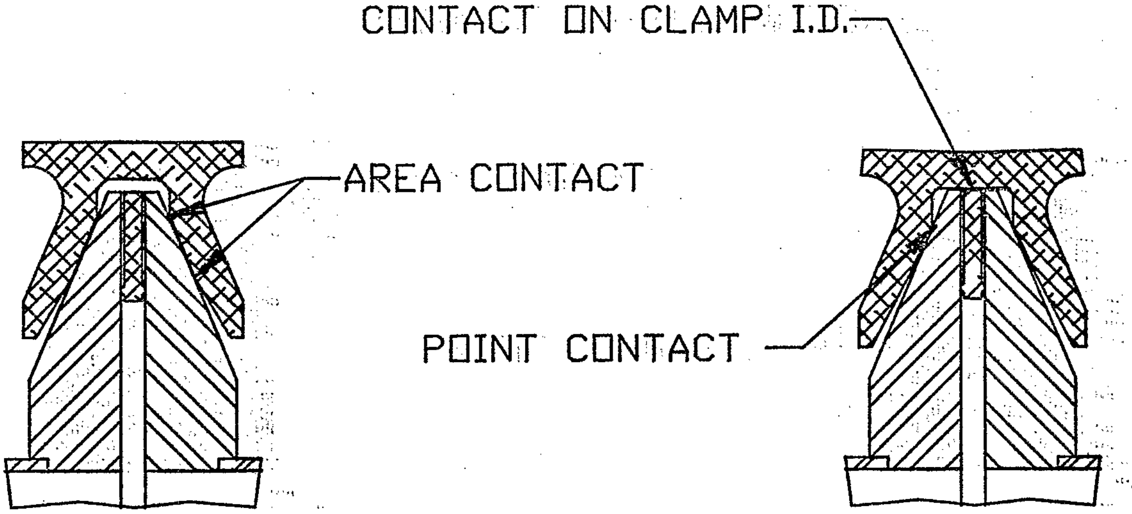
The present prototype design with a larger O.D. would be an excellent alternative. It would have all the design qualities required and may be more adaptable to the present clamp supports than the connector's design. As noted previously the only short coming of the first prototypes was distortion of the arms under high loads, increasing the O.D. should eliminate this problem. The final choice between both designs will be the manufacturing cost.

FIG. 1



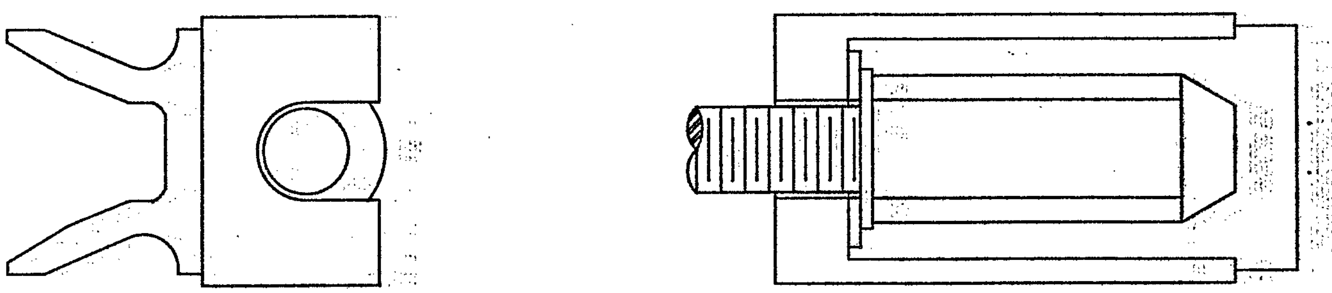
③ STAKE IN (3) PLACES ON OUTSIDE. EACH SIDE, AFTER ASSY
C-DO5-M-935-1
C-DO5-M-873-1
⑥
C-DO5-M-936-1
①
C-DO5-M-934-4
C-DO5-M-901-1
④
⑤
C-DO5-M-933-3
C-DO5-M-903-1
MARK "DRAWING NO. C-DO5-M-933-3" & "JOB NO. C-DO5-M-31"
WITH 3/16" HIGH LETTERS IN THIS AREA
C-DO5-M-902-1
②
C-DO5-M-937-1
CODE #02

CLAMP DEFORMATION

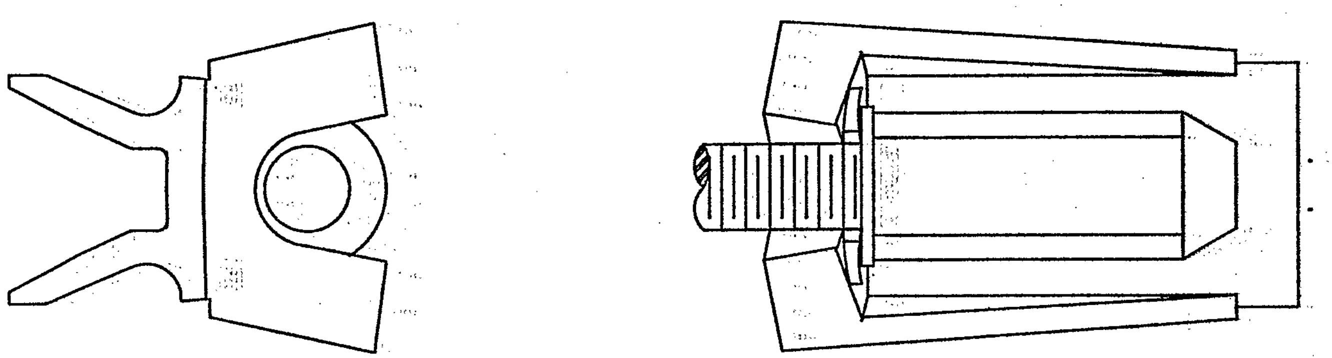


IDEAL CLAMPING   DEFORMED CLAMPING

SLOT DEFORMATION

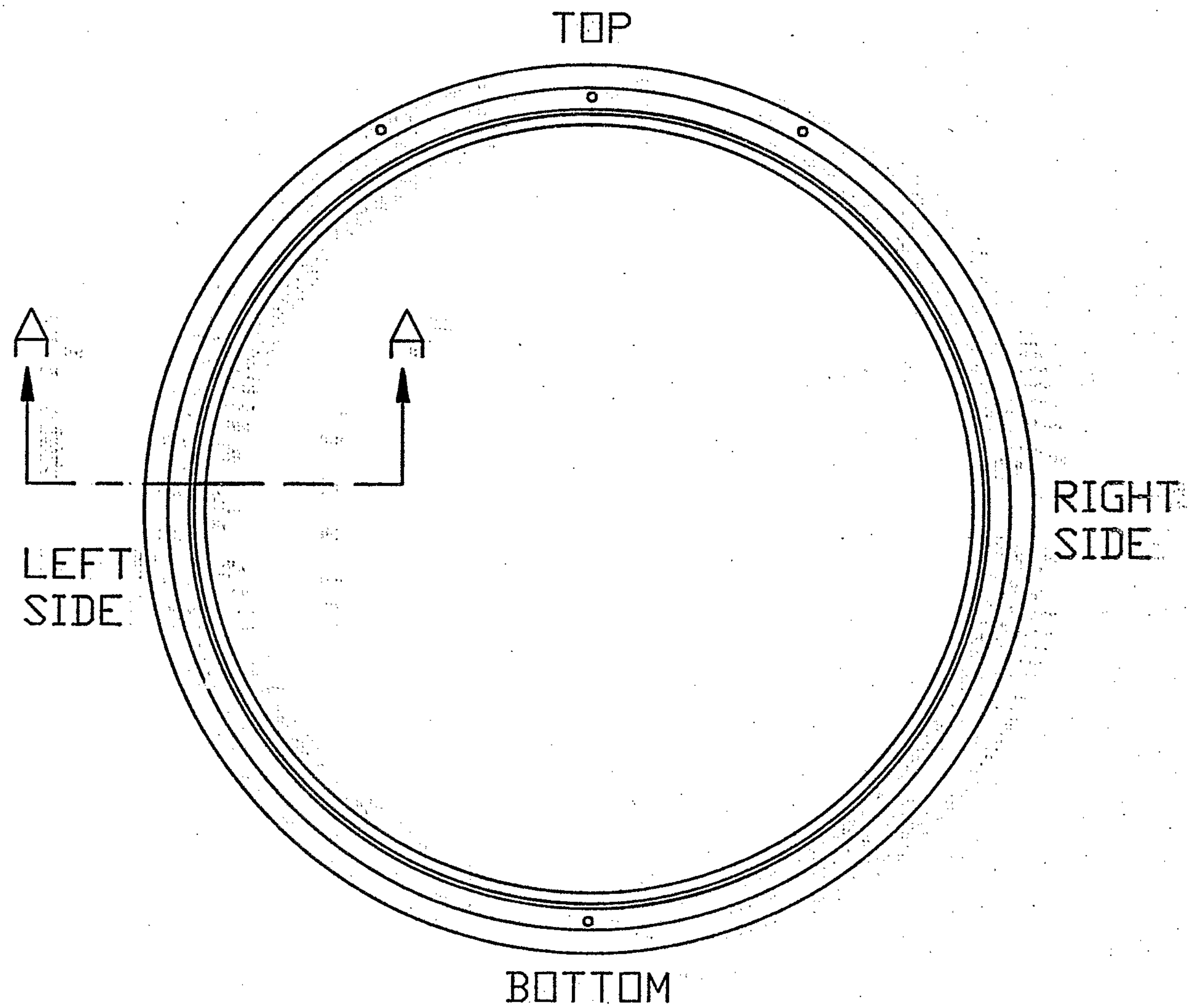


UNDEFORMED CLAMP



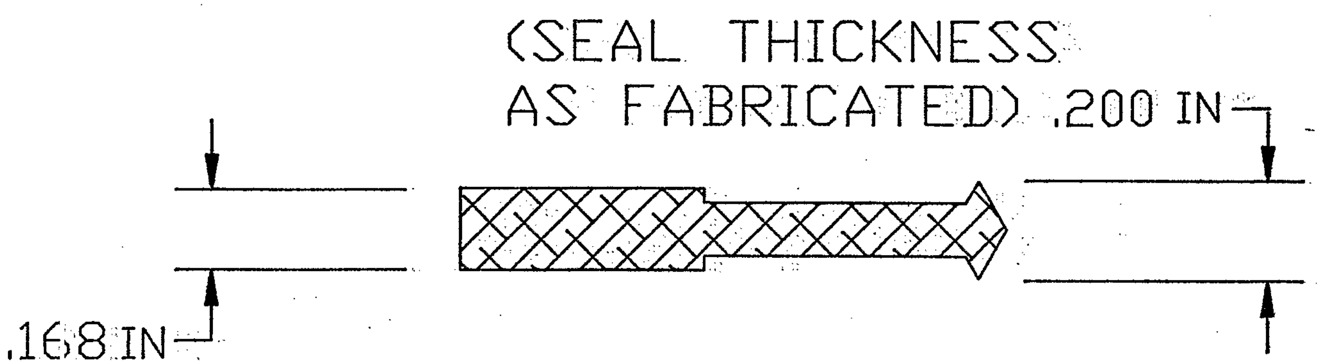
DEFORMED CLAMP

FIGURE 2



UPSTREAM VIEW

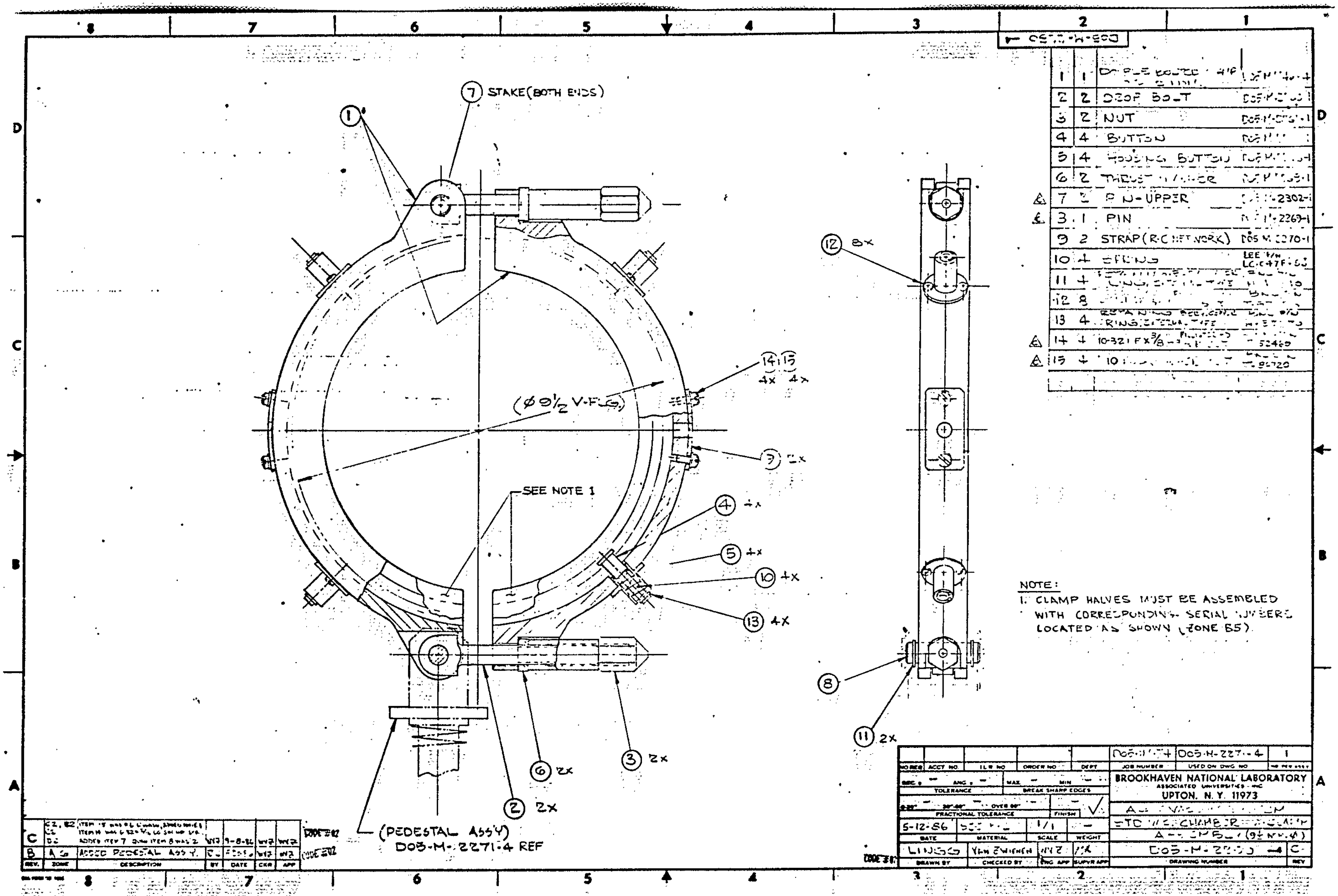
(PINS ALWAYS PROTRUDE FROM DOWNSTREAM SIDE TO ENGAGE DOWNSTREAM FLANGE SLOTS)



VIEW A-A

FIGURE 3

FIG. 4



STAKE (BOTH ENDS)
SEE NOTE 1
(PEDESTAL ASSY)
D03-M-2271-4 REF
NOTE:
1. CLAMP HALVES MUST BE ASSEMBLED WITH CORRESPONDING SERIAL NUMBERS LOCATED AS SHOWN (ZONE B5).
DROP BOLT
NUT
BUTTON
PIN-UPPER
PIN
STRAP (R.C. NETWORK)
ADDED PEDESTAL ASS'Y.
BROOKHAVEN NATIONAL LABORATORY
ASSOCIATED UNIVERSITIES, INC.
UPTON, N.Y. 11973
5-12-86

CLAMP CROSS SECTIONS

(DIMENSIONS IN INCHES)

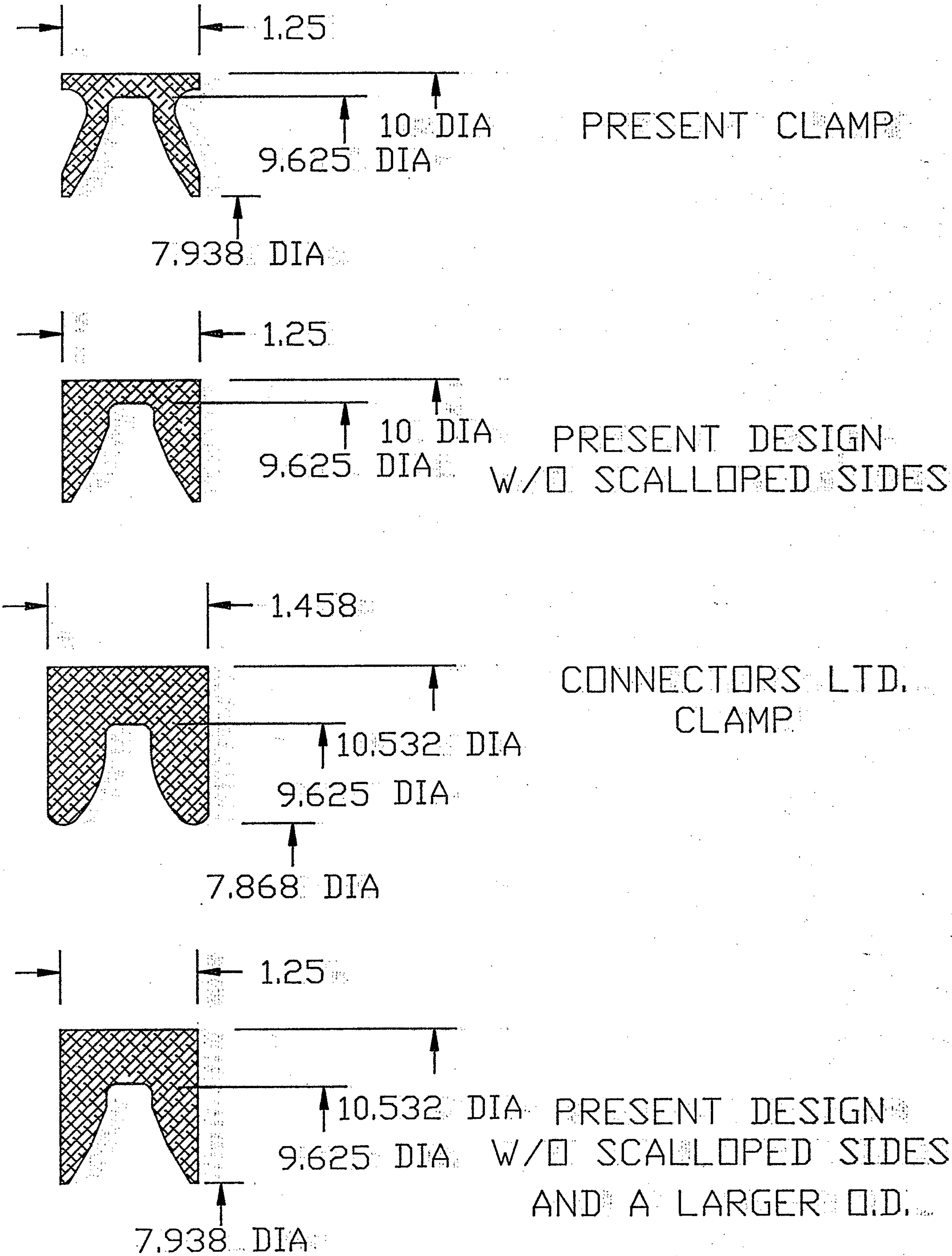


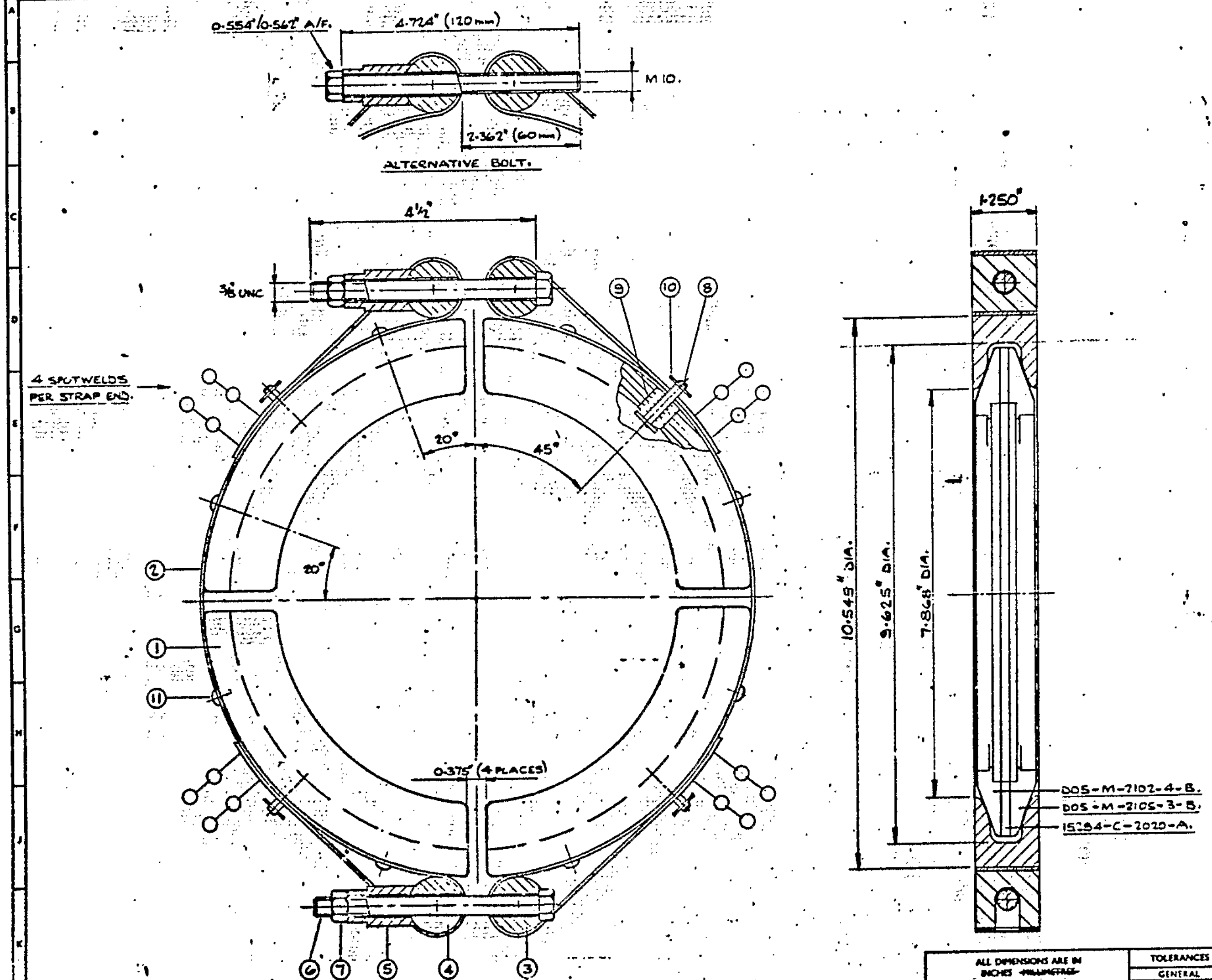
FIGURE 5

DRG. No. SK 102. THIRD ANGLE PROJECTION

DRAWING OFFICE DATA: BROOKHAVEN NATIONAL LABORATORY.

REVISIONS

ISSUE	ZONE	DESCRIPTION	DATE	CHANGE No.
1	—	ORIGINAL ISSUE	15-7-86	—



NOTES:-

(1) MATERIAL FOR SEGMENTS, ALUMINIUM ALLOY RR 58. SEE DATA SHEETS.

(2) CLAMP ASSEMBLY TO BE DE-GREASED.

(3) SIDES OF INTERNAL PROFILE OF SEGMENTS, & BOLT THREADS, TO BE COATED WITH MOLYKOTE R DRY-FILM LUBRICANT THINLY & UNIFORMLY.

(4) FINISHED CLAMP ASSEMBLIES TO BE INDIVIDUALLY SEALED IN PLASTIC BAGS.

(5) SPINDLES (ITEM 8) TO BE ENAMEL COATED ON THE HEAD OR MADE FROM A NON-CONDUCTIVE MATERIAL.

	PART No	QTY	DESCRIPTION	REMARKS
11		8	DRIVE SCREW	
10		4	CIRCLIP	
9		4	BELLEVILLE WASHER	
8		4	SPINDLE	
7		2	HEX NUT 3/8" UNC	
6		2	HEX BOLT 3/8" UNC x 4½"	
5		2	SPACER	
4		1	SLOTTED TRUNNION	
3		3	DRILLED TRUNNION	
2		2	STRAP	
1		4	SEGMENT	

ALL DIMENSIONS ARE IN INCHES ~~MILLIMETRES~~

BREAK ALL SHARP EDGES 0.1 TO 0.5mm (.004" TO .020") UNLESS OTHERWISE SPECIFIED. POSITIONAL TOLERANCES ARE REGARDLESS OF FEATURE SIZE.

TOLERANCES	MM	INCHES
GENERAL	—	±0.010"
MACHINED		
ANGLES		
HOLE POSITIONS		

DRAWN:- S.H.

CHECKED:-

APPROVED:-

HEAT TREATMENT | PROTECTIVE TREATMENT

PART NAME: VEE CLAMP FOR BELLOWS FLANGE.

DRAWING No. SK 102

NOTE: SHEETS 1 & 2 ARE A4 SIZE.

SCALE 1:1 SHEET 3 OF 3 SHEETS

CONNECTORS (KLY) LTD
KEIGHLEY, YORKS, ENGLAND.

APPENDIX

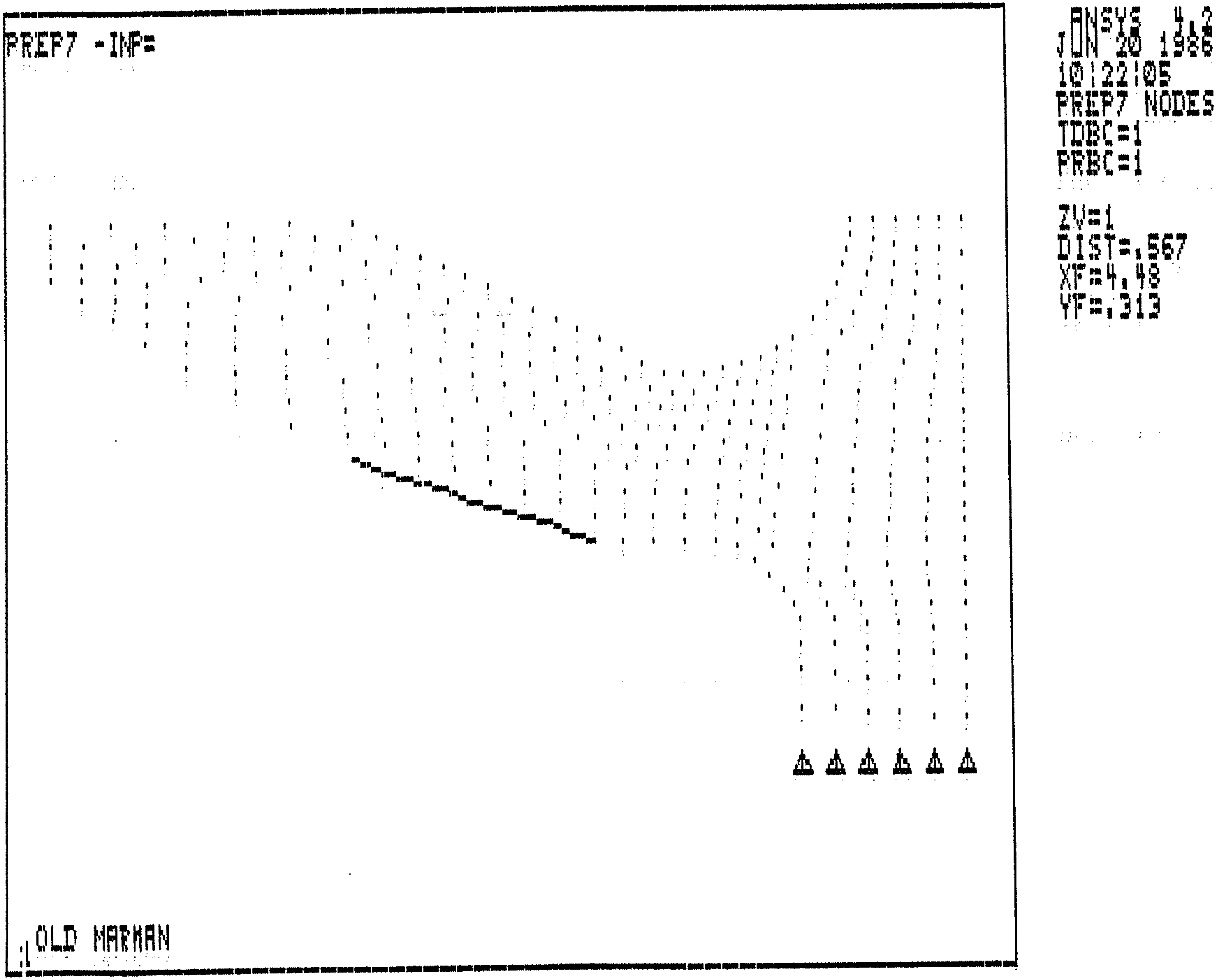
A.1.1 Old Marman Node Plot
A.1.2 Old Marman Load Step 1, Tensile Stress
A.1.3 Old Marman Load Step 1, Tensile Stress Enlarged
A.1.4 Old Marman Load Step 4, Tensile Stress Enlarged

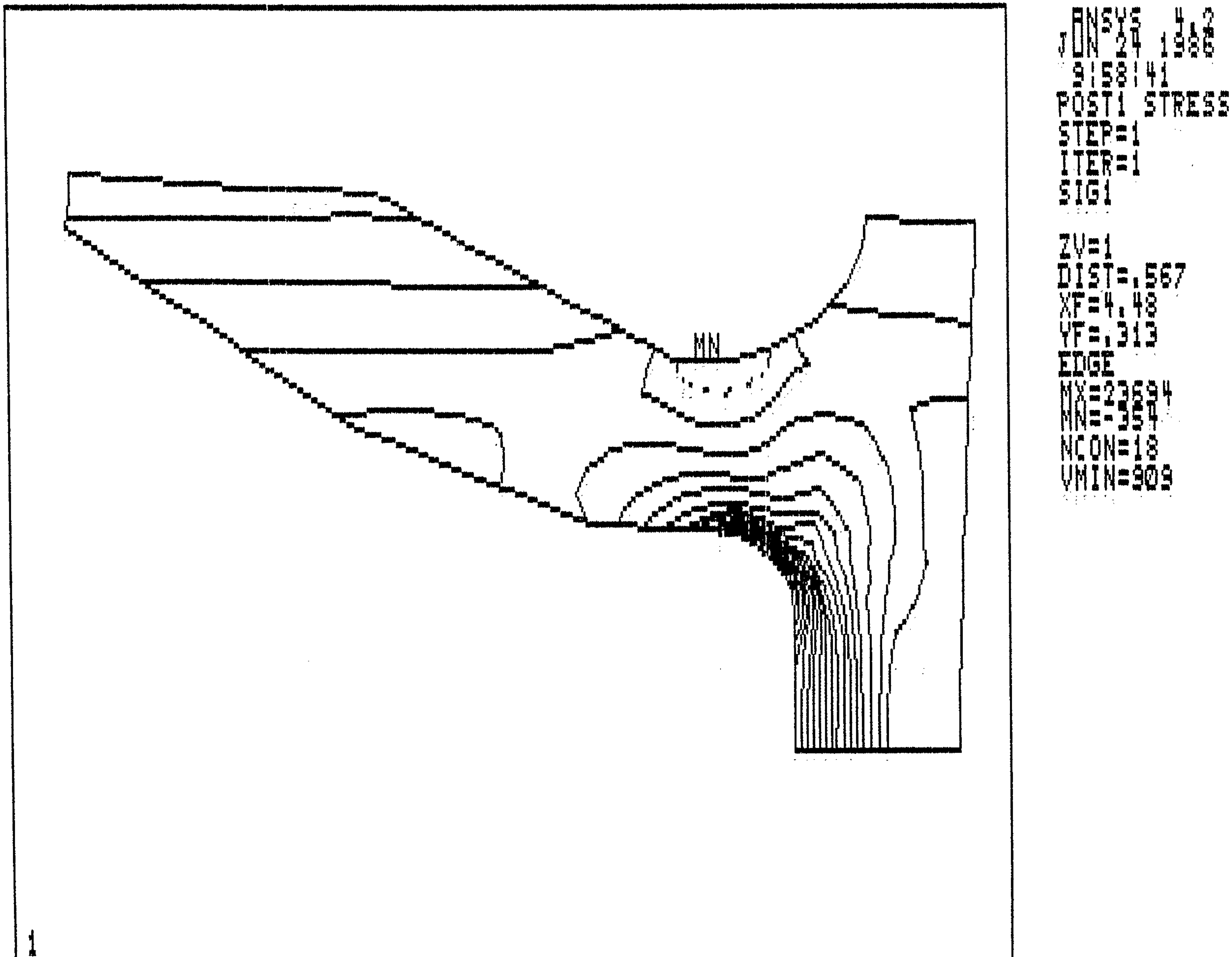
A.2.1 New Marman Node Plot
A.2.2 New Marman Element Plot
A.2.3 New Marman Load Step 1, Tensile Stress
A.2.4 New Marman Load Step 1, Tensile Stress Enlarged
A.2.5 New Marman Load Step 4, Tensile Stress Enlarged

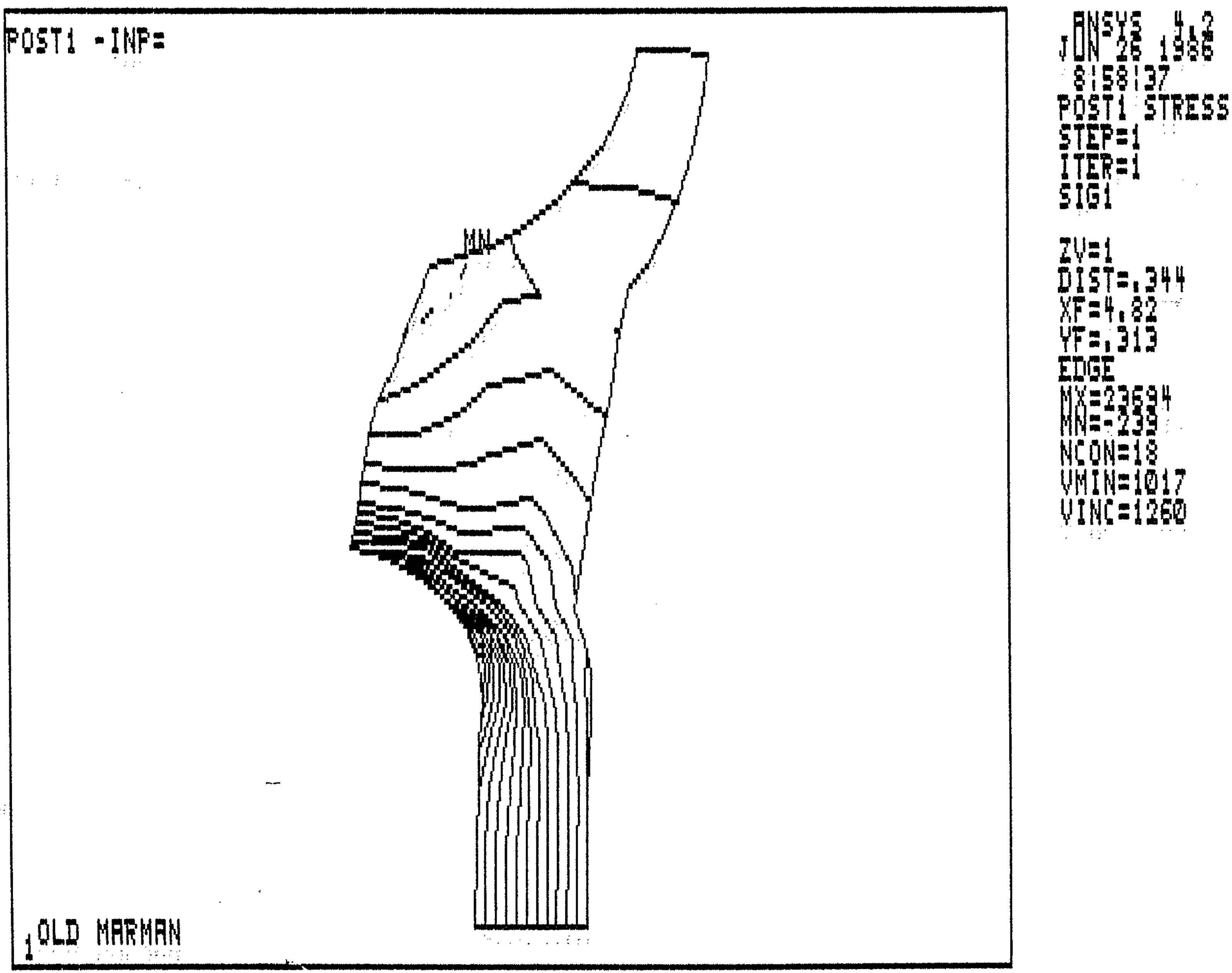
A.3.1 Connectors Ltd. Marman Node Plot
A.3.2 Connectors Ltd. Marman Element Plot
A.3.3 Connectors Ltd. Marman Load Step 1, Tensile Stress Enlarged
A.3.4 Connectors Ltd. Marman Load Step 1, Equivalent Stress
A.3.5 Connectors Ltd. Marman Load Step 4, Tensile Stress
A.3.6 Connectors Ltd. Marman Load Step 4, Equivalent Stress

A.4.1 Large O.D. New Marman Node Plot
A.4.2 Large O.D. New Marman Element Plot
A.4.3 Large O.D. New Marman Load Step 1, Tensile Stress Enlarged
A.4.4 Large O.D. New Marman Load Step 1, Equivalent Stress
A.4.5 Large O.D. New Marman Load Step 4, Tensile Stress
A.4.6 Large O.D. New Marman Load Step 4, Equivalent Stress

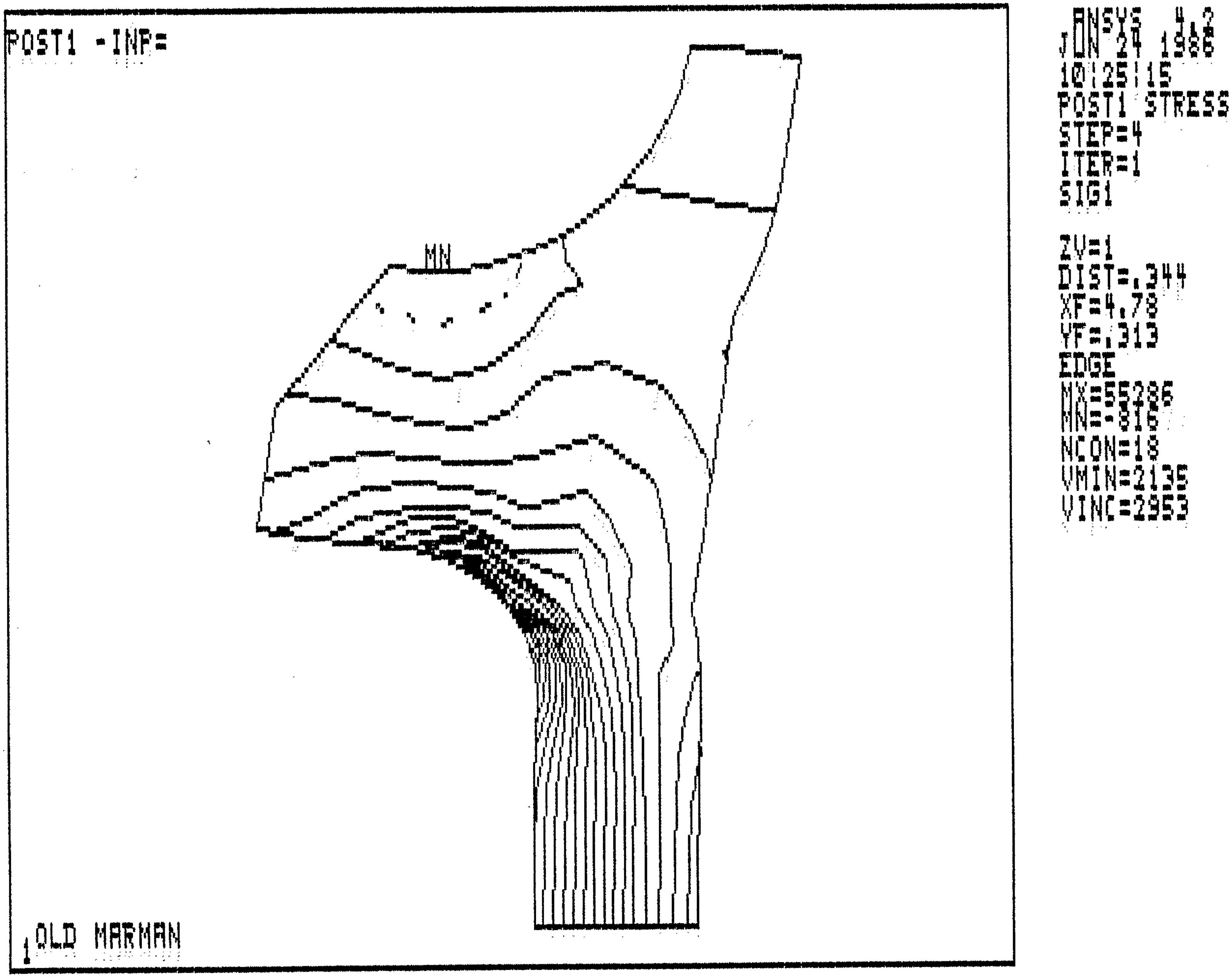
A.5 Connectors Marman Strap Stress Calculation

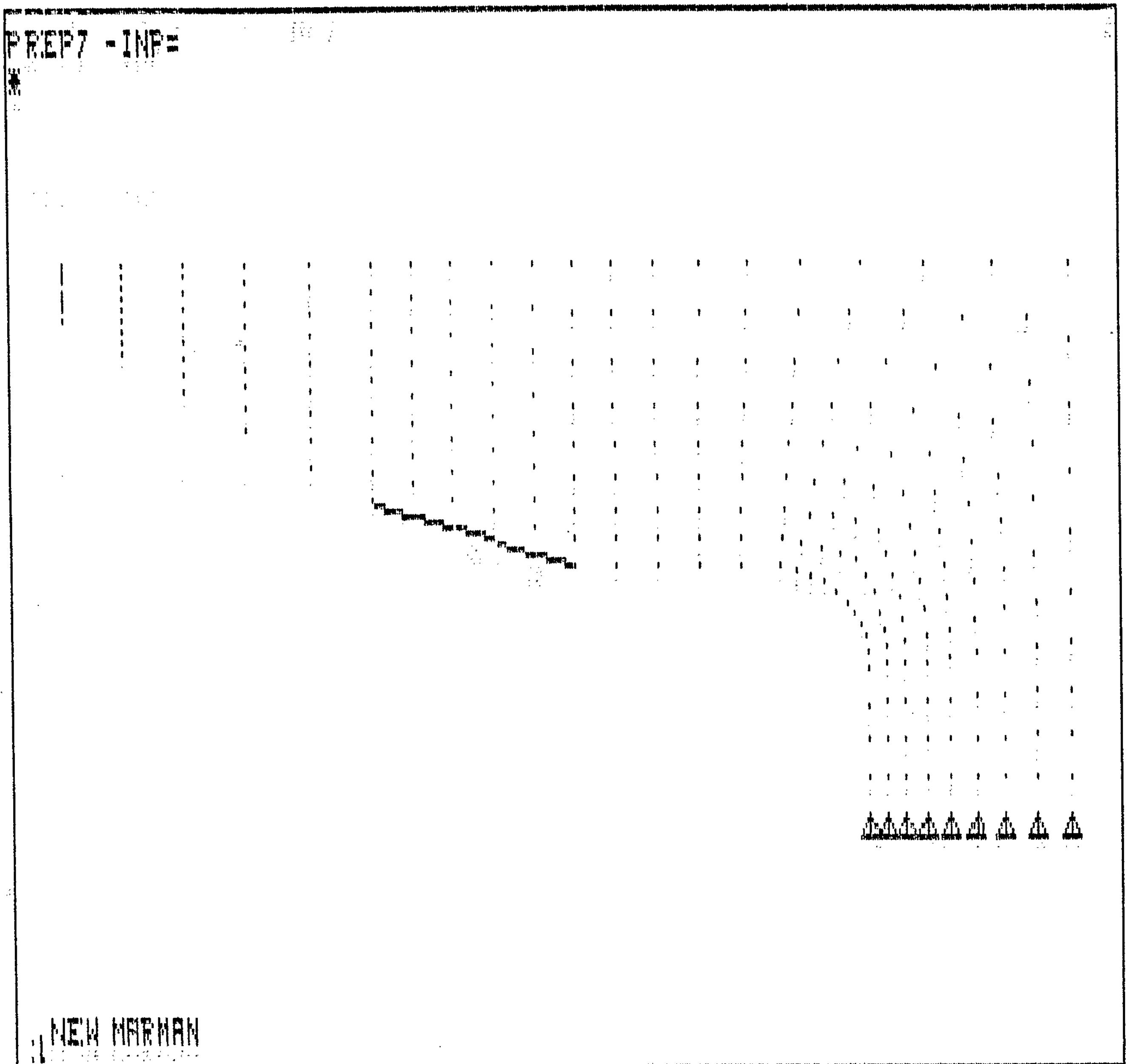

PREP7 -INP=
ANSYS 4.2
JUN 20 1986
10:22:05
PREP7 NODES
TDBC=1
PRBC=1
ZV=1
DIST=.567
XF=4.48
YF=.313
OLD MARMAN


MN
1
ANSYS 4.2
JUN 24 1986
9:58:41
POST1 STRESS
STEP=1
ITER=1
SIG1
ZV=1
DIST=.567
XF=4.48
YF=.313
EDGE
MX=23694
MN=-354
NCON=18
VMIN=909


POST1 -INP=
MN
1 OLD MARMAN
ANSYS 4.2
JUN 26 1986
8:58:37
POST1 STRESS
STEP=1
ITER=1
SIG1
ZV=1
DIST=.344
XF=4.82
YF=.313
EDGE
MX=23694
MN=-239
NCON=18
VMIN=1017
VINC=1260

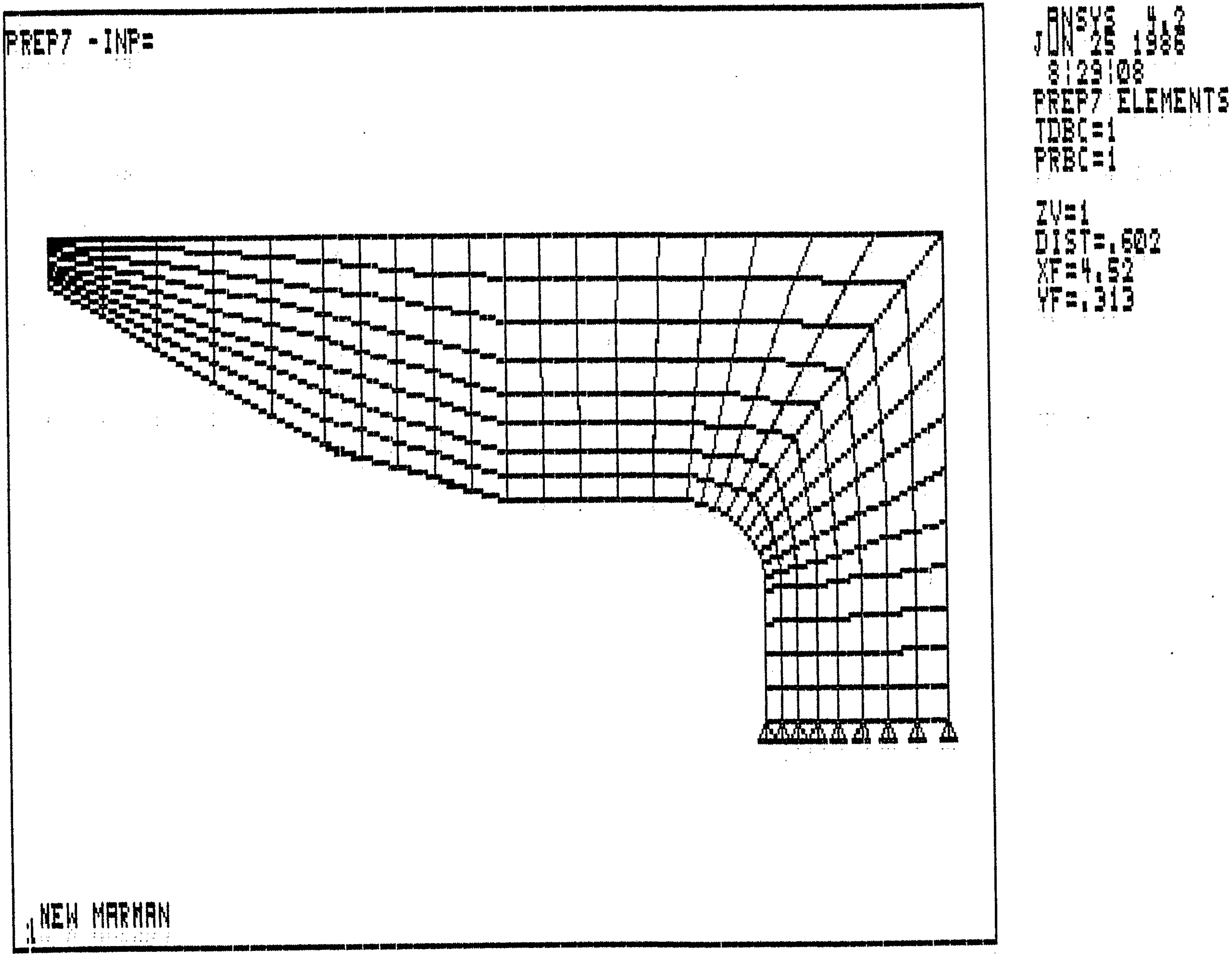
A.1.4

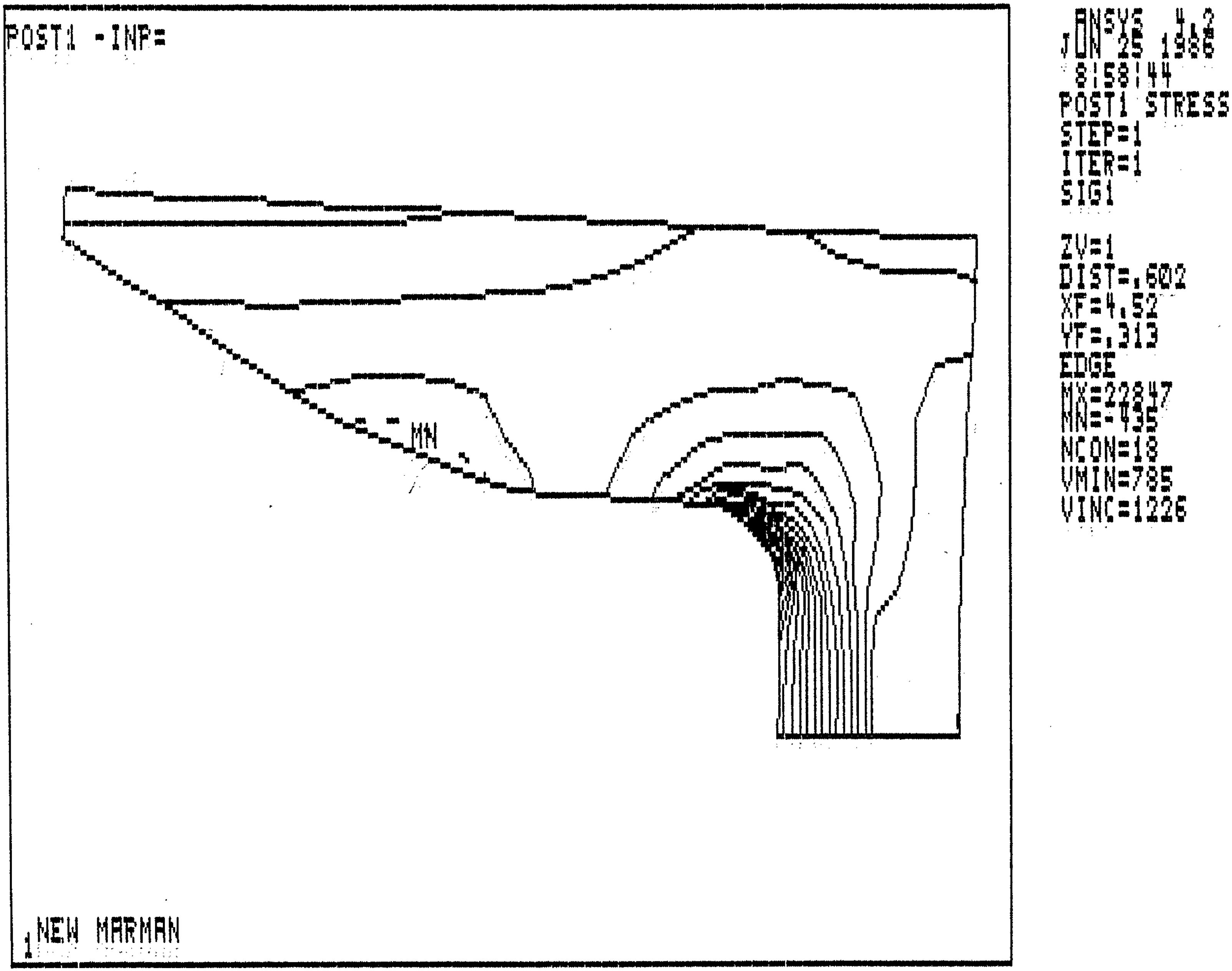

POST1 -INP=
MN
1 OLD MARMAN
ANSYS 4.2
JUN 24 1986
10:25:15
POST1 STRESS
STEP=4
ITER=1
SIG1
ZV=1
DIST=.344
XF=4.78
YF=.313
EDGE
MX=55286
MN=-816
NCON=13
VMIN=2135
VINC=2953

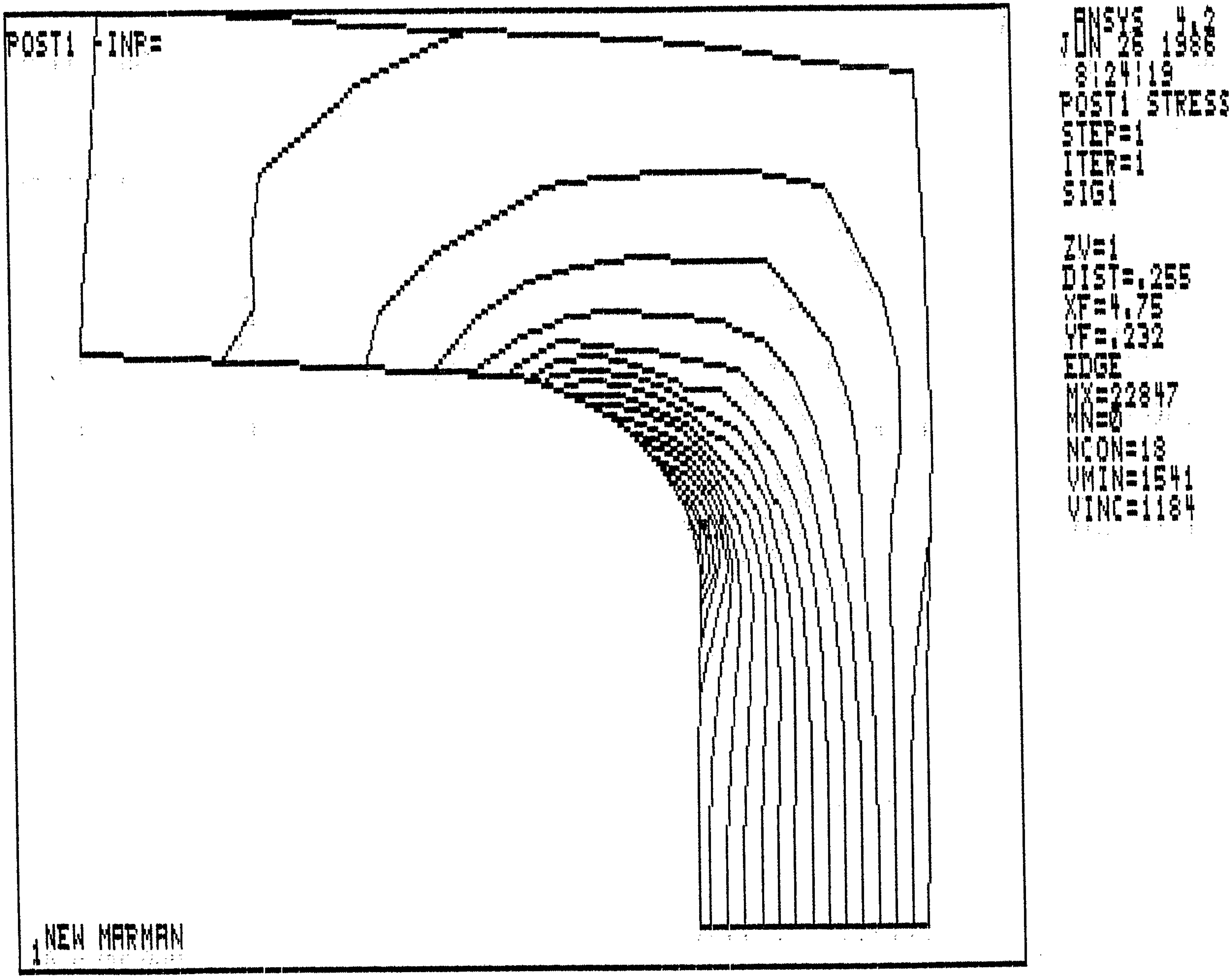


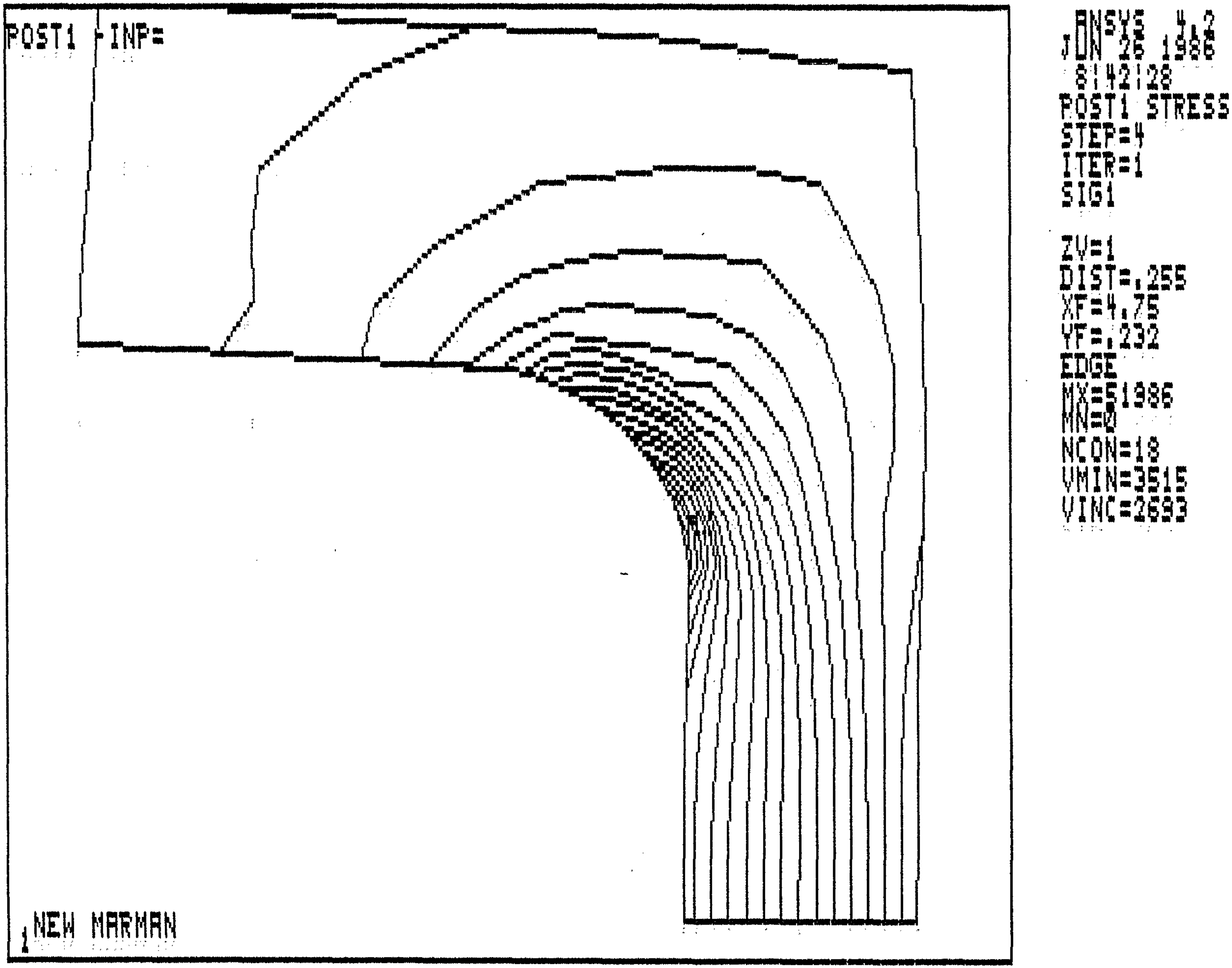
ANSYS 4.2
JUN 24 1986
13:22:16
PREP7 NODES
TDBC=1
FRBC=1

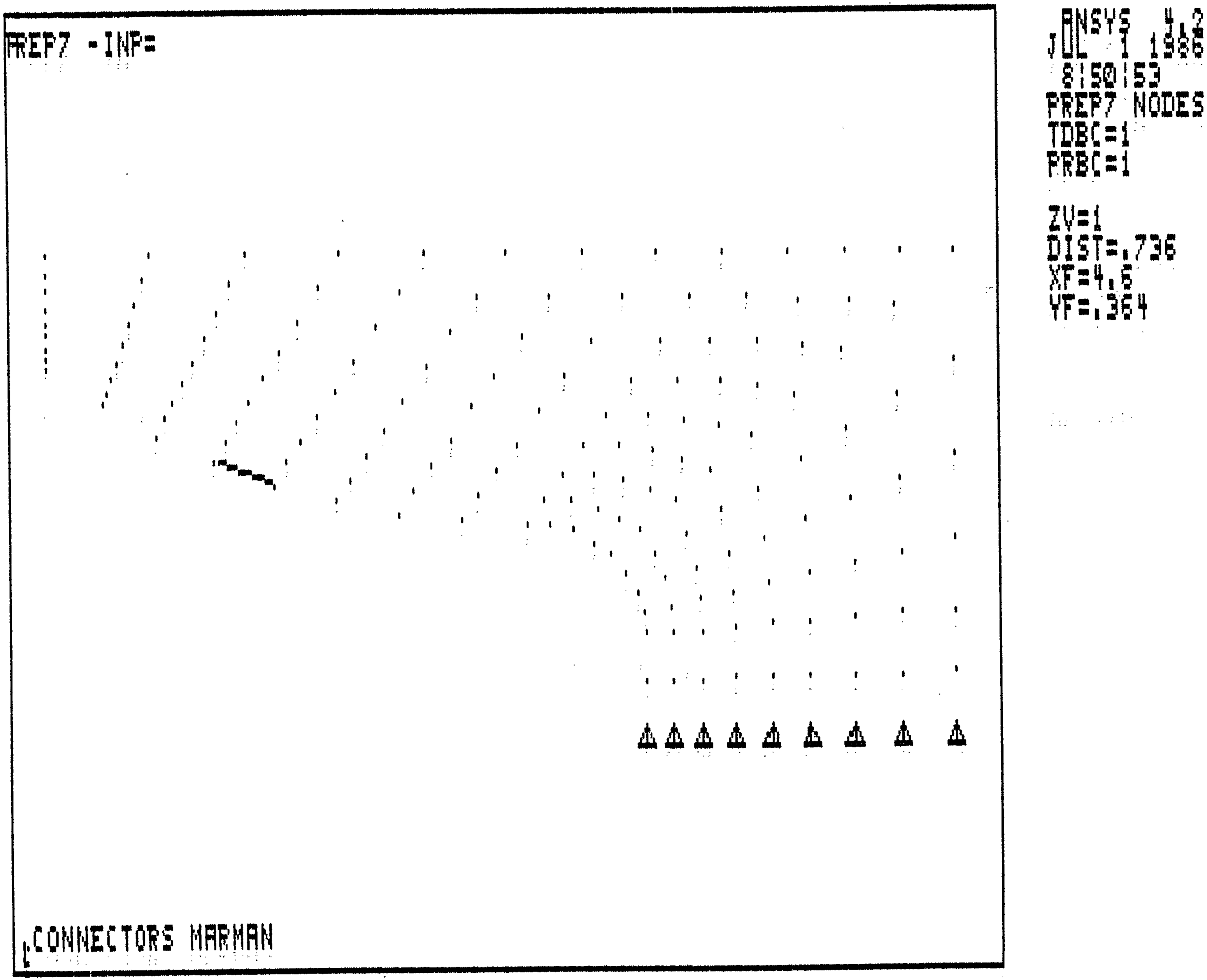
ZV=1
DIST=.602
XF=4.52
YF=.313

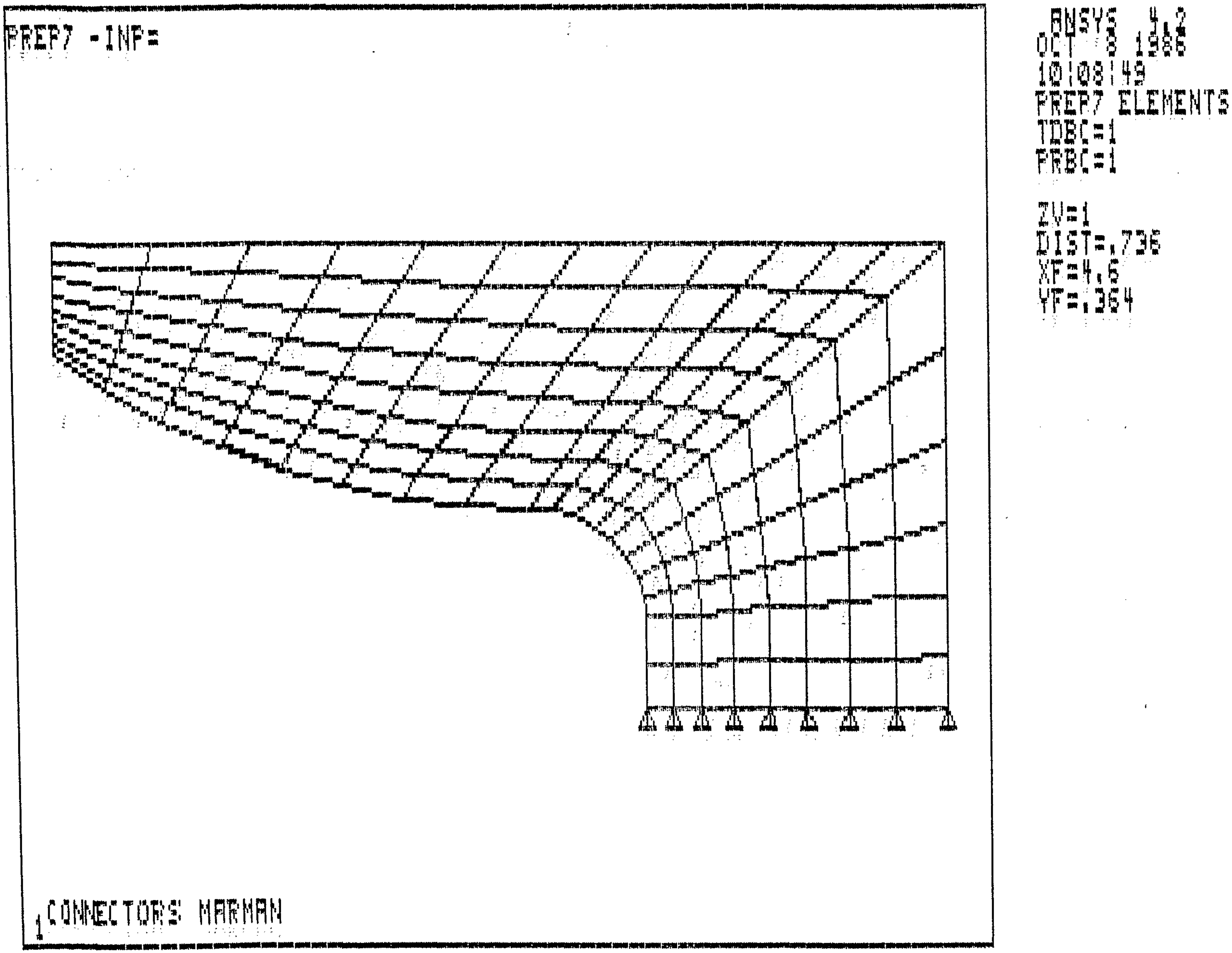

PREP7 -INP=
NEW MARMAN
ANSYS 4.2
JUN 25 1986
8:29:08
PREP7 ELEMENTS
TDBC=1
PRBC=1
ZV=1
DIST=.602
XF=4.52
YF=.313


POST1 -INP=
MN
1 NEW MARMAN
ANSYS 4.2
JUN 25 1986
8:58:44
POST1 STRESS
STEP=1
ITER=1
SIG1
ZV=1
DIST=.602
XF=4.52
YF=.313
EDGE
MX=22847
MN=-435
NCON=13
VMIN=785
VINC=1226


POST1 INP=
1 NEW MARMAN
ANSYS 4.2
JUN 26 1986
8:24:19
POST1 STRESS
STEP=1
ITER=1
SIG1
ZV=1
DIST=.255
XF=4.75
YF=.232
EDGE
MX=22847
MN=0
NCON=18
VMIN=1541
VINC=1184

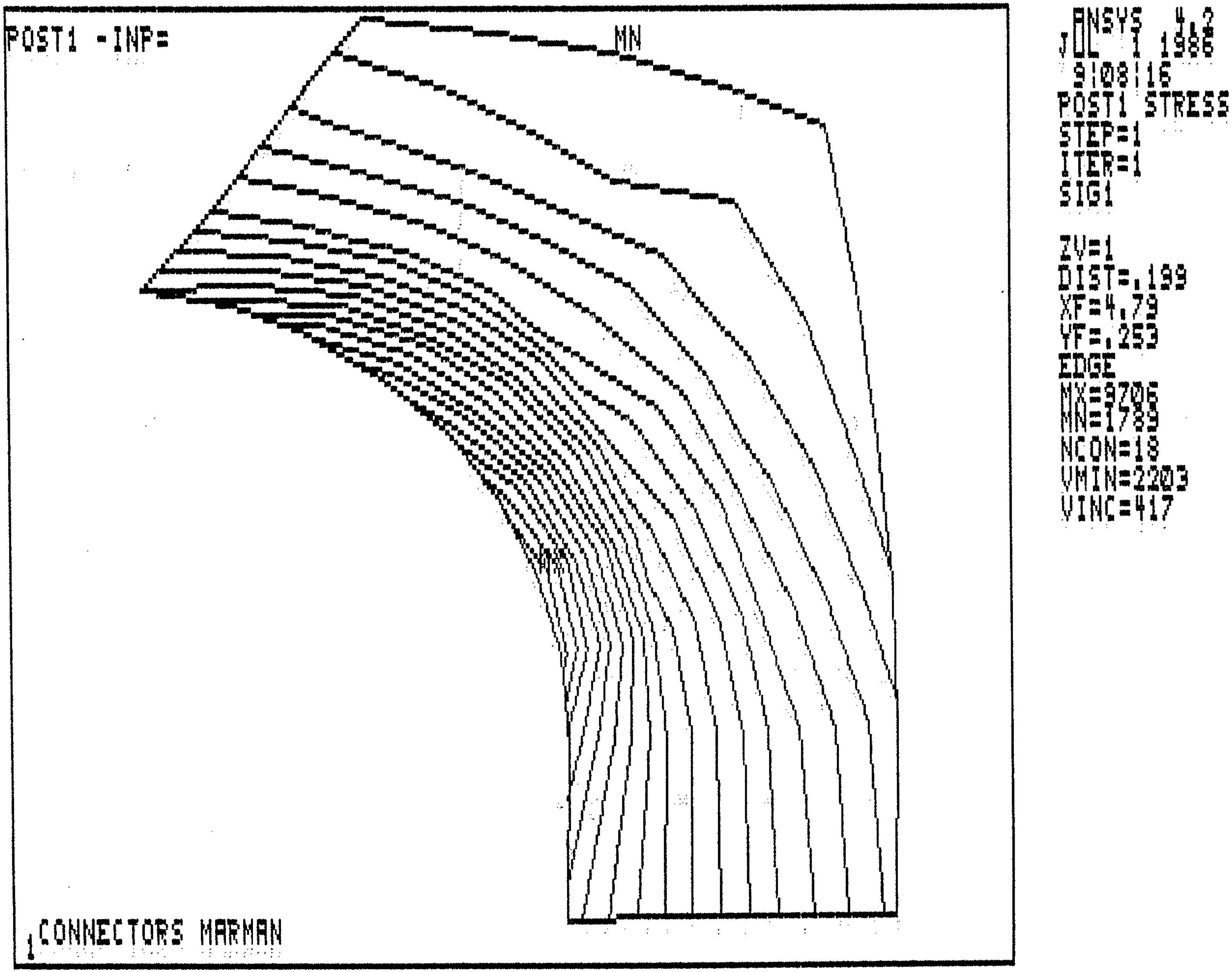

POST1 INP=
ANSYS 4.2
JUN 26 1986
8:42:28
POST1 STRESS
STEP=4
ITER=1
SIG1
ZV=1
DIST=.255
XF=4.75
YF=.232
EDGE
MX=51986
MN=0
NCON=18
VMIN=3515
VINC=2693
1 NEW MARMAN


PREP7 -INP=
ANSYS 4.2
JUL 1 1986
8:50:53
PREP7 NODES
TDBC=1
FRBC=1
ZV=1
DIST=.736
XF=4.6
YF=.364
CONNECTORS MARMAN

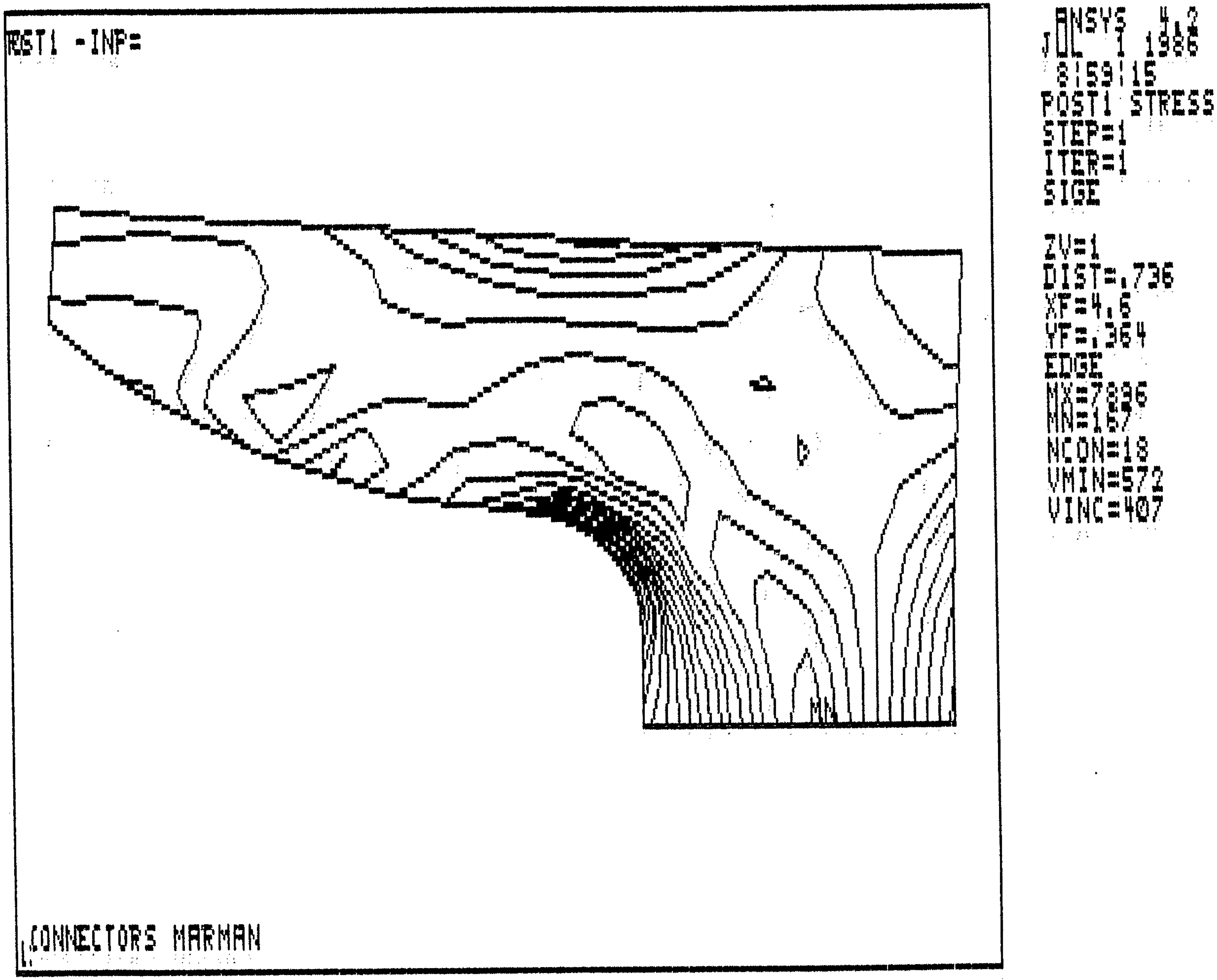


FREP7 -INF=
ANSYS 4.2
OCT 8 1986
10:08:49
FREP7 ELEMENTS
TDBC=1
FRBC=1
ZV=1
DIST=.736
XF=4.6
YF=.364
1 CONNECTORS MARMAN

A.3.3

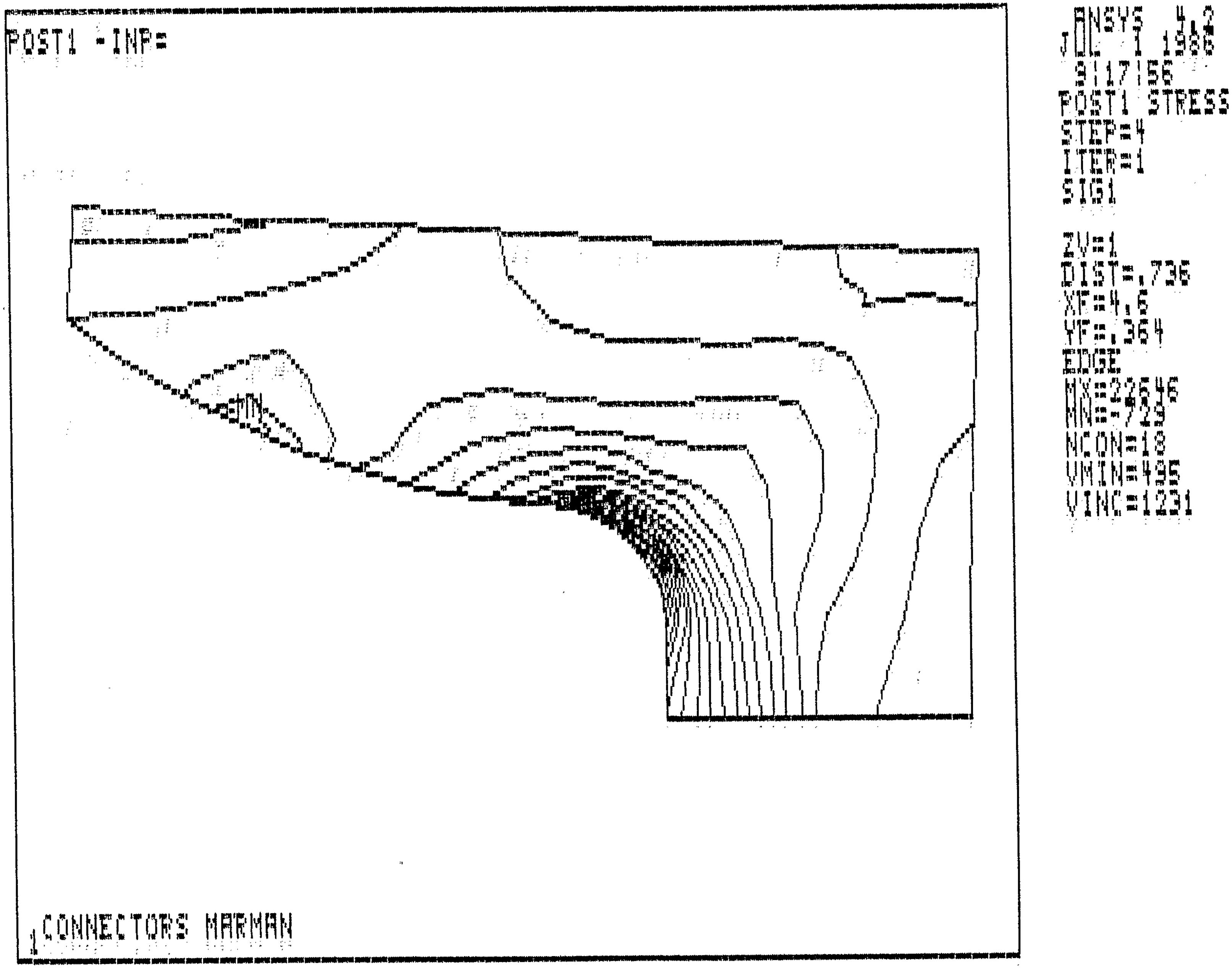

POST1 -INP=
MN
MX
1 CONNECTORS MARMAN
ANSYS 4.2
JUL 1 1986
9:08:16
POST1 STRESS
STEP=1
ITER=1
SIG1
ZV=1
DIST=.199
XF=4.79
YF=.253
EDGE
MX=9706
MN=1789
NCON=18
VMIN=2203
VINC=417

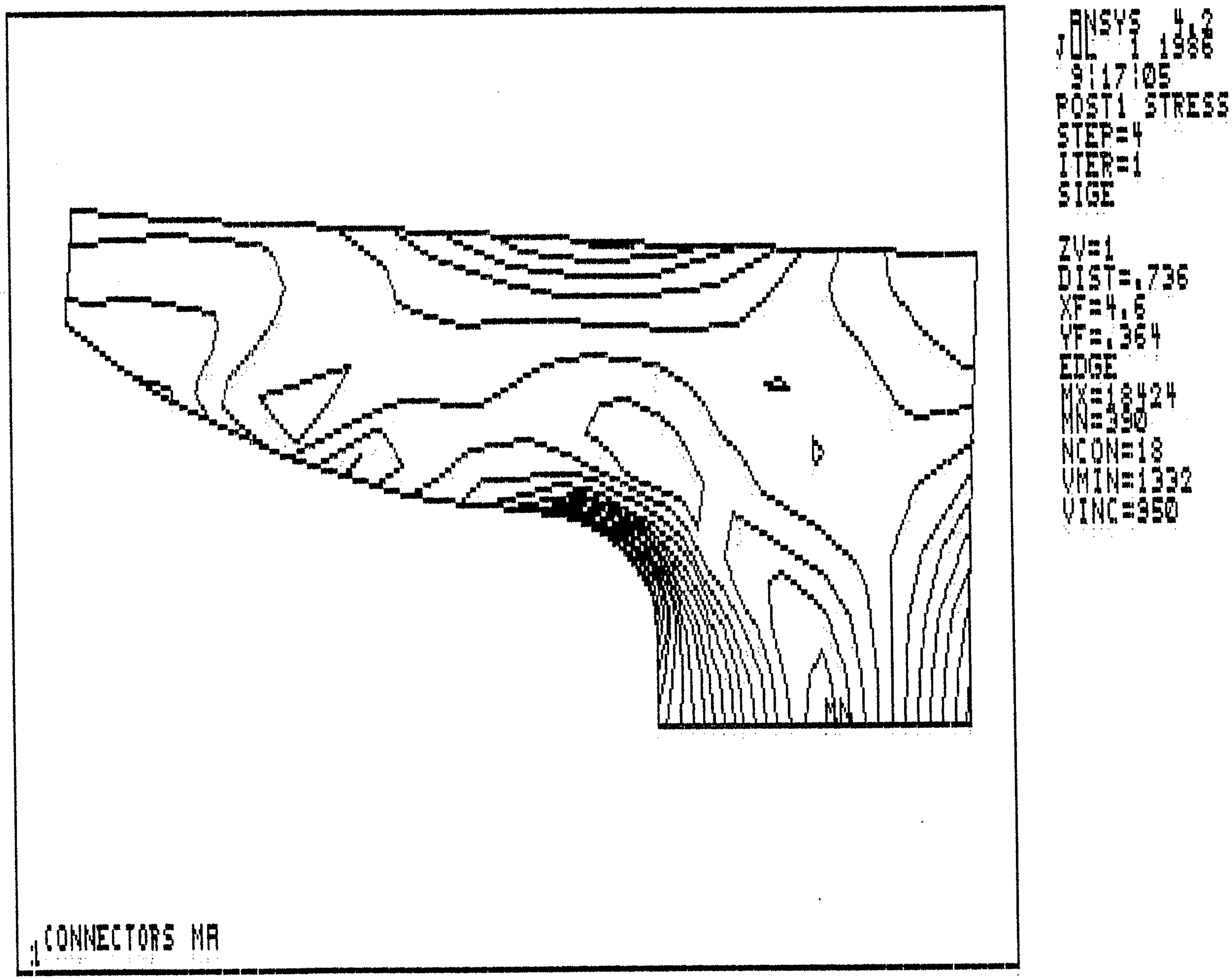
A.3.4



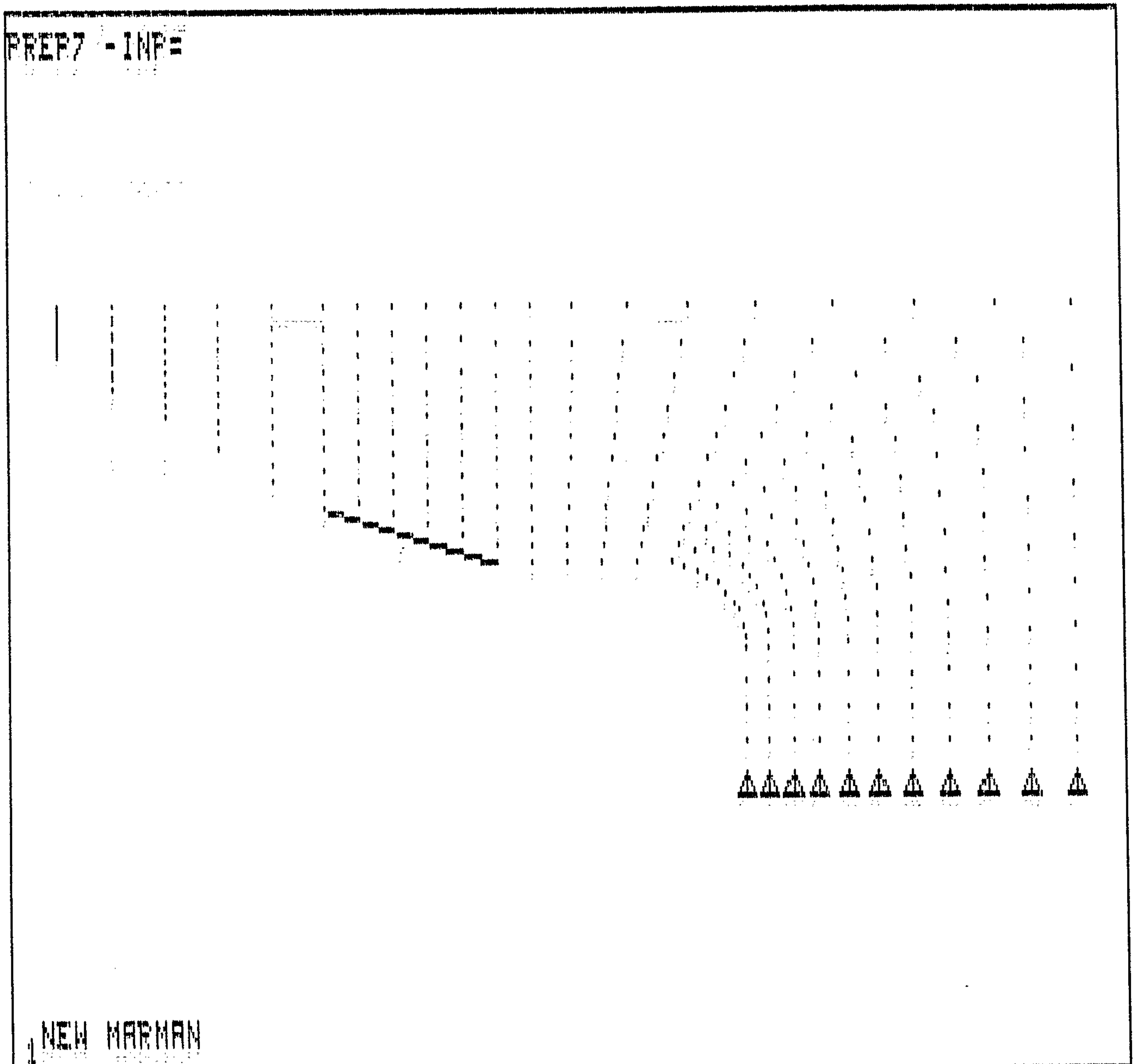
RST1 -INP=
ANSYS 4.2
JUL 1 1986
8:59:15
POST1 STRESS
STEP=1
ITER=1
SIGE
ZV=1
DIST=.736
XF=4.6
YF=.364
EDGE
MX=7836
MN=167
NCON=18
VMIN=572
VINC=407
MN
CONNECTORS MARMAN

A.3.5


POST1 -INP=
MN
1
CONNECTORS MARMAN
ANSYS 4.2
JUL 1 1986
9:17:56
POST1 STRESS
STEP=4
ITER=1
SIG1
ZV=1
DIST=.736
XF=4.6
YF=.364
EDGE
MX=22546
MN=-729
NCON=18
VMIN=495
VINC=1231

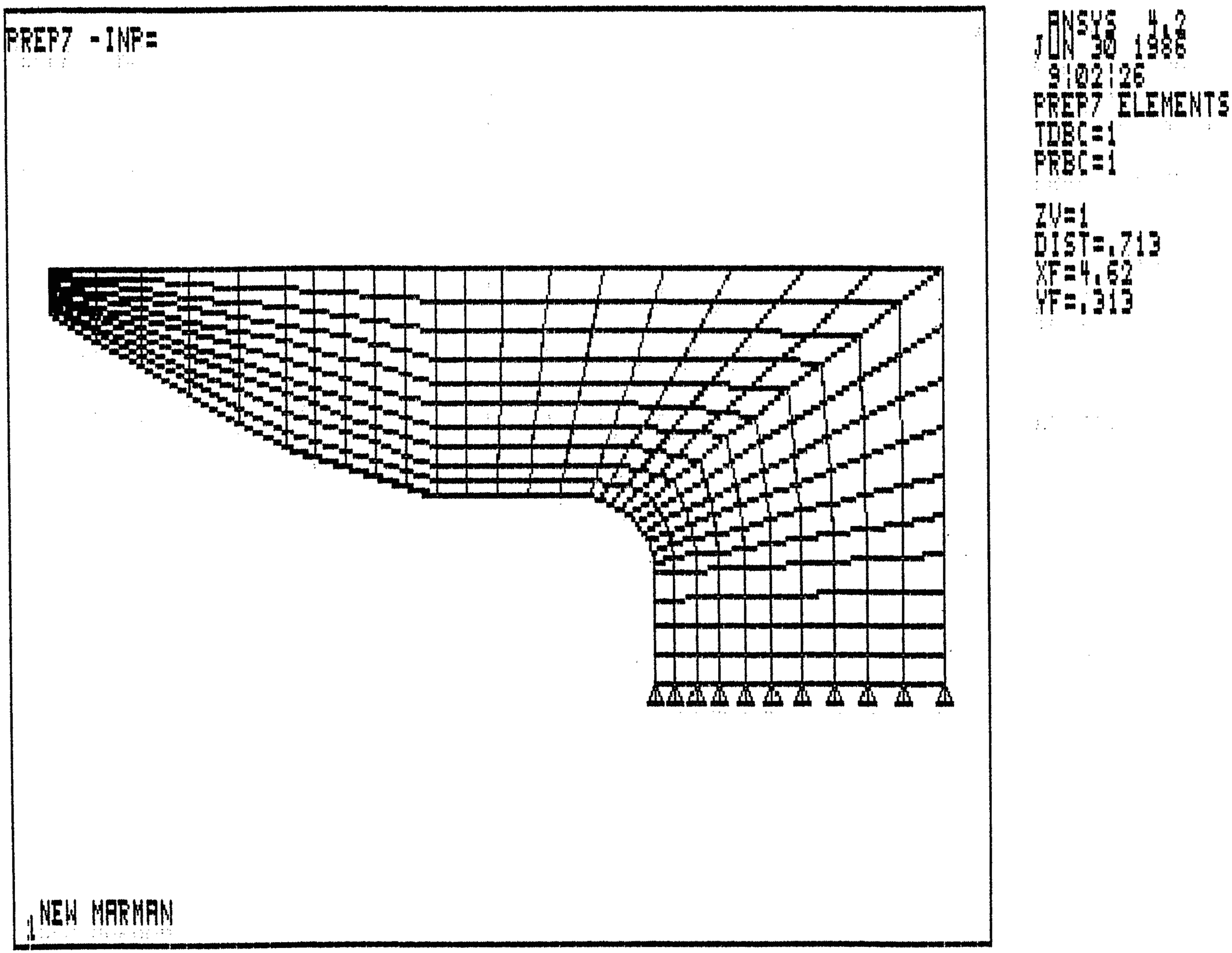

ANSYS 4.2
JUL 1 1986
9:17:05
POST1 STRESS
STEP=4
ITER=1
SIGE
ZV=1
DIST=.736
XF=4.6
YF=.364
EDGE
MX=18424
MN=390
NCON=18
VMIN=1332
VINC=950
MN
1
CONNECTORS MA

A.4.1

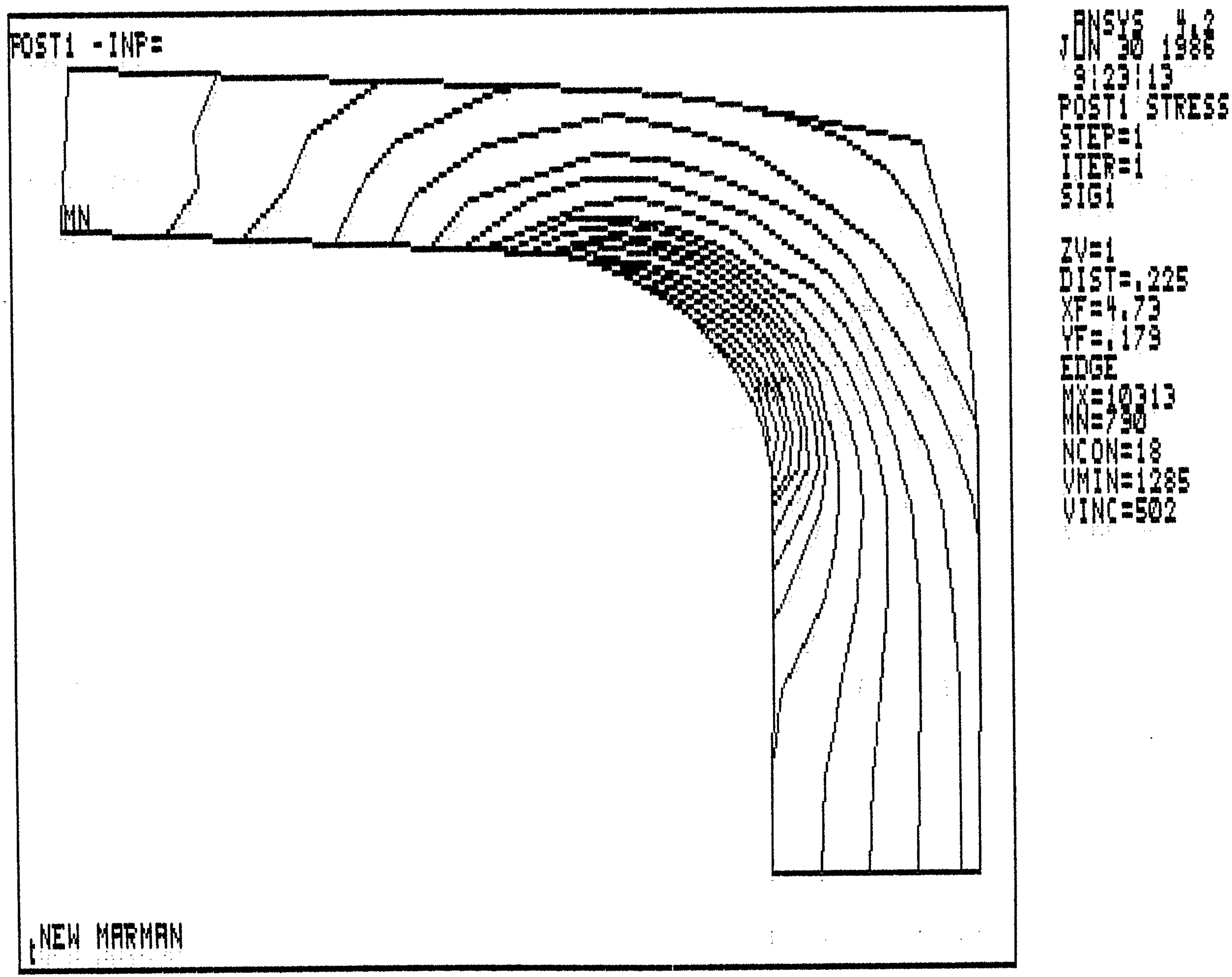


ANSYS 4.2
JUN 30 1986
9:03:15
PREP7 NODES
TDBC=1
PRBC=1

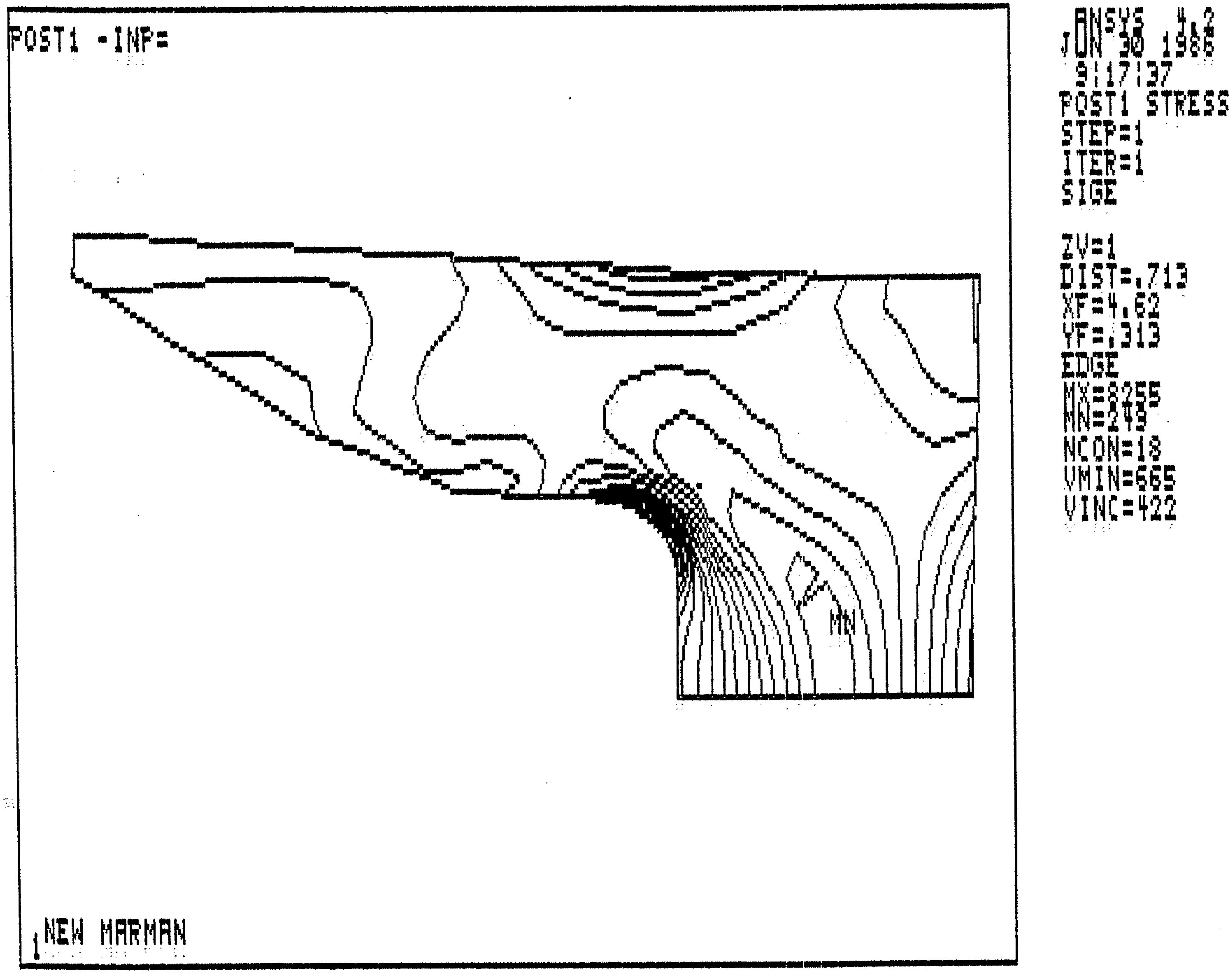
ZV=1
DIST=.713
XF=4.62
YF=.313

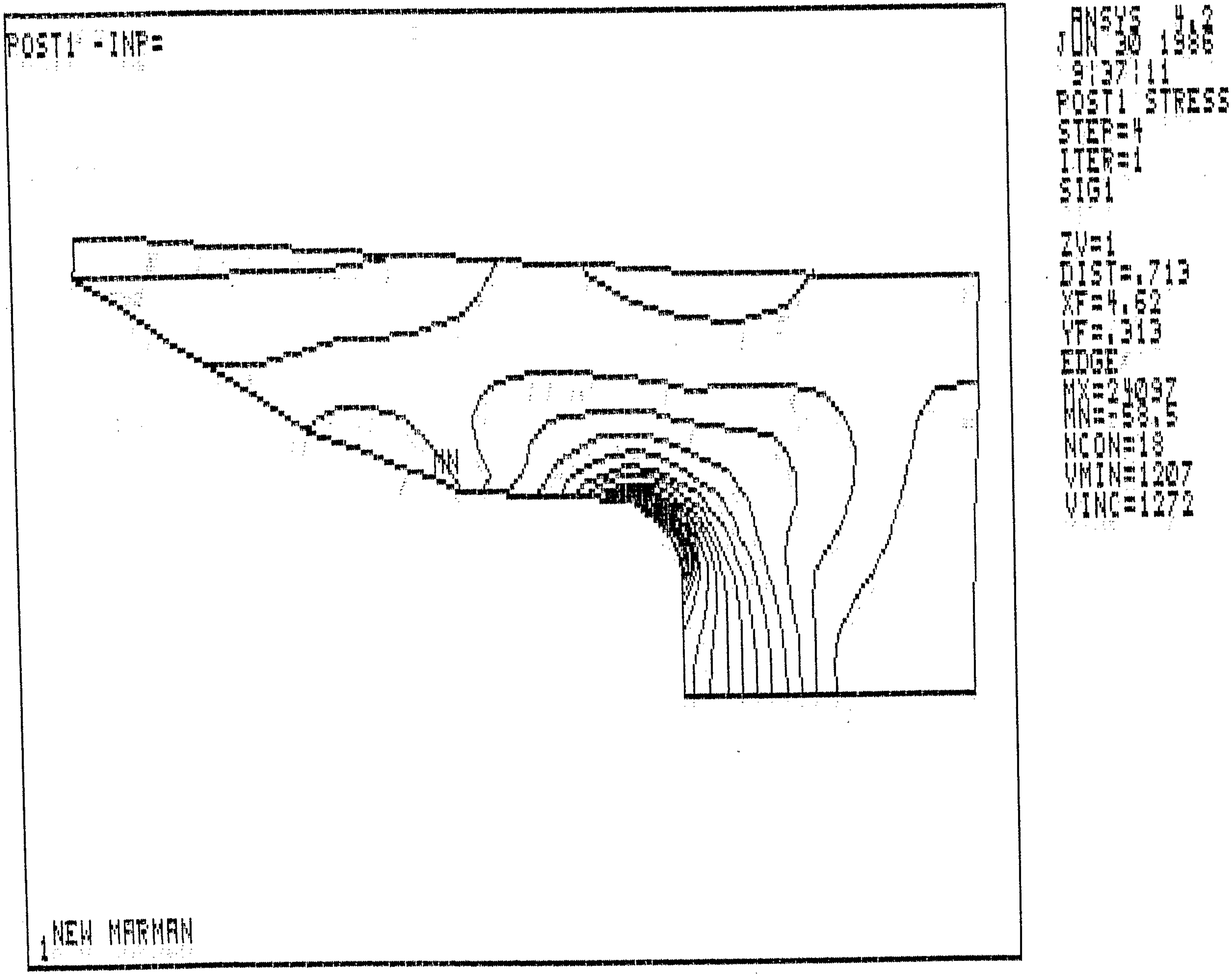

PREP7 -INP=
ANSYS 4.2
JUN 30 1986
9:02:26
PREP7 ELEMENTS
TDBC=1
PRBC=1
ZV=1
DIST=.713
XF=4.62
YF=.313
1 NEW MARMAN

A.4.3

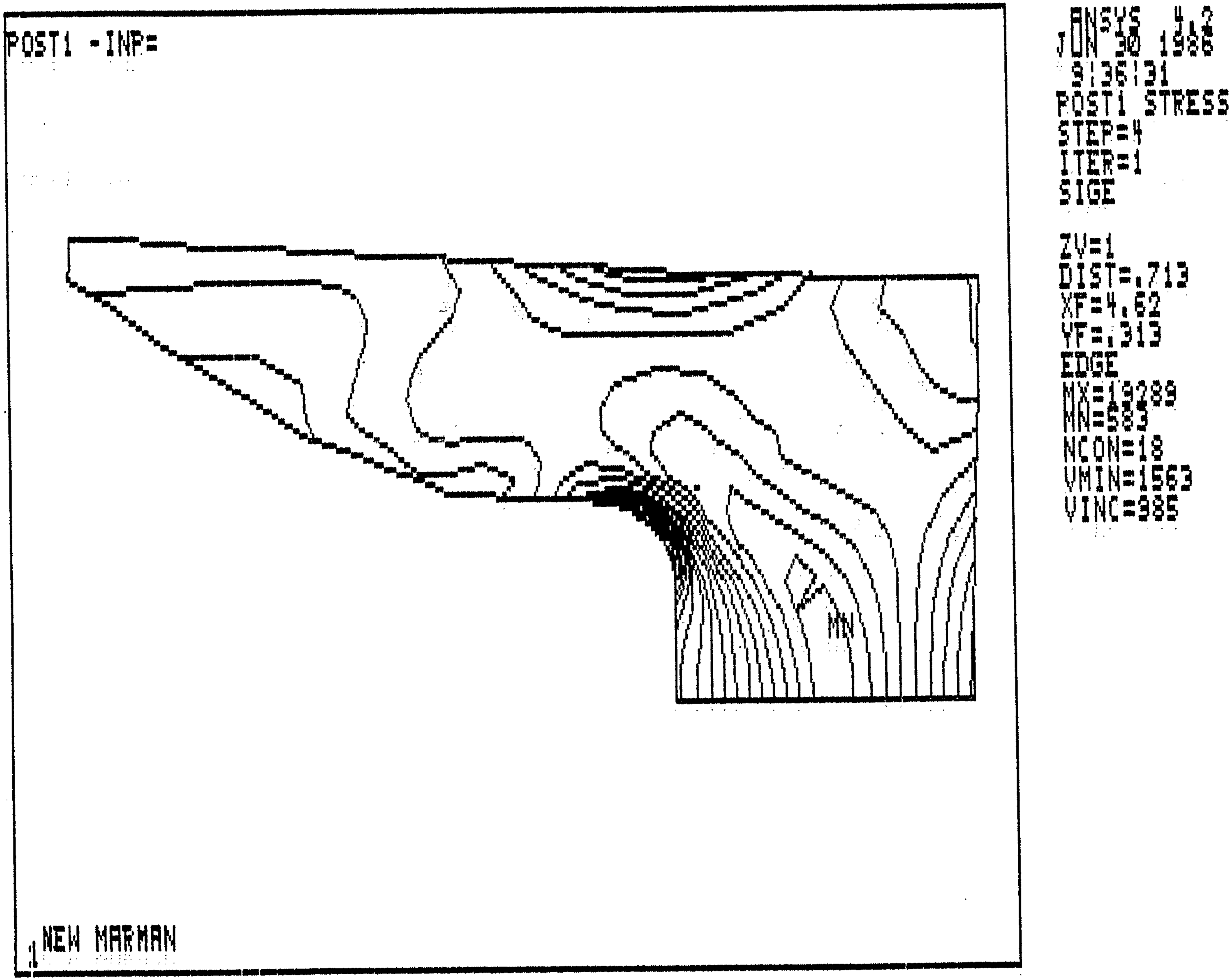

POST1 -INP=
MN
NEW MARMAN
ANSYS 4.2
JUN 30 1986
9:23:13
POST1 STRESS
STEP=1
ITER=1
SIG1
ZV=1
DIST=.225
XF=4.73
YF=.179
EDGE
MX=10313
MN=790
NCON=18
VMIN=1285
VINC=502

A.4.4


POST1 -INP=
MN
1 NEW MARMAN
ANSYS 4.2
JUN 30 1986
9:17:37
POST1 STRESS
STEP=1
ITER=1
SIGE
ZV=1
DIST=.713
XF=4.62
YF=.313
EDGE
MX=8255
MN=249
NCON=18
VMIN=665
VINC=422


POST1 -INP=
MN
1 NEW MARMAN
ANSYS 4.2
JUN 30 1986
9:37:11
POST1 STRESS
STEP=4
ITER=1
SIG1
ZV=1
DIST=.713
XF=4.62
YF=.313
EDGE
MX=24097
MN=-58.5
NCON=18
VMIN=1207
VINC=1272

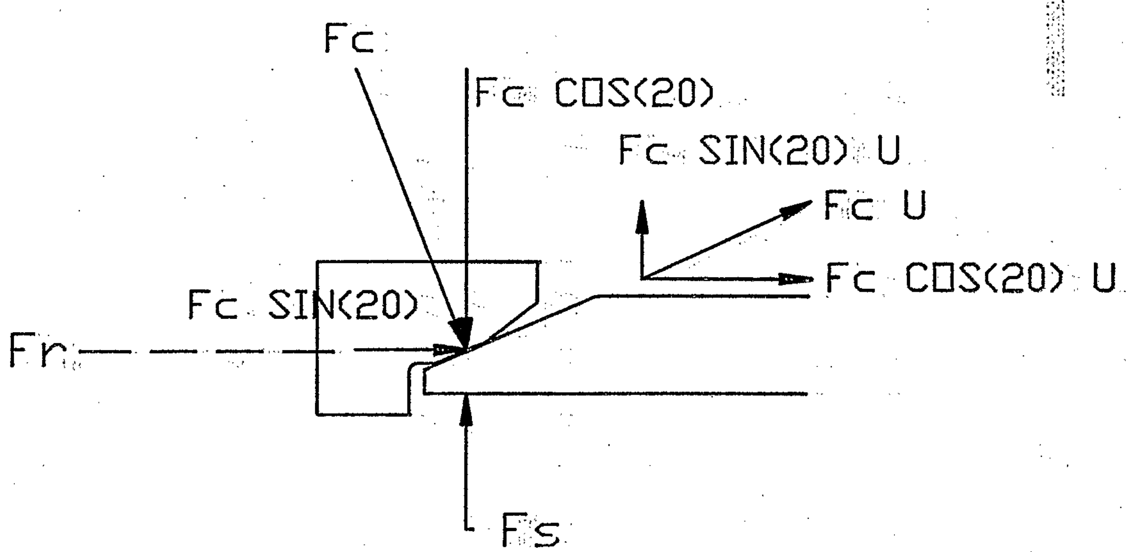
A.4.5


POST1 -INP=
ANSYS 4.2
JUN 30 1986
9:36:31
POST1 STRESS
STEP=4
ITER=1
SIGE
ZV=1
DIST=.713
XF=4.62
YF=.313
EDGE
NCON=18
VMIN=1563
VINC=985
MN
1
NEW MARMAN

A.4.6

STRESSES ON THE CLAMP STRAP

THE CLAMP ACTS AS A WEDGE ON THE FLANGE TRANSFERING RADIAL FORCE INTO SEALING FORCE ON THE FLANGE. A FORCE DIAGRAM FOR HALF OF THE SYSTEM IS SHOWN BELOW WITH SYMMETRY BEING MAINTAINED ALONG THE CENTERLINE AS IN THE COMPUTER MODEL



$$Fr \text{ (RADIAL FORCE)} = Fc\ SIN(20) + Fc\ COS(20)\ U$$

WHERE U = THE FRICTION FACTOR AT THE CONTACT POINT AND 20 DEG IS THE CLAMP ANGLE

$$Fs \text{ (SEALING FORCE)} = Fc\ COS(20) - Fc\ SIN(20)\ U$$

COMBINING THE TWO EQUATIONS GIVES:

$$Fr = Fs\ \frac{TAN(20) + U}{1 - TAN(20)\ U}$$

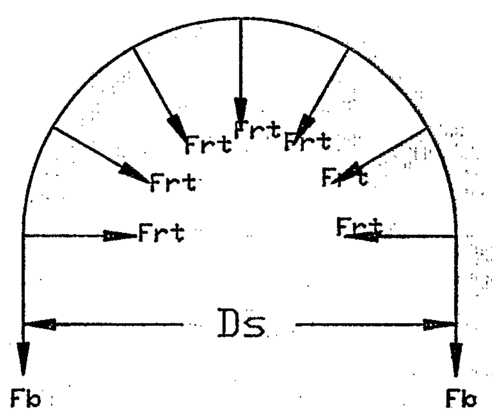
THE TOTAL FORCE EXERTED BY THE CLAMP IS THE SUM OF BOTH HALVES ($Frt = 2Fr$). TAKING U = .10 FOR THE DRY LUBRICANT USED ON THE CLAMPS GIVES:

$$Frt = .96\ Fs$$

TAKING THE STRAP AS A CYLINDRICAL ELEMENT WITH AN INTERNAL PRESSURE EQUIVALENT TO Frt WITH UNITS OF FORCE PER UNIT LENGTH AT THE SEAL DIAMETER (Ds). THE TENSILE FORCE ON THE STRAP; WHICH, EQUALS THE CLAMP BOLTING FORCE; CAN BE FOUND:

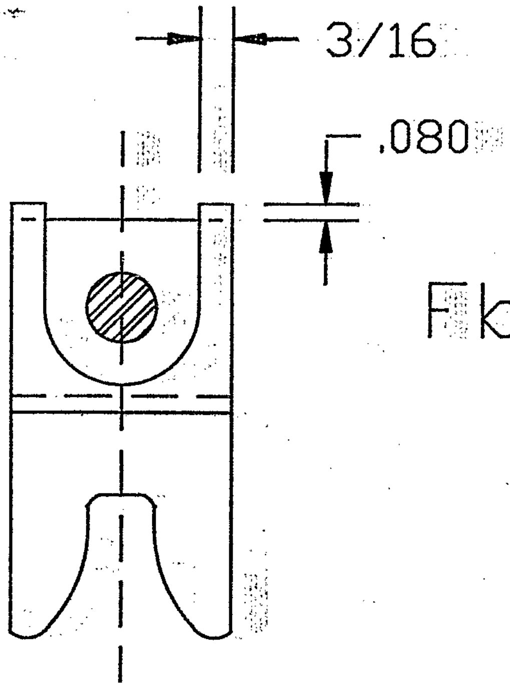
$$Fb = (Frt\ Ds)/2$$

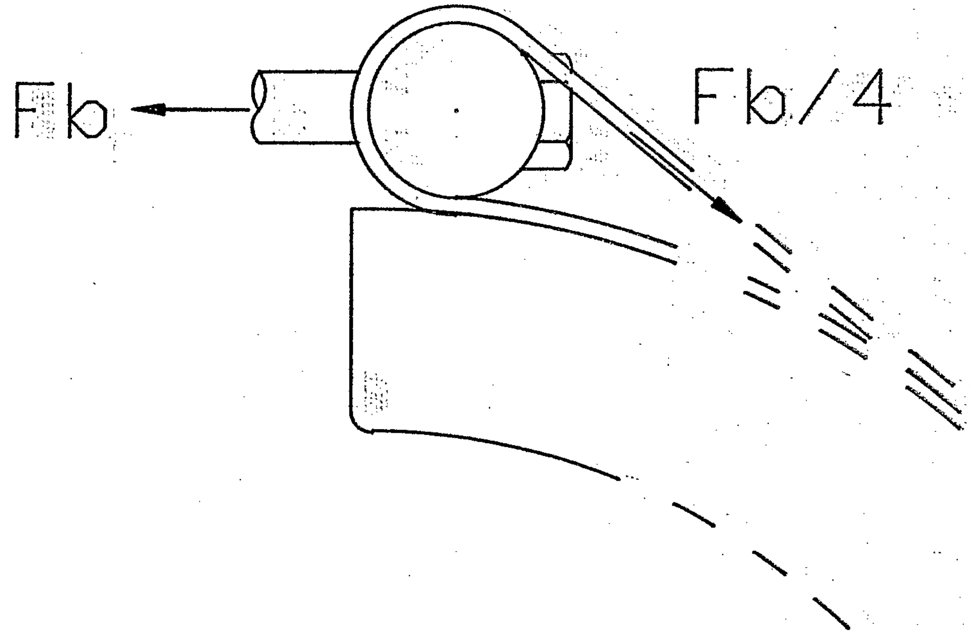
$$Fb = .48\ Ds\ Fs$$



IN THE EQUATION Fb HAS UNITS OF FORCE (LBS), Fs UNITS OF FORCE PER UNIT LENGTH (LBS/IN), AND Ds LENGTH (IN). FOR OUR SEAL Ds = 8.5 IN, THEREFORE:

$$Fb = 4.09\ Fs$$





The bolting force is transferred to the straps via cylindrical trunnions on the top and bottom of the clamp. The trunnion moves freely in the strap which splits the force evenly between the top and bottom. Where the strap passes over the top of the trunnion, it is opened to allow movement of the bolt and room for a socket wrench. This splits the force again between the strap segments on either side of the bolt. This results in a force equal to Fb/4 on a strap cross section of only .080 in. x 1/4. Taking this area as being in tensile stress from the bolt force only, gives the stresses listed below:

SEALING FORCE (LBS./IN.)	BOLT FORCE (LBS.)	STRAP STRESS (PSI)	BOLT TORQUE* (IN. LBS.)
300	1230	15,400	100
450	1840	23,000	150
600	2455	30,700	200
700	2865	35,800	220
840	3435	44,000	260

*Torque found through T = KDF, where D = 10 mm for the Connectors Ltd. bolt, F = the bolt force and K = torque friction coefficient taken as .19 from Connectors Ltd. data.