BOSOR5 PACKET

BOSOR4 & BOSOR 5

WARNING !!

Thé purpose of pp. ~~8-11~~ 2-5 is to provide a stern warning to avoid using the general geometry (NSHAPE = 4) branch whenever possible. Results are always better, and sometimes a great deal better (as in the example on pp. 2-5 ~~8-11~~), if you divide a general meridional shape into separate segments, each of which is of type NSHAPE = 1 (straight meridian) or NSHAPE = 2 (meridian of constant meridional curvature). Figure 1 shows a pressurized test specimen; Fig. 2 shows the meridian modeled with NSHAPE = 4 (general geometry for which z,r pairs are the input); Fig. 3 shows the meridian modeled with use of 4 segments, each of which is either of type NSHAPE = 1 or NSHAPE = 2; and Fig. 4 shows comparisons between test and theory. Note especially the poor predictions yielded by the model in which NSHAPE = 4 is used: the analysis predicts far too much stiffness. It is not known for sure what causes this behavior. It seems ok to introduce small sinusoidal imperfections, for example, which leads to continuously changing meridional curvature, but it is clearly not possible to represent large changes in meridional curvature within a single shell segment.

the last Page ~~17~~ of this packet ~~attachment~~ gives my ~~new~~ address and telephone number. Please let me know if you have a new BOSOR "caretaker", so that I can update my address list.

Lockheed
Missiles & Space Company, Inc

David Bushnell
Sr. Staff Scientist
Applied Mechanics Laboratory

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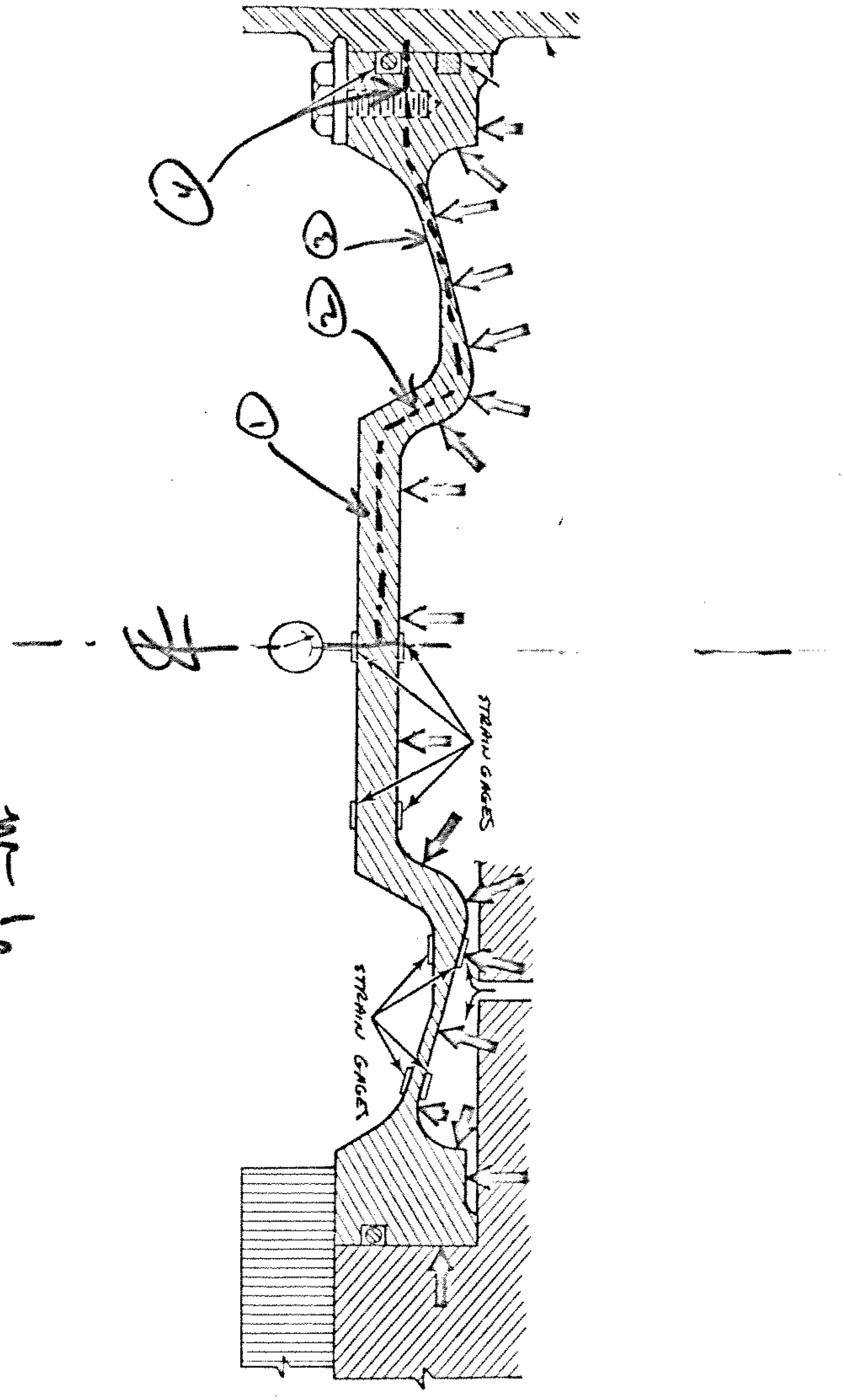
(415) ~~858-4037~~ O/~~52-33~~ B/~~255~~ 251
424-3237 / 93-30



BOSOR4 &
BOSOR5

Fig. 1 Test Specimen

ALWT BULKHEAD WITH HIGH MODU
INITIAL UNDEFORMED STRUCTURE

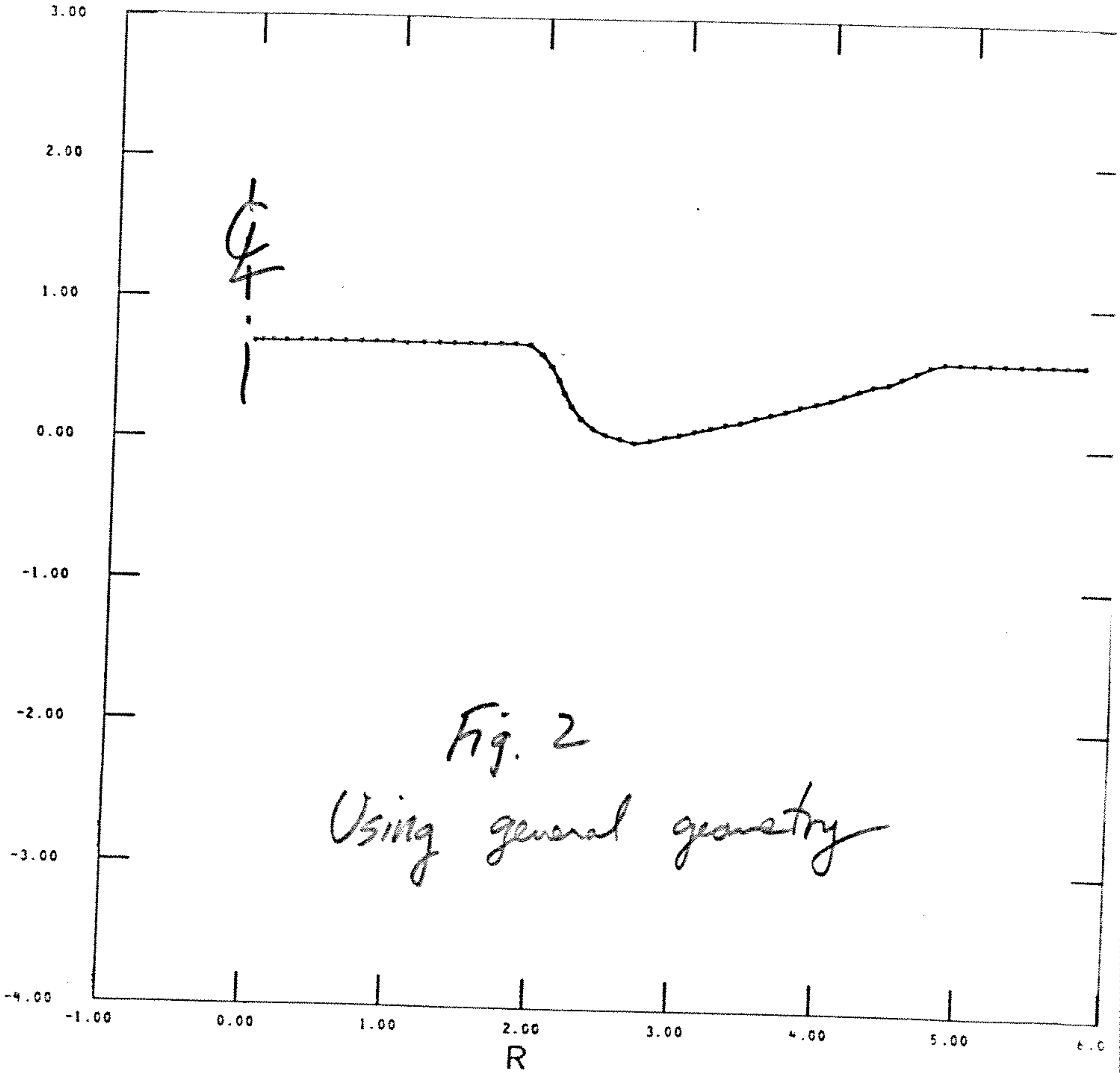


Fig. 2
Using general geometry

(P3)

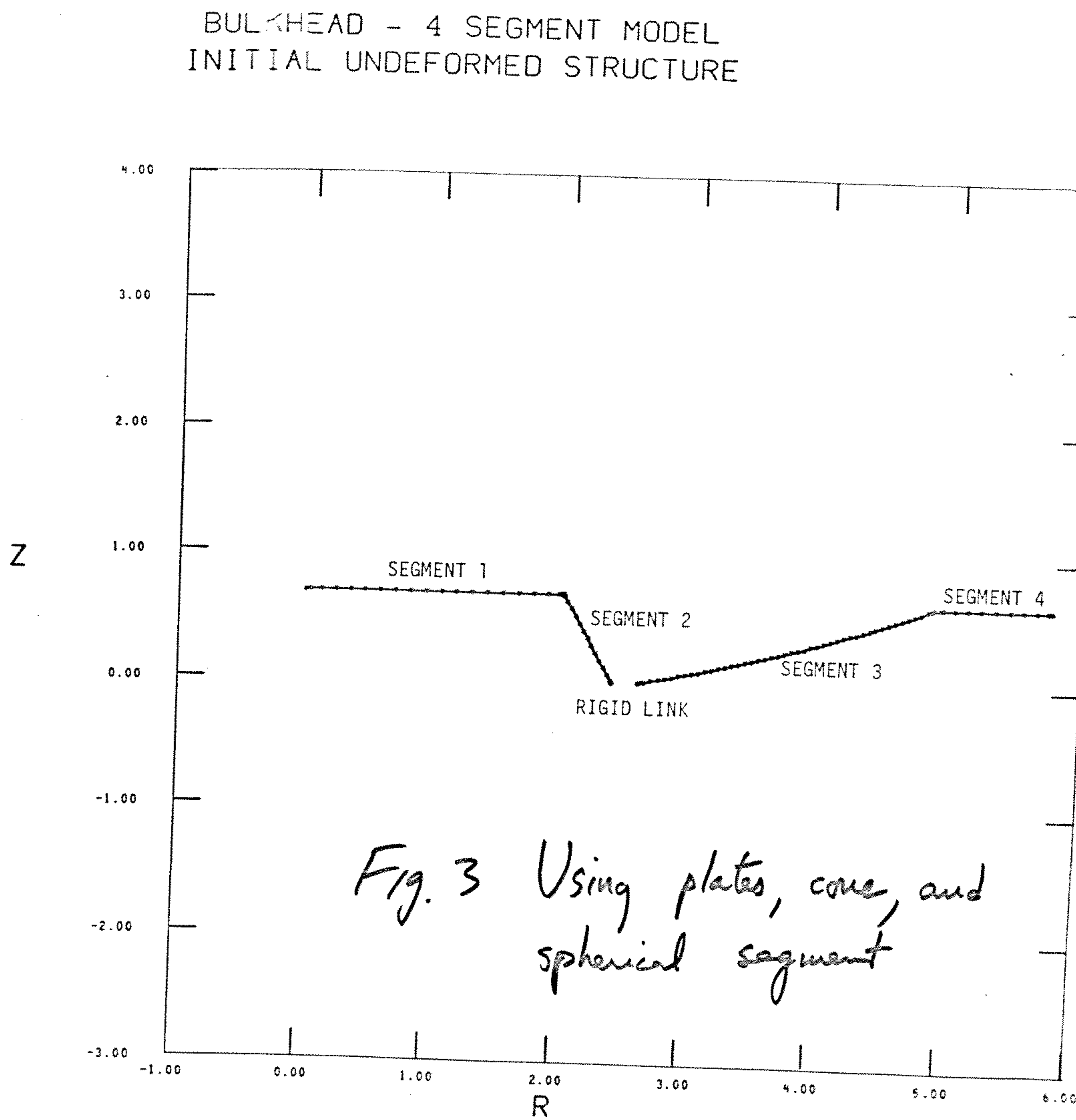


Figure 3. BOSOR5 Bulkhead Model

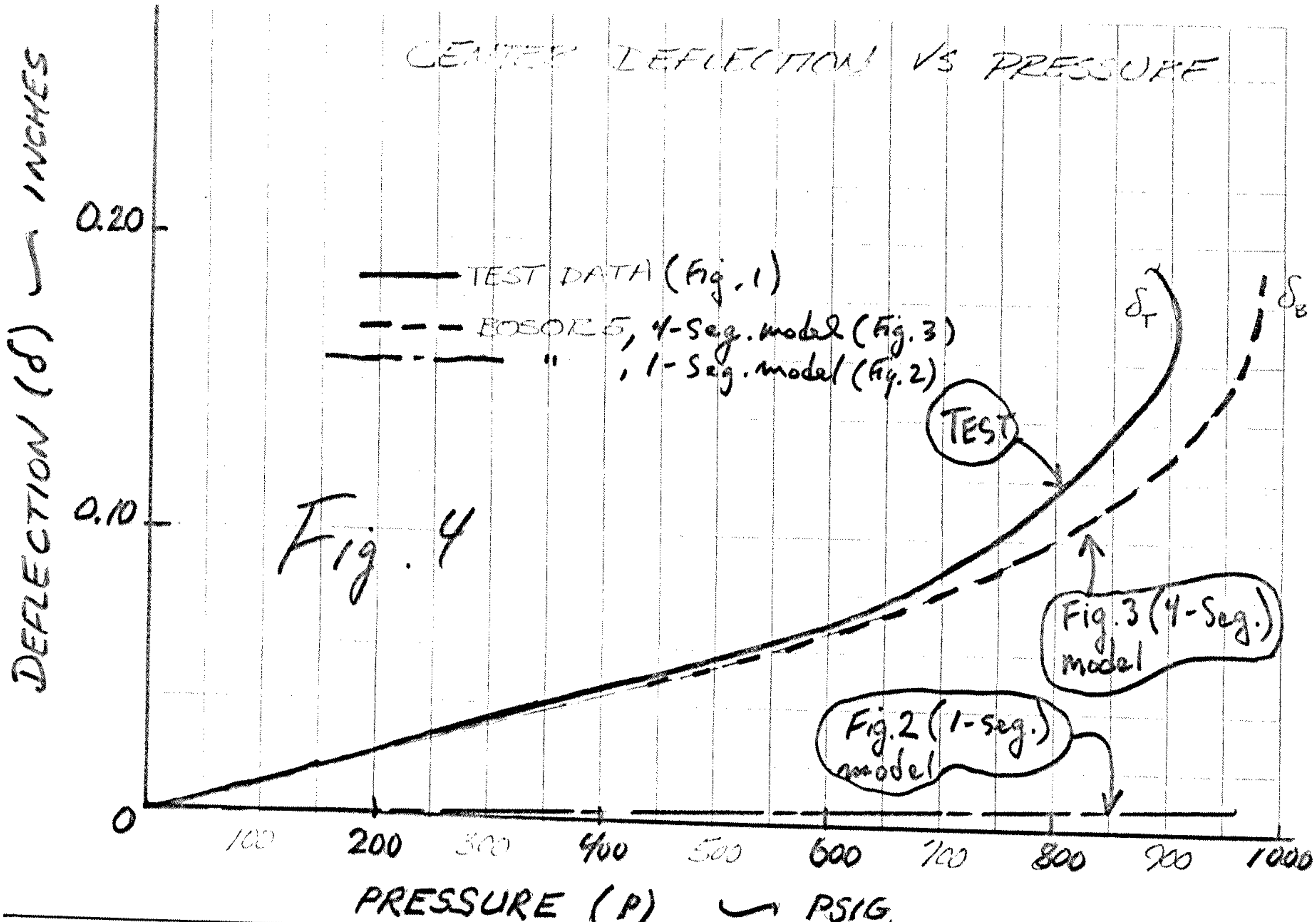


Figure 4. Bulkhead Center Deflection. Test v. Two BOSOR5 models

Lockheed
PALO ALTO
RESEARCH
LABORATORY

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Miscellaneous BOSOR5 ~~Update~~ Information

~~September 1977~~

BOSOR5 user's manual

Dear BOSOR5 User,

~~The enclosed updates should be made to your versions of BOSOR5.~~ The updates lead to the introduction of a new input datum, ICPRE. ~~(See Page 4 of updates)~~ The purpose of this new input datum is explained on the two pages of Enclosure 1. p7-8 Note that these updates are especially important if you are studying buckling of internally pressurized vessel heads. In particular, if there is only one type of load, such as pressure, acting on a pressure vessel, then the input datum ICPRE should be set equal to unity. This will lead to a more reliable solution for the minimum buckling load and the circumferential wave number at which this minimum occurs. ~~The updates are enclosed as Enclosure 2.~~

While running buckling analyses of internally pressurized very thin torispherical vessel heads made of material with very little strain hardening, I have run into cases of numerical instability when the plastic strains get quite large (more than one per cent, for example). Enclosure 3 p9-12 shows some examples. The instability seems to originate at places such as A, B, C, and D (called out in Specimen #6 in Fig. 1), which correspond to segment ends and locations where the nodal point spacing changes. I'm not sure what causes this problem. Putting more mesh points in the model helps. For analyses of buckling of elastic-plastic, internally pressurized vessel heads, use mesh point distributions such as shown in Figure 1.(The cylinder might even have more points in it.)

Please let me know how you are doing with BOSOR5. Do your predictions agree with tests? Do you have the plotting capability "up" on CALCOMP or other hardware? Should I change the name and address of my contact at your facility for future BOSOR5 news? Also, tell me what problems you've had with BOSOR5.

Sincerely yours,

David Bushnell

David Bushnell

Sept. 1977

DB:jck

LOCKHEED MISSILES & SPACE COMPANY, INC. · A SUBSIDIARY OF LOCKHEED AIRCRAFT CORPORATION

This page and the next are taken from a paper on buckling of internally pressurized elastic-plastic torispherical vessel heads.

ENCLOSURE #1

Nonsymmetric Buckling Analysis. Bifurcation buckling loads corresponding to nonsymmetric buckling modes are calculated in the following way: The user of BOSOR5 first selects an initial number of circumferential waves n_o which he feels corresponds to the minimum bifurcation load. For this wave number n_o the stability determinant is calculated for each load increment. The load is increased until the stability determinant changes sign or until eigenvalues are detected between two sequential load steps or until the maximum allowable user-specified load has been reached. At this point in the calculations a series of eigenvalue problems of the form

$$[A(n) + \lambda_n B(n)]x_n = 0 \qquad (1)$$

is set up, where

$A(n)$ = the stiffness matrix corresponding to n circumferential waves of the structure as loaded by L_1 (see following definitions)

IF ICPRE=0 This is True!

$B(n)$ = the load-geometric matrix corresponding to the prestress increment resulting from the load increment $L_2 - L_1$

ICPRE=0

(See next page for case ICPRE=1)

L_1 = the load state just before the sign change of the stability determinant

L_2 = the load state just after the sign change of the stability determinant

λ_n = the eigenvalue

x_n = the eigenvector

n = the number of circumferential waves lying in a range $n_{min} \le n \le n_{max}$, with n_{min} and n_{max} provided by the program user--note that the initial guess n_o also lies in the range $n_{min} \le n_o \le n_{max}$

ENCLOSURE #1

BOSOR5 computes a series of eigenvalues λ_n and eigenvectors x_n for $n_{min} \le n \le_{max}$ in wave number increments of n_{incr} which is also supplied by the program user.

In most cases (but not all!) the minimum λ_n corresponds to the critical bifurcation buckling load. It sometimes happens, however, that as the load increases the prestress in the shell decreases. If this phenomenon occurs in the load range for which the stability determinant first vanishes, as is often the case for internally pressurized vessel heads of the type being considered here, the critical wave number may be incorrectly predicted. This problem was solved in the cases investigated here by calculating B(n) from L_2 alone, rather than from the difference $L_2 - L_1$. (Notice, however, that this change in strategy is not appropriate for cases involving combinations of loads, some of which are eigenvalue parameters and others of which are not.)

ICPRE = 1

Summary:

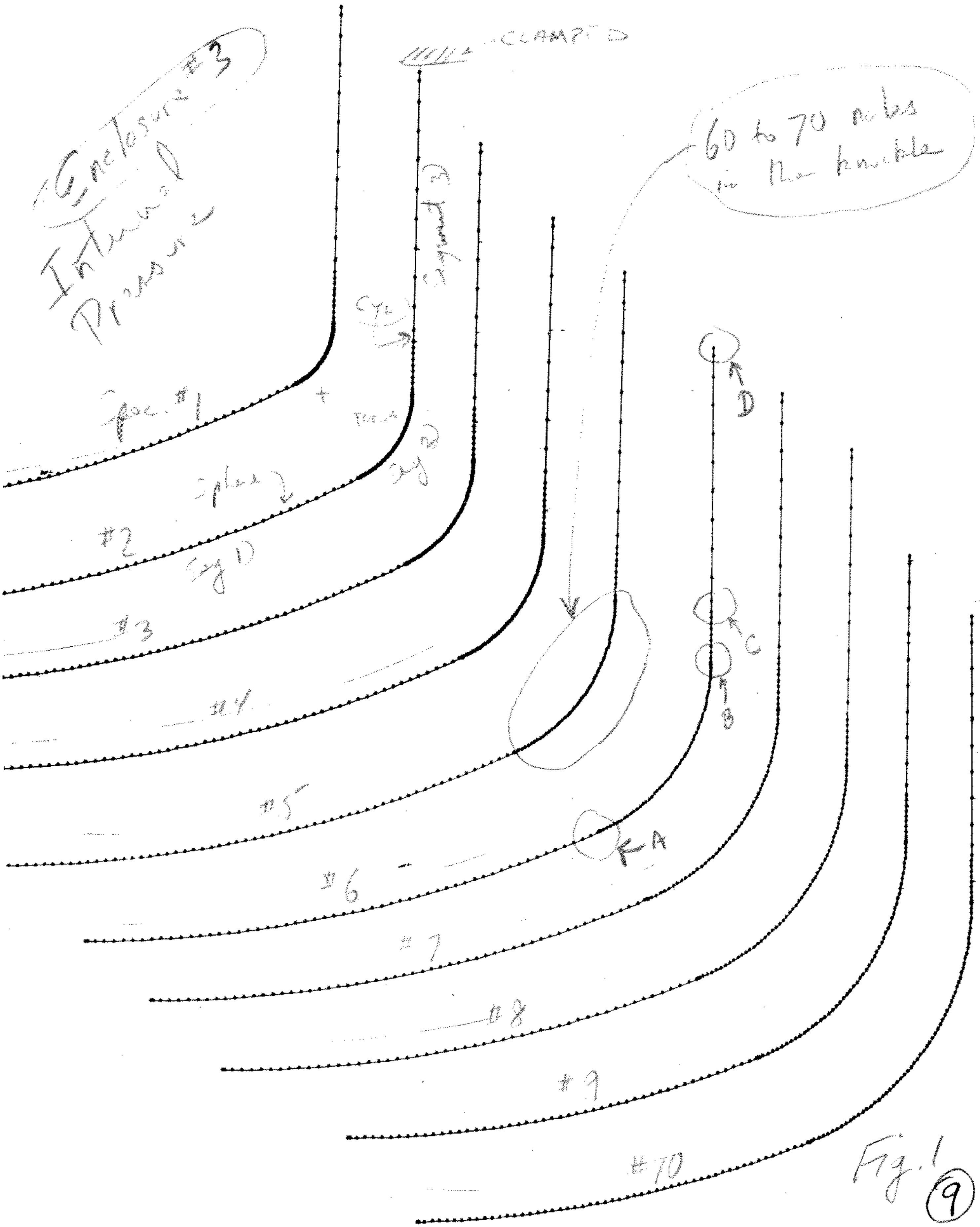
ICPRE = 0:

B(n) from $L_2 - L_1$

ICPRE = 1:

B(n) from L_2

Recommendation: Set ICPRE = 1 if there is only one kind of load acting on the structure, or if all loads (temperature is a load) vary proportionally. (Page 2 of 2)



Enclosure #3
Internal Pressure
CLAMPED
60 to 70 nodes in the knuckle
#1
#2
#3
#4
#5
#6
#7
#8
#9
#10
A
B
C
D

Fig. 1

Enclosure #3

SPECIMEN 9DA (KIRK AND GILL 1976 TESTS)

U1108/SC4020 0000 0002

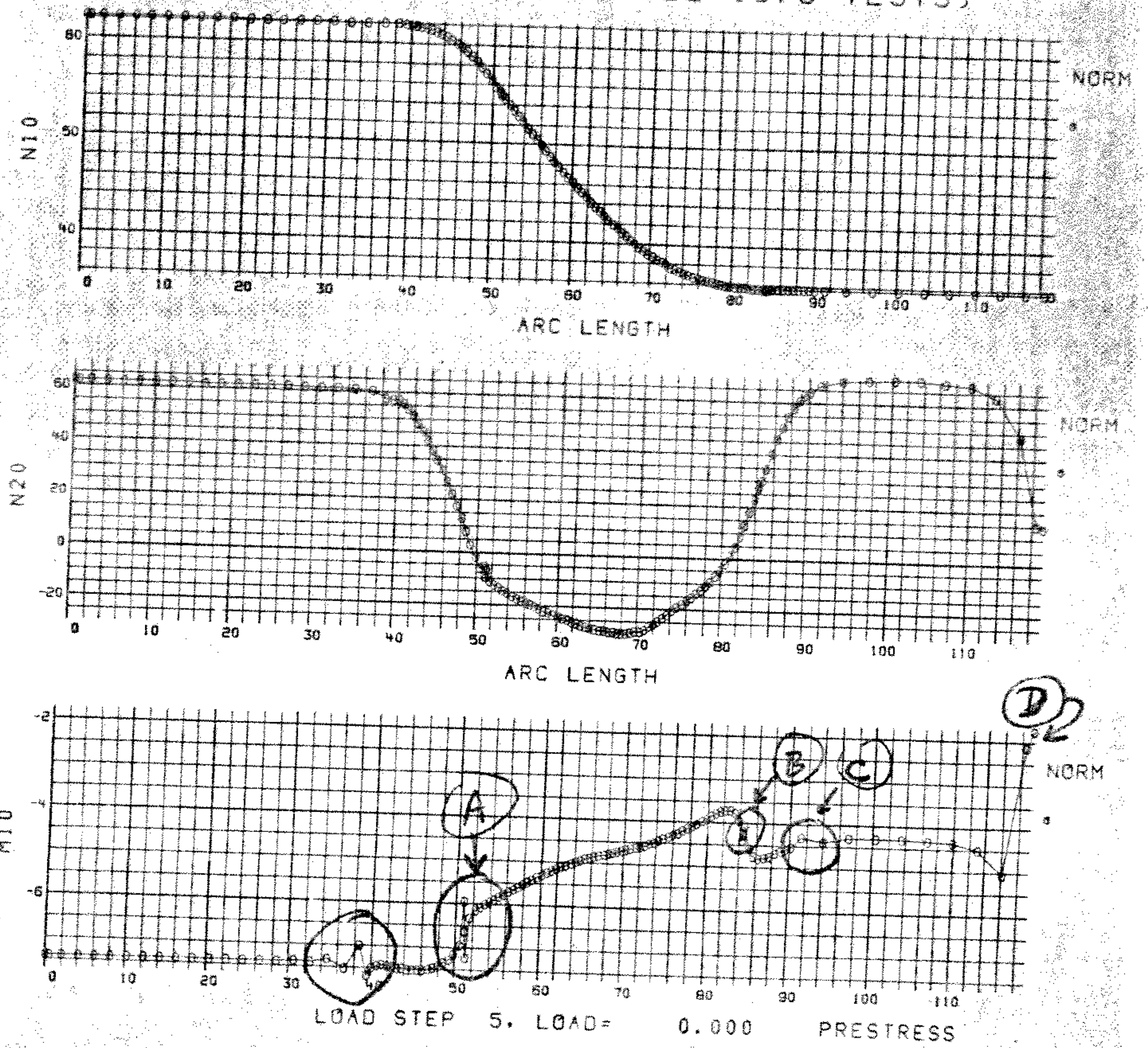


Fig 2

Enclosure #3

SPECIMEN 90A (KIRK AND GILL 1976 TESTS)
DEFORMED STRUCTURE
LOAD STEP 11, LOAD= 0.000 PRESTRESS

U1100/SC4020 0000 0001

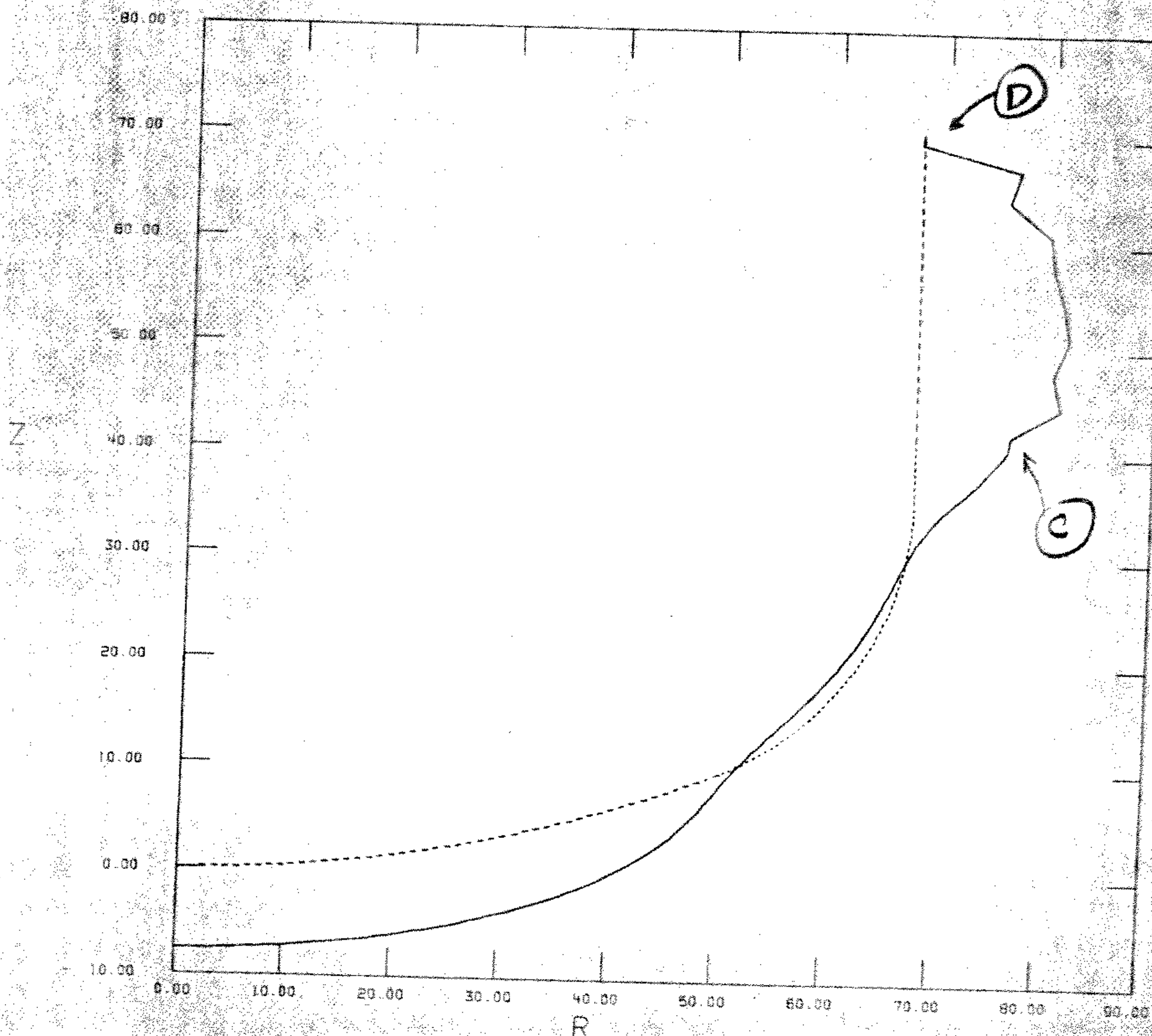
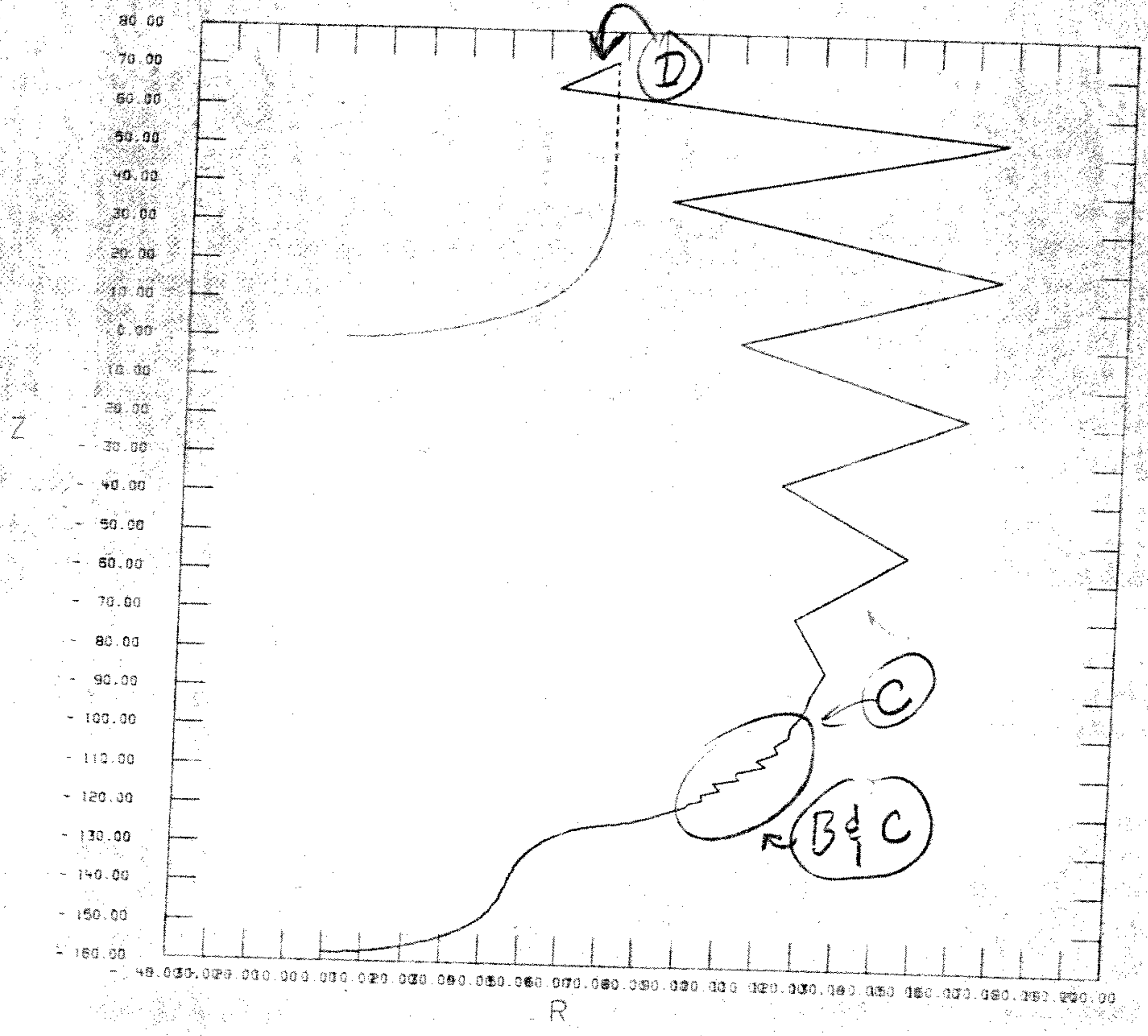


Fig 3

(11)

Enclosure II 3

SPECIMEN 10DA (KIRK AND GILL 1976 TESTS)
DEFORMED STRUCTURE
LOAD STEP 12. LOAD= 0.000 PRESTRESS



(12) Fig. 4

BOSOR5 Information

(Please return this page to David Bushnell as soon as possible. Thank you.)

(1) Further BOSOR5 news letters should be sent to:
(Please give name, Dept., address, telephone number)

(2) BOSOR5 Plotting Software at Our Facility:
(Please indicate type of computer BOSOR5 runs on and type of plotting equipment that you have successfully written BOSOR5 plotting software for.)

(3) Problems we have had with BOSOR5:

(4) Improvements we would like to see in BOSOR5:

(5) We would like to order additional copies of the BOSOR4 User's Manual ~~at $5.00/copy~~:

How many copies: $\square$

ITEM (3)

BOSOR5 plotting is virtually identical to BOSOR4.

List of BOSOR4 Users who have converted the BOSOR4 SC4020 plotting package with other than SC4020 plotter:

Name & Address	Version of BOSOR4	Plotting Hardware
*Mr. Paul C. Hermann Structural Research Chicago Bridge & Iron Co. Route 59 Plainfield, ILL 60544 (815) 436-2912 x220	IBM	Calcomp
Mr. D. G. Jenkins Lloyd's Register of Shipping 71 Fenchurch Street London, EC3M 4BS England	IBM	Calcomp 915/1036
*J. F. Imbert Chef, Departement Structures CNES - CST/PRT/SST 18 Av. Edouard Belin 31055 Toulouse Cedex France	CDC	Calcomp/Benson
Mr. Andrew Jay, EB3S-3 Pratt & Whitney Aircraft Group 400 Main Street East Hartford, CONN 06108 USA	IBM	Calcomp
Mr. Robert Zirin Bldg. 53-332 General Electric Co. 1 River Road Schenectady, New York 12345 USA	Honeywell 6080	Calcomp 925
Monsieur Venon et Madame Bressaud Section CS Group MSN Service technique des Constructions et Armes Navales 8 Boulevard Victor - 7501S Paris, France	UNIVAC 1110	Benson Type 122
BOSOR4 User MASCHINENFABRIK AUGSBURG - NURNBERG "Neue Technologie" Abtlg. EGS Dachauer Strasse 665 8000 Munchen 50 West Germany	CDC CYBER 175	Houston Instruments
Mr. Barry Bonnickson, MS 82/1720 TRW Systems Group 1 Space Park, Redondo Beach, CA 90278 (213) 536-3610	[illegible]C	CALCOMP

ITEM 4

BOSOR5 User's Manual Vol. 1 (Correction)

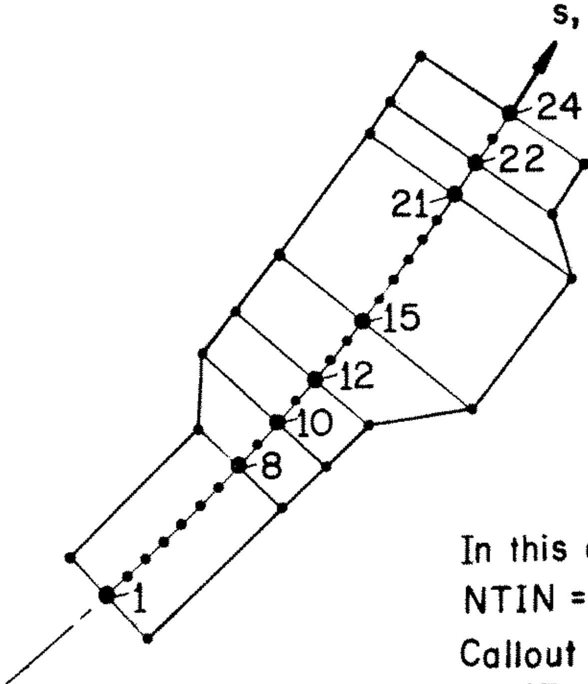
SHELL WALL MATERIAL CREEP LAW:

$$\bar{\varepsilon}^c = \left(\frac{\bar{\sigma}}{\sigma_y}\right)^M (t + t_o)^N$$

CREEPM = M (Floating Point)
CREEPN = N (Floating Point)
CREEPA = σ_y (~0.2 % Yield Stress)
CREEPB = t_o

or use value that leads to agreement between test and theory on simple material tests.

s, Increasing Arclength



In this example:
NTIN = 8
Callout Points are 1, 8, 10, 12, 15, 21, 22 and 24

COLUMN BUCKLING OF CYLINDRICAL SHELLS
UNDER INTERNAL PRESSURE

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Lockheed Palo Alto Research Laboratory
Palo Alto, California 94304

It has been pointed out by Den Hartog [1] and Canton et al. [2] that the long cylindrical tube shown in Figure 1 buckles at the Euler load even though the stress resultants in the cylindrical shell are positive or zero. The purpose of this note is to present equations of Marlowe [3], modified for application to shells of revolution, which yield the correct bifurcation loads for such problems.

Work Done by Prebuckling Stress Resultants During Buckling

For shells of revolution Marlowe [3] gives the following relations for reference surface strains:

$$\begin{aligned} \epsilon_1 &= \gamma_1 + \tfrac{1}{2}(\gamma_1^2 + \chi^2 + \gamma_{21}^2) \\ \epsilon_2 &= \gamma_2 + \tfrac{1}{2}(\gamma_2^2 + \psi^2 + \gamma_{12}^2) \\ \epsilon_{12} &= \tfrac{1}{2}(\gamma_{12} + \gamma_{21}) + \tfrac{1}{2}(\gamma_2 \gamma_{21} + \gamma_1 \gamma_{12} + \chi\Psi) \end{aligned} \qquad (1)$$

in which

$$\begin{aligned} \gamma_1 &\equiv u' + w/R_1 \\ \gamma_2 &\equiv \dot{v}/r + w/R_2 + ur'/r \\ \chi &\equiv w' - u/R_1 \\ \Psi &\equiv \dot{w}/r - v/R_2 \\ \gamma_{12} &\equiv \dot{u}/r - r'v/r \\ \gamma_{21} &\equiv v' \end{aligned} \qquad (2)$$

where

$$(\ )' \equiv \partial(\ )/\partial s; \qquad (\dot{\ }) \equiv \partial(\ )/\partial\theta \tag{3}$$

Subscripts 1 and 2 denote meridional and circumferential; s, θ are the meridional arc length and circumferential coordinate; and subscript 12 denotes shear or component of rotation about a normal $\bar{n}$ to the shell surface. The displacement components u, v, w and radius r are shown in Figure 2. The quantities R_1 and R_2 are the meridional and normal circumferential radii of curvature.

Corresponding to Eqs (1), the energy expression for bifurcation buckling of axisymmetrically loaded shells of revolution with zero prebuckling torque contains the terms

$$\tfrac{1}{2} N_{10} (\gamma_1{}^2 + \chi^2 + \gamma_{21}{}^2) + \tfrac{1}{2} N_{20} (\gamma_2{}^2 + \Psi^2 + \gamma_{12}{}^2) \tag{4}$$

These terms represent the work done by the prebuckling meridional and circumferential stress resultants N_{10}, N_{20} during the buckling process. The analogous terms for a buckling analysis based on Sanders' equations [5] are:

$$\tfrac{1}{2} N_{10} (\chi^2 + \gamma^2) + \tfrac{1}{2} N_{20} (\Psi^2 + \gamma^2) \tag{5}$$

in which γ, the "average" rotation about the normal to the shell surface, is defined as

$$\gamma \equiv \tfrac{1}{2}(\gamma_{12} - \gamma_{21}) \tag{6}$$

Effect of Uniform Normal Pressure Acting on a Shell

Marlowe [6] gives

$$W = \int_S p[w + \tfrac{1}{2} w(\gamma_1 + \gamma_2) - \tfrac{1}{2}(\chi u + \Psi v)]\, dS \tag{7}$$

for the work done by the uniform normal pressure during the buckling process.

Equation (7) is to be compared with an analogous expression given by Cohen [7], which for uniform pressure simplifies to

$$W = \int_S p[(1 + \gamma_1 + \gamma_2)w - \tfrac{1}{2} w^2(\frac{1}{R_1} + \frac{1}{R_2}) + \tfrac{1}{2}(\frac{u^2}{R_1} + \frac{v^2}{R_2})]\, dS \quad (8)$$

In Eqs. (7) and (8) the integration is over the shell surface area S.

Use in BOSOR4 [8] or BOSOR5 [9] of expressions (5) and (8) yields incorrect values for the buckling load of a long cylinder loaded as shown in Figure 1. For example, such a cylinder with modulus $E = 10^7$ psi, Poisson's ratio $\nu = 0.3$, thickness $t = 1.0$ in, radius $R = 10$ in and length $L = 600$ in should buckle at $p_{cr} = 2741$ psi, according to Euler's formula

$$\pi R^2 p_{cr} = P_{cr} = \pi^2 E I/L^2 \text{ or } p_{cr} = \pi^2 E R t/L^2 \quad (9)$$

The BOSOR5 program, as based on Eqs. (5) and (8) yields $p_{cr} = -6.9$ psi for this problem. After modification of BOSOR5 such that the analysis is based on Eqs. (4) and (7), the predicted critical pressure is $p_{cr} = 2739$ psi.

Practical Application

The above comments might seem academic at first because once one is aware of the possibility of buckling of long tubes loaded as shown in Figure 1, one simply applies Euler's formula and doesn't turn to a computer program such as BOSOR5 for the solution. However, there is a class of problems for which a computerized solution is sometimes necessary: elastic or elastic-plastic bifurcation buckling of bellows. Because the axial stiffness of such bellows is much, much less than the circumferential or shear stiffnesses, the critical bifurcation buckling mode corresponds to a column-type buckling, which is called "squirm" by engineers who are concerned with bellows design.

References

1. Den Hartog, J. P., Comments made in a keynote lecture given at the Third Canadian Congress of Applied Mechanics, Calgary, Canada, May 17-21, 1971

2. Canton, B., A. Hoffman, R. L. Roche, and C. Troclet, "Elastic and Elastic-Plastic Buckling of Vessel Heads - Computation by the CEASEMT System", Trans. 4th Int. Conf. on Structural Mechanics in Reactor Technology, Vol. G, Paper 7/4, Aug. 1977

3. Marlowe, M. B., and W. Flügge, "Some New Developments in the Foundations of Shell Theory", Lockheed Missiles & Space Co. Report No. LMSC-6-78-68-13, May 1968, p. 144, Eqs. B.8C

4. Sanders, J. L., Jr., "Nonlinear Theories for Thin Shells", Quarterly of Applied Mathematics, Vol. 21, pp. 21-36, 1963

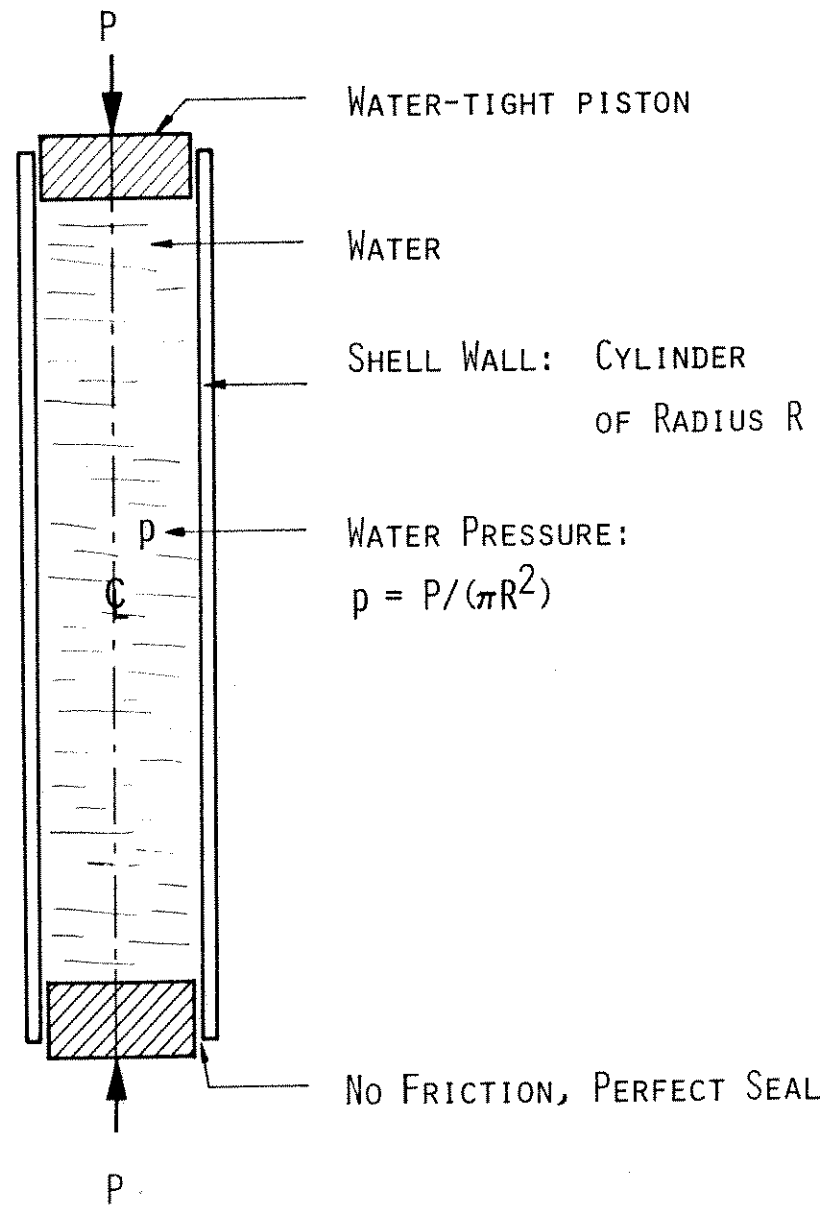
5. Bushnell, D., "Analysis of Buckling and Vibration of Ring-Stiffened, Segmented Shells of Revolution", Int. J. of Solids & Structures, Vol. 6, pp. 157-181, 1970

6. Marlowe, M. B., Unpublished notes

7. Cohen, G. A., "Conservativeness of a Normal Pressure Field Acting on a Shell", AIAA J., Vol. 4, No. 10, pp. 1886, 1966

8. Bushnell, D., "Stress, Stability and Vibration of Complex, Branched Shells of Revolution", Computers & Structures, Vol. 4, pp. 399-435, 1974

9. Bushnell, D., "BOSOR5 - Program for Buckling of Elastic-Plastic Complex Shells of Revolution Including Large Deflections and Creep", Computers & Structures, Vol. 6, pp. 221-239, 1976



EULER BUCKLING LOAD FOR PINNED ENDS:

$$P_{cr} = \pi^2 E I/L^2$$

Figure 1. Long, Water-Filled Cylindrical Column under Axial Compression

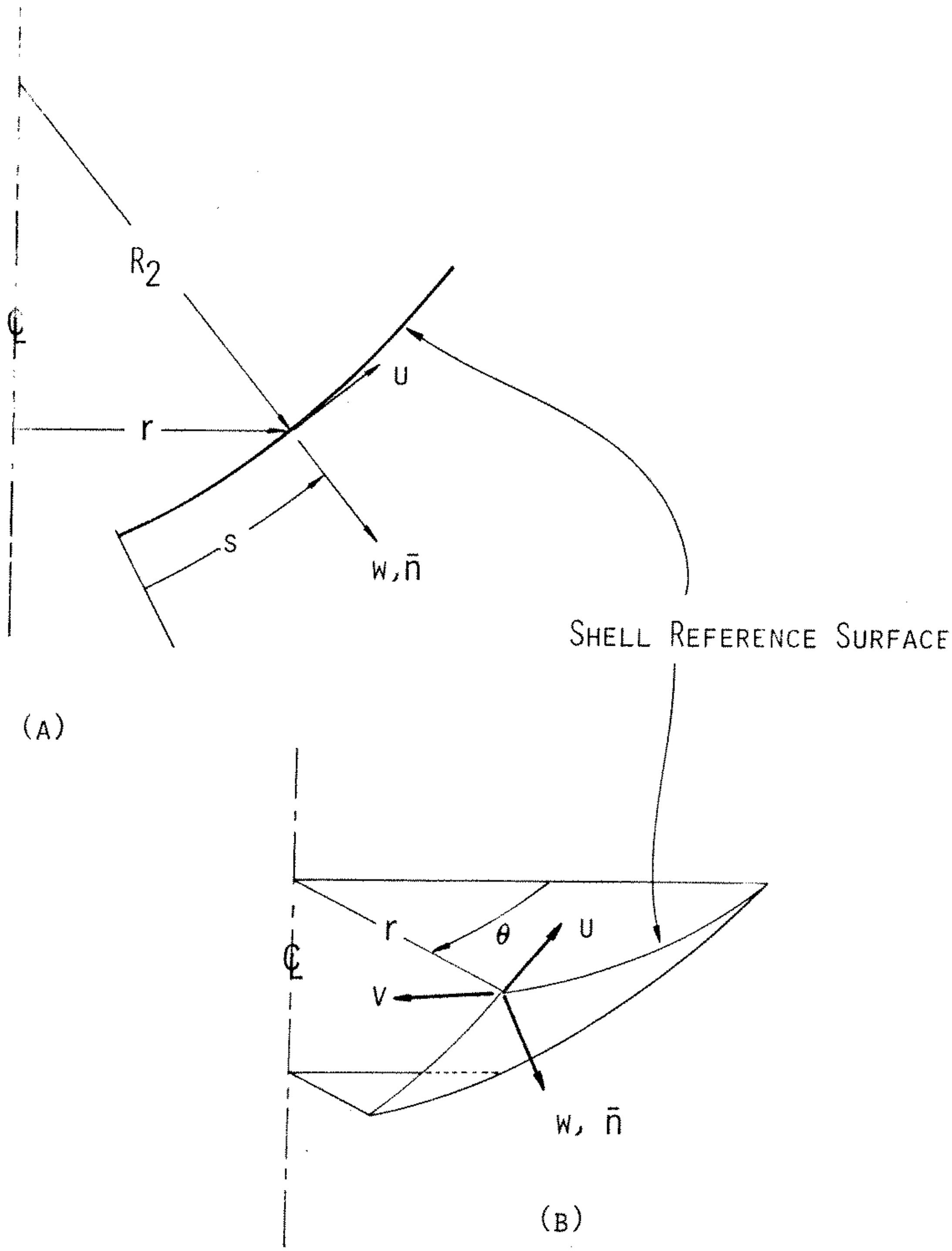


Figure 2. Shell of Revolution Coordinates and Displacement Components

ALL BOSOR5 VERSIONS

How to Perform Linear Elastic Bifurcation Buckling with BOSOR5.

You may want to use BOSOR5 to calculate elastic bifurcation buckling loads for a wide range of circumferential wave numbers in order to obtain an approximate idea of the buckling phenomenon without using up a lot of computer time in the calculation of nonlinear material and geometrical effects. For those of you familiar with BOSOR4, the following treatment in BOSOR5 would be equivalent to an INDIC=1 analysis with BOSOR4:

Suppose you are analyzing a shell with just one kind of load (pressure, for example). Then in the preprocessor (see pp P60-P62 in the BOSOR5 User's Manual) you might set:

```
IUTIME = 1
DTIME, TMAX = 1., 5000.
NFTIME = 1
NPOINT (1) = 2
F(1, 1), F(1, 2) = 0., 5000.
T(1, 1), T(1, 2) = 0., 5000.
NCOND = etc.
      -
      -
      -
```

In the mainprocessor you would set:

```
INDIC, IDEFORM = -2, 0
KSTEP, KMAX, MAXTRL, ITMAX, ITIME = 0, 2, 1, 10, 0
NOB, NMINB, NMAXB, INCRB, NVEC = 10, 10, 60, 10, 1 (for example)
TIME = 0.
      -
      -
      -
```

Note that in the preprocessor I have set F(I, J) = T(I, J) (see p. P62, bottom). This is always good practice if you have only one kind of load or if all loads vary proportionally in time. With this input you can see to it that time = load, so that the output in the main processor will indicate the current load (although it will call it "time", of course).

Given this input, BOSOR5 will do the following:

(1) Calculate prebuckling state for time (load) = 0.0.

(2) Calculate prebuckling state for time (load) = 1.0.

(3) Calculate stability determinant for NOB = 10 for time (load) = 0.0 and time (load) = 1.0.

(4) Calculate eigenvalues λ_n and mode shapes q_n from $[K_1(n) + \lambda_n K_2(n)]\{q_n\} = 0$ for $n = 10, 20, 30, 40, 50, 60$ circumferential waves.

In (4), $K_1(n)$ is the stability stiffness matrix for the shell as loaded at time (load) = 0.; $K_2(n)$ is the "load-geometric" matrix corresponding to the "unit" load at time (load) = 1.0; λ_n is the eigenvalue; and q_n is the eigenvector.

(In the above discussion it has been tacitly assumed that a unit load, p = 1.0 psi, for example, is very small compared to a value that would give rise to nonlinear material or geometrical effects. In some systems of units, or for some structures, p = 1.0 may be a large load. Then the user should set DTIME equal to some very small "unit" number such as .01 or .001.)

w nodes) which will be read

IHVALU= nodal point callouts for which spacing is to be given. Spacing will vary linearly between these callouts. See Fig. A4.

HVALU= spacing between adjacent w nodes at callout points. HVALU(I) is the meridional arc length between w(IHVALU(I)) and w(IHVALU(I)+1). See Fig. A4 for an example. Only relative sizes of spacing are required, not absolute values.

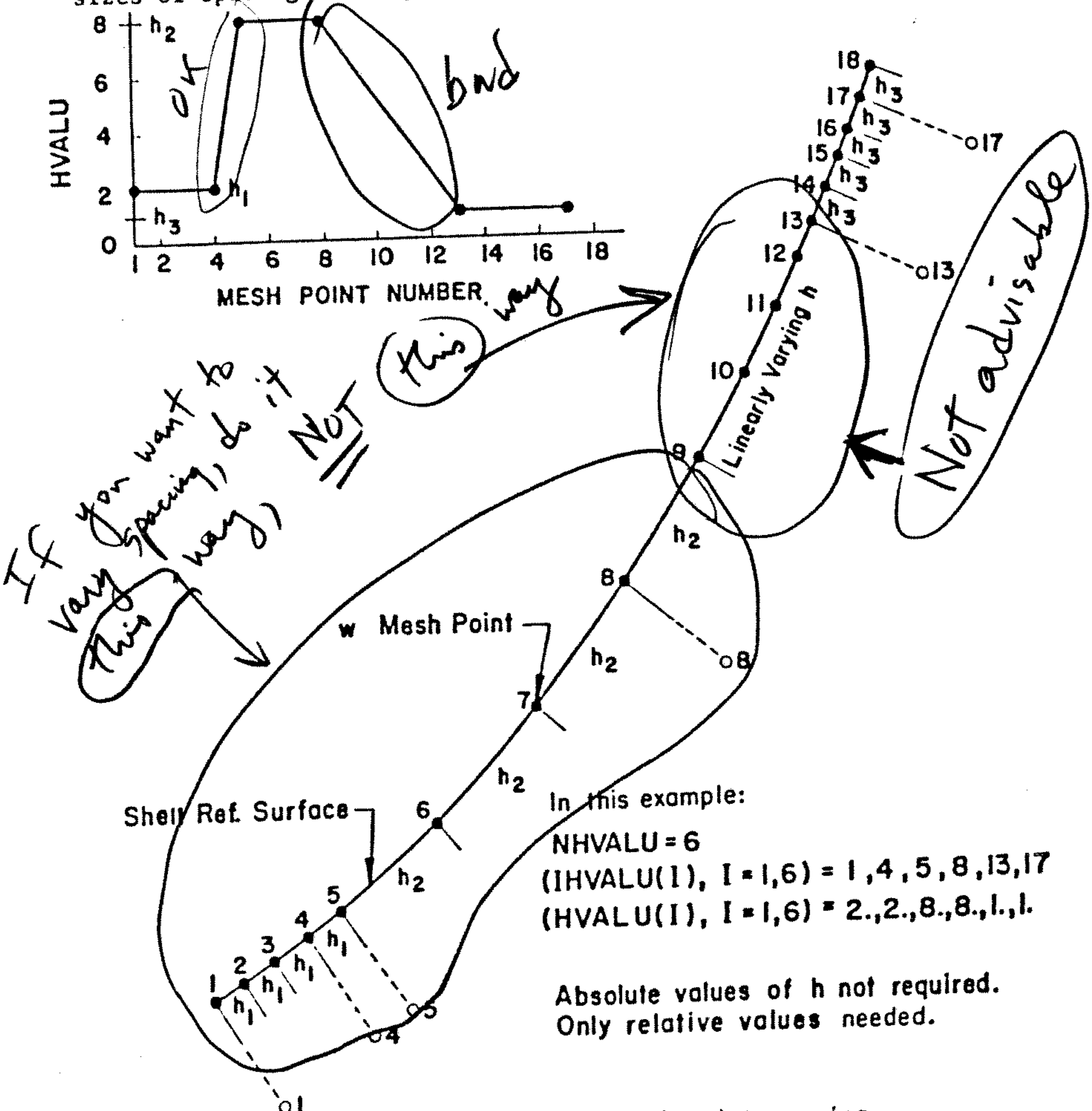


Fig. A4 Input for variable nodal point spacing

from BOSOR4 User's manual

Dear BOSOR users,

The enclosed material comprises the first newsletter sent in several years. Let me summarize the important items here:

1. In September, 1985, Nijhoff in the Netherlands released a book, **COMPUTERIZED BUCKLING ANALYSIS OF SHELLS** which "goes" very well with the BOSOR4 and BOSOR5 computer programs. Much of the book is devoted to BOSOR-type of shell buckling problems. A publisher's order form is enclosed. (It should be sent to the publisher, not me.)

2. The VAX version of BOSOR4 has been considerably enhanced, both in 1984 and in 1985. (Note in particular the new capability to calculate stresses throughout the wall thickness in laid up composite laminated walls, an enhancement completed in August, 1985.) Since the BOSOR programs are not expensive, I strongly urge you to invest in the latest versions. Please fill out the BOSOR4, BOSOR5, PANDA order form and return in to me.

3. My department number and telephone number have changed. I still do the same kinds of things, but applied mechanics at Lockheed is now part of Dept. 93-30 in Bldg. 255. My telephone number is (415)424-3237.

Sincerely yours,

David Bushnell

David Bushnell Dept. 93-30, Bldg. 255 (415)424-3237

Nov. 4 1985