EFFECTS OF INTERNAL PRESSURE ON AXIAL COMPRESSION STRENGTH OF CYLINDERS

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INTRODUCTION

Experimental investigations have shown that the critical buckling stress of an axially loaded cylinder is significantly increased by internal pressure. The destabilizing effect of initial imperfections is reduced. The circumferential tensile stress induced by pressurization inhibits the diamond buckling pattern associated with elastic buckling of cylinders, and at sufficiently high pressurization, the cylinder buckles in the classical axisymmetric mode at approximately the classical buckling stress.

Tests have been performed by several researchers on cylinders of varying geometries and materials. There was a large variation in the buckling stresses of the cylinders without internal pressure with minimum stresses of about 8% of the classical value and maximum stresses of over 50% of the classical value. The increases in buckling stresses due to internal pressure were found to have a much smaller variation. An analysis has been made of available test data and a design equation has been proposed for determining the increase in buckling stress of an axially compressed cylinder due to internal pressure.

ANALYSIS OF TEST DATA

The following is a summary of tests that have been performed on cylinders to determine the effect of internal pressure.

Reference Number	R/t	$\frac{L}{\sqrt{Rt}}$	Material	Ring Stiffened	No. of Sets of Tests
11	682-783	7-26	Aluminum	Yes	9
15	583-1750	152-263	Aluminum	No	3
20	1750	103	Aluminum	No	1
20	1006,2734	56-146	Stainless Steel	No	3
22	514	90-136	Aluminum	No	1
23	602	52	Aluminum	No	1
24	1250	141	Aluminum	No	1
43	400-2000	40-89	Mylar	No	38

The data taken from these references is plotted in Figures 1a-1m in nondimentional form of $\bar{\sigma}$ versus $\bar{p}$.

$$\bar{\sigma} = \frac{\sigma_{cr}}{E}\frac{R}{t} \quad \text{and} \quad \bar{p} = \frac{p}{E}\left(\frac{R}{t}\right)^2$$

σ_{cr} = buckling stress determined from tests, psi
p = internal pressure, psi
E = Young's modulus, psi
R = radius, in.
t = thickness, in.

The tests by Dow and Peterson (11) and those by Weingarten, et. al. (43) show that the maximum buckling stresses obtained approach the classical buckling stress of:

$$\sigma_{cl} = \frac{1}{\sqrt{3\ (1-\nu^2)}}\ E\,\frac{t}{R} = 0.605\ E\,\frac{t}{R} \qquad \text{for } \nu = 0.3 \qquad (1)$$

This corresponds to $\bar{\sigma}$ = 0.605.

The buckling stresses of the same cylinders without internal pressure ($\bar{p}$ ~ 0) varied widely from $\bar{\sigma}$ = 0.14 to 0.29. There is also a great variation in the minimum and maximum values of $\bar{\sigma}$ for the other sets of test data.

A study was also made to determine the increase in the buckling stress due to internal pressure for each set of tests. The results are presented in Figures 2a-2e in nondimensional form of C_p vs. $\bar{p}$.

$$\sigma_{cr} = (C_o + C_p)\ E\,\frac{t}{R} \qquad (2)$$

$$C_p = \bar{\sigma} - C_o$$

where C_o is the value of $\bar{\sigma}$ for an unpressurized cylinder ($\bar{p} = 0$).

The tests performed by Weingarten, et. al. (43) on mylar cylinders show increases in the values of C_p with increases in the R/t ratio. No similar correlation was found for the aluminum and stainless steel cylinders. The shapes of the curves are seen to be quite similar for all materials and R/t ratios.

A lower bound curve of all test data has been developed and is recommended for a design equation for elastic buckling cylinders with R/t ratios of 400 to 2000. The formula for this curve is given by Eq. 3.

$$C_p = \frac{1.06}{3.24 + 1/\bar{p}} \quad (3)$$

DESIGN PROCEDURE

The allowable buckling stress for a cylinder subjected to combined axial compression and internal pressure can be determined by the following equation.

$$F_{xc} = \frac{\eta F_{xe}}{F_s} \quad (4)$$

$$F_{xe} = (C_o + C_p) E \frac{t}{R} \quad (5)$$

F_{xc} = design buckling stress, psi
F_{xe} = elastic design buckling stress, psi
FS = factor of safety
η = plasticity reduction factor
C_p = given by Eq. 3

The following equations taken from API Bulletin 2U (45) are recommended for determining the buckling stresses of unpressurized fabricated cylinders with or without ring stiffeners.

$$C_o = \begin{cases} \dfrac{102.2}{195 + R/t} < 0.9 & R/t < 610 \qquad (6) \\ 0.125 & R/t > 610 \qquad (7) \end{cases}$$

For ring stiffened cylinders subjected to internal pressure, use the greater of $C_o\bar{c}$ or $C_o + C_p$ in Eq. 5. The term $\bar{c}$ is defined below.

$$\bar{c} = \begin{cases} 2.64 & M \leq 1.5 \\ 3.13/M^{0.42} & 1.5 < M < 15 \\ 1.0 & M > 15 \end{cases}$$

where:

$M = L/\sqrt{Rt}$

L = length of cylinders between rings or total length of an unstiffened cylinder, in

To determine the buckling stress in the inelastic range, the effective stress, F_e, rather than the elastic buckling stress, F_{xe}, will be used to determine the plasticity reduction factor η.

$$F_e = (F_{xe}^2 + F_\theta^2 + |F_{xe}|\ F_\theta)^{1/2} \qquad (7)$$

$$F_\theta = p\ R/t \qquad (8)$$

$$
\eta = \begin{cases} 1 & \Delta < 0.55 \\ .45/\Delta + 0.18 & 0.55 < \Delta < 1.6 \\ \dfrac{1.31}{1 + 1.15\Delta} & 1.6 < \Delta < 6.25 \\ 1/\Delta & \Delta \geq 6.25 \end{cases} \tag{9}
$$

$$
\Delta = \frac{F_e}{F_y}
$$

COMPARISON WITH OTHER DESIGN RECOMMENDATIONS

Design rules are given in Space Vehicle Design Criteria, NASA SP-8007 (46) and Shell Analysis Manual, NASA CR-912 (47). Both references present a graph for determination of C_p which was based upon Fig. 13 of Ref. 20. The following equation is a very close approximation to the curve given in these references.

$$
C_p = \frac{1.25}{5 + 1/\bar{p}} \tag{10}
$$

The following is another approximation developed by CBI Oak Brook Engineering of the design curve given in the above references.

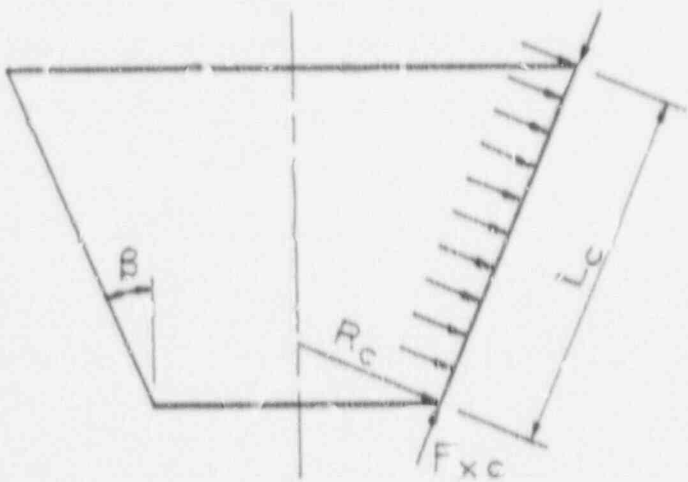
$$
\begin{aligned} C_p = {} & 0.01983 + 0.7886\bar{p} - 1.5272\bar{p}^2 + 1.5208\bar{p}^3 \\ & - 0.73323\bar{p}^4 + 0.13398\bar{p}^5 \end{aligned} \tag{11}
$$

These two equations are compared with Eq. 3 in Fig. 3. The equations fall slightly above Eq. 3 for values of $\bar{p} < 0.2$ and below Eq. 3 for $\bar{p} > 0.2$. The increased values of C_p given by Eq. 3 are supported by all of the available test data which is significantly greater than the data base considered in Ref. 20.

Design rules are also given in Buckling of Steel Shells, European Recommendations (48) with C_p given as a function of the R/t ratio as well as $\bar{p}$. These rules follow closely the curves derived from Ref. 43, and shown on Fig. 2e, for values of $\bar{p} < 0.2$ but are much flatter for larger values of $\bar{p}$. These rules are compared with Eq. 3 in Fig. 4. The C_p values derived from these rules may be unconservative for values of $\bar{p} < 0.4$.

APPLICATION TO CONES

The design procedure for cylinders can be applied to pressurized cones by substituting R_c for R and L_c for L in the design equations.



APPLICATION TO SPHERE

The theoretical buckling stress of a cylinder under uniform external pressure is the same as a cylinder under axial compression. API specifications (49,50) are based upon the assumption that unstiffened spheres under axial compression only will fail at the same stress as an unstiffened cylinder under axial compression. It therefore appears reasonable to assume that the rules determined for cylinders subjected to internal pressure can be applied to spheres.

REFERENCES

References 1 to 44 are the output from a search of the CBI computerized bibliography on shell stability for references which contain applicable test data or discuss test data given in other references.

R E F E R E N C E S

SHELL TYPE: ALL TYPES INCLUDED
STIFFENING: STIFFENED AND/OR UNSTIFFENED
LOAD TYPE: COMBINED LOADS
AXIAL COMPRESSION
INTERNAL PRESSURE
DATA TYPE: TEST RESULTS
CONSTRUCTION ALL TYPES
MATERIAL ALL MATERIALS

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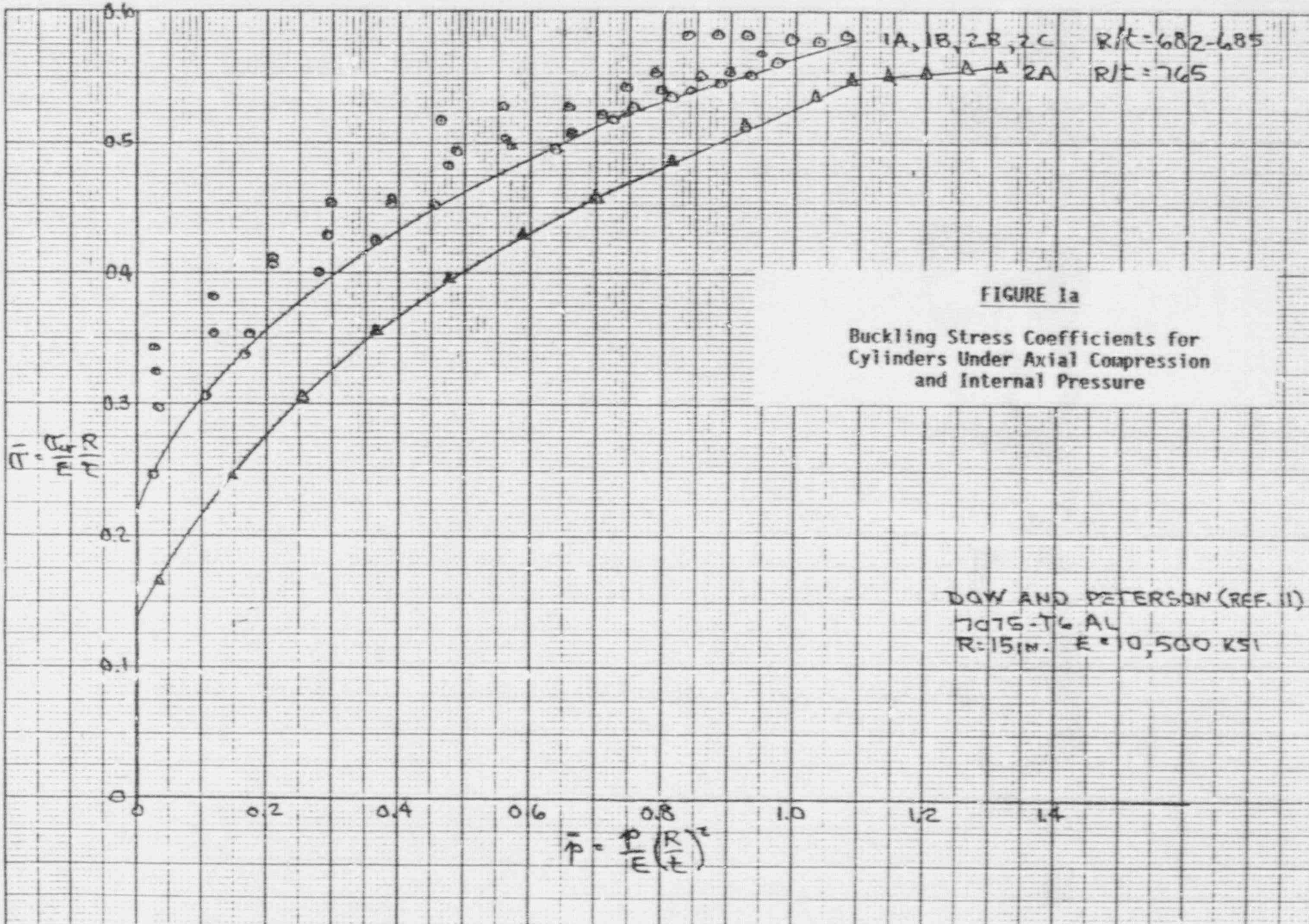


FIGURE 1a

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

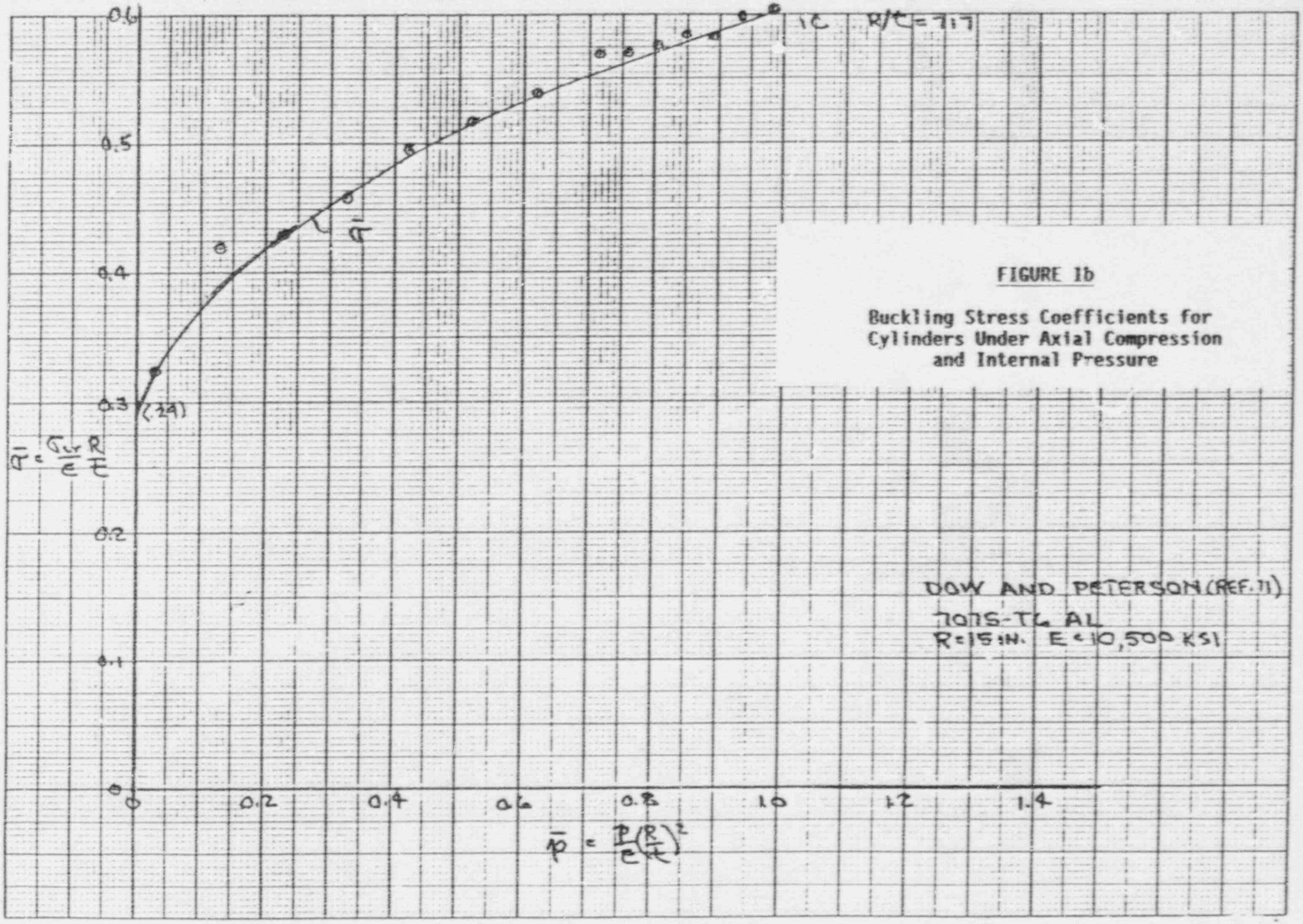


FIGURE 1b

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

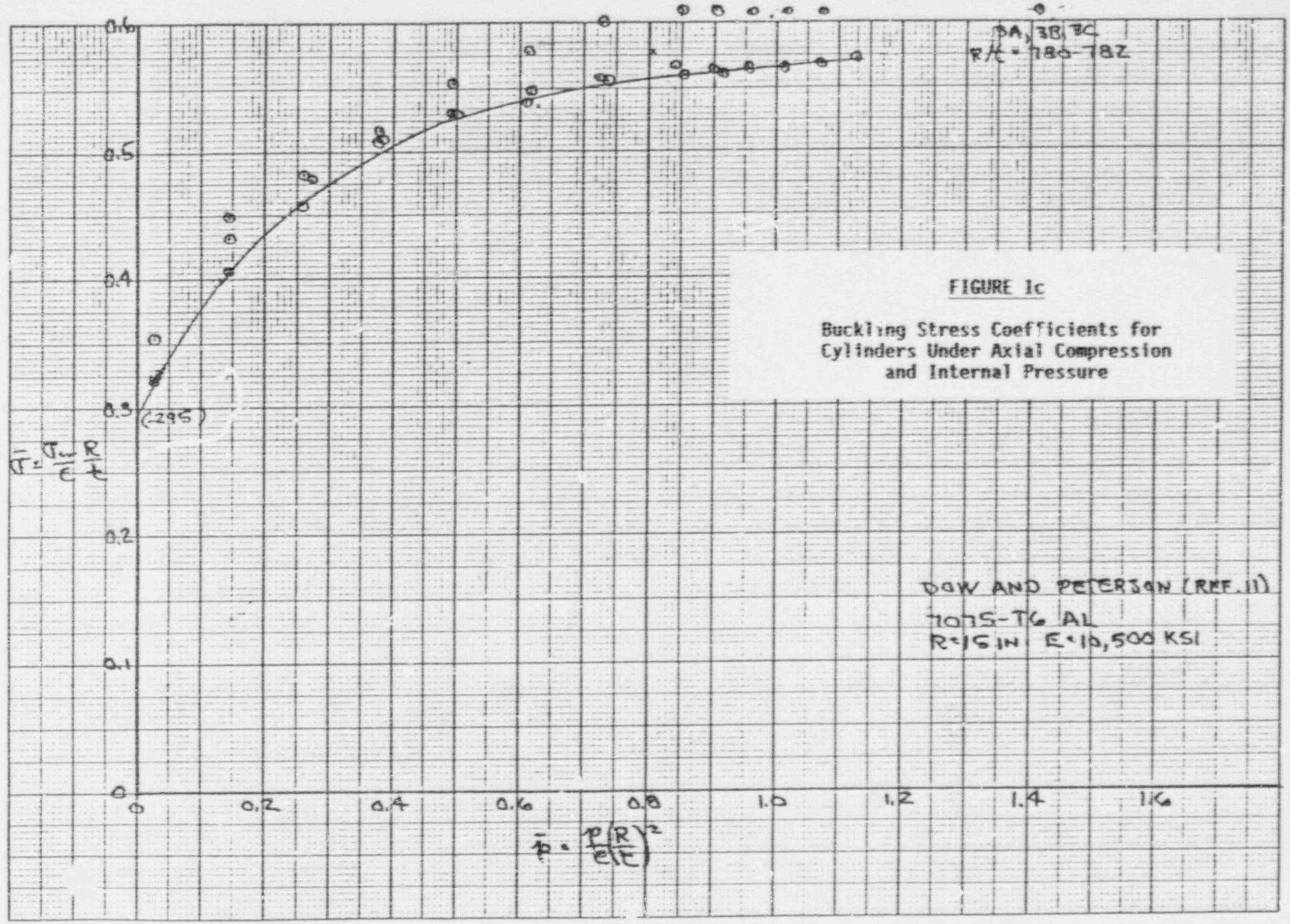


FIGURE 1c

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

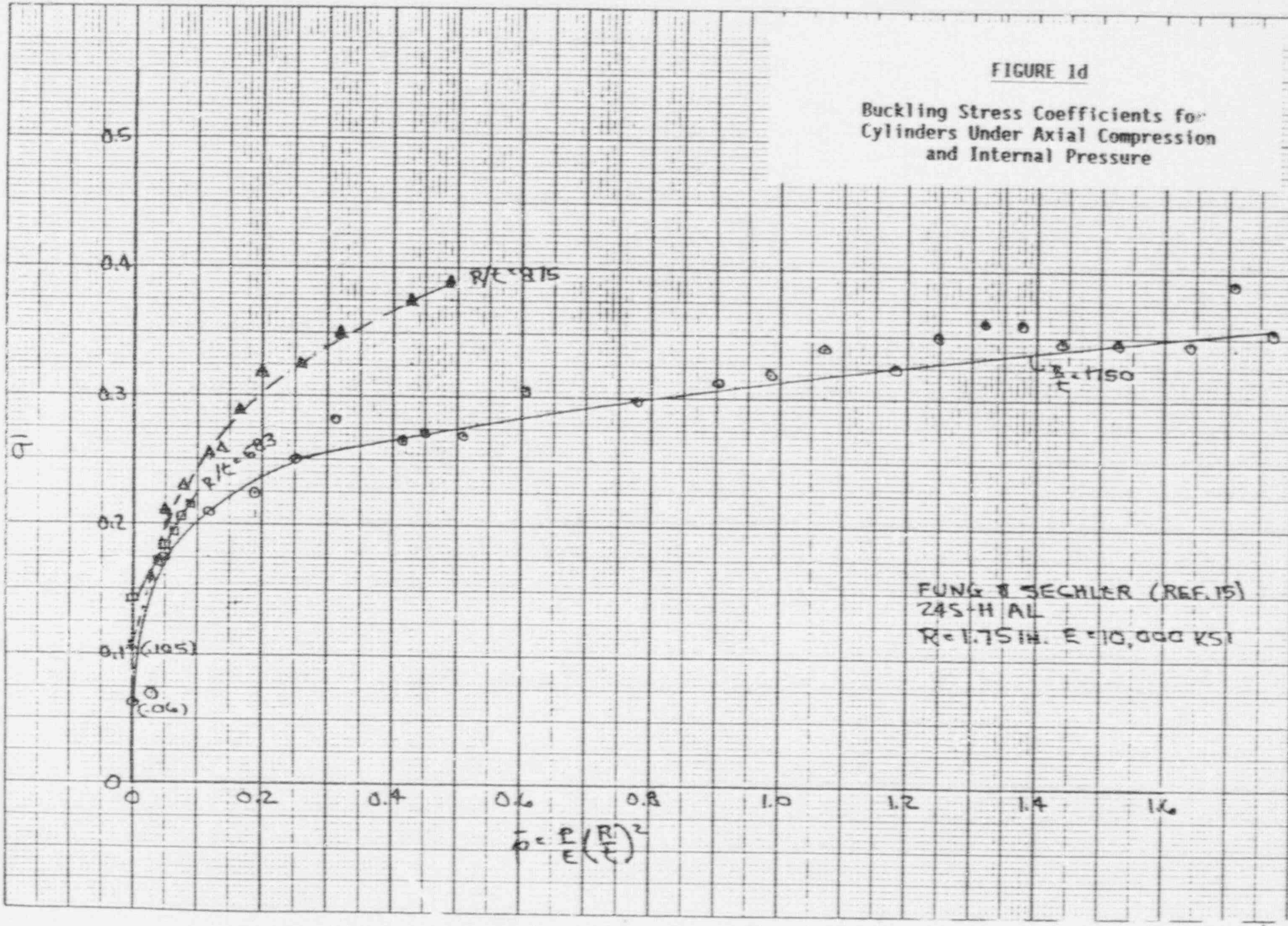
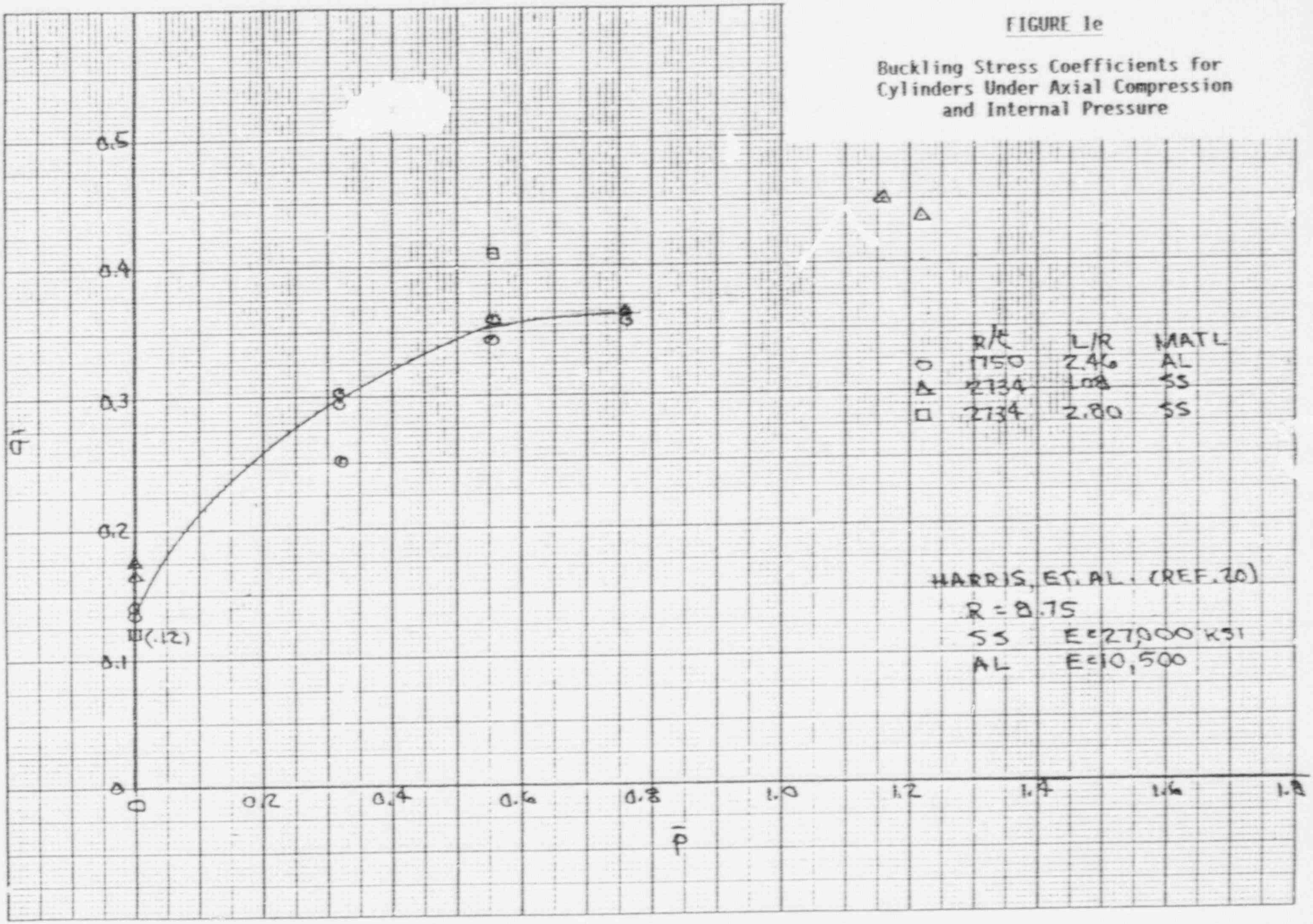


FIGURE 1d

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

FIGURE 1e

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure



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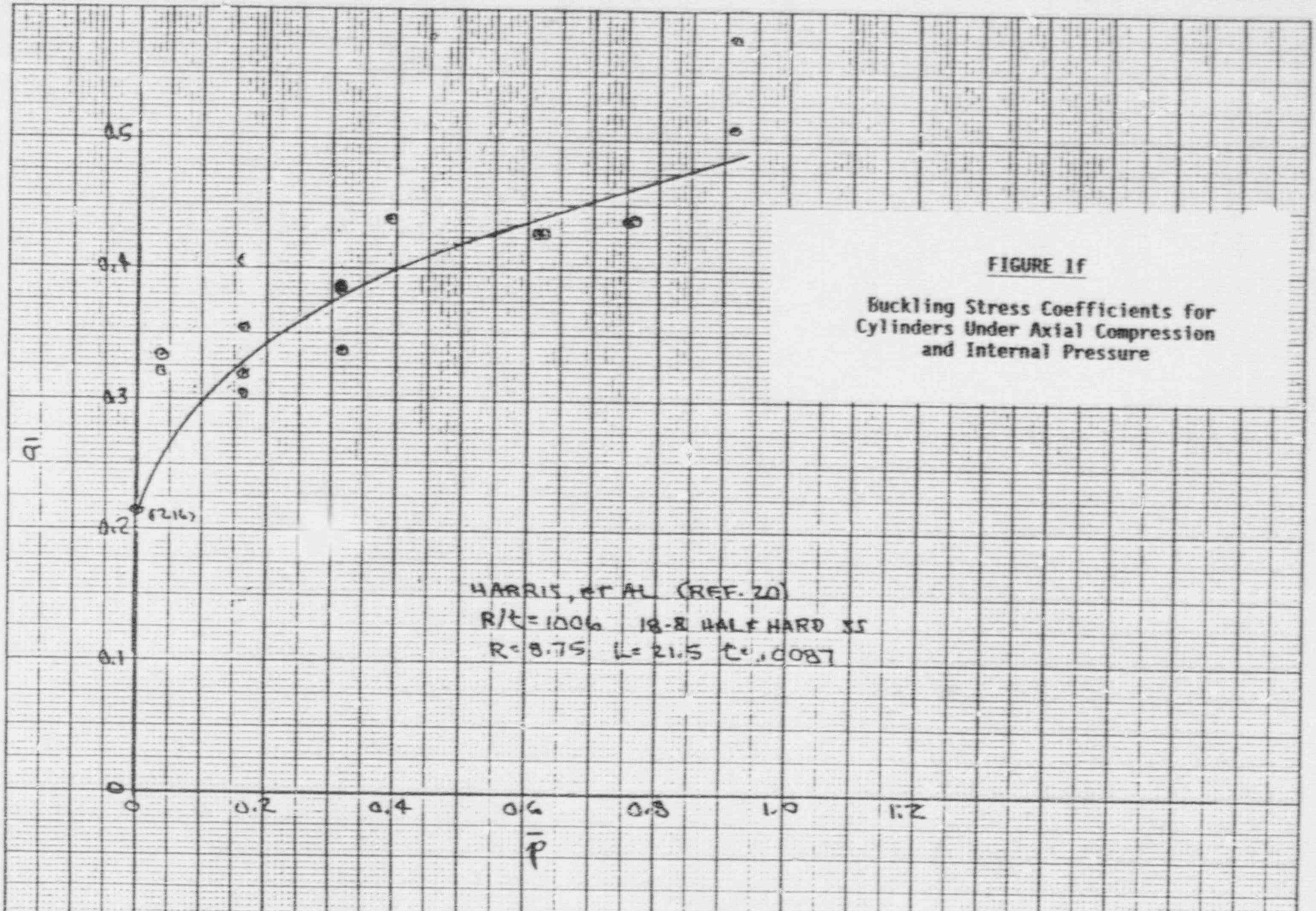

FIGURE 1f

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

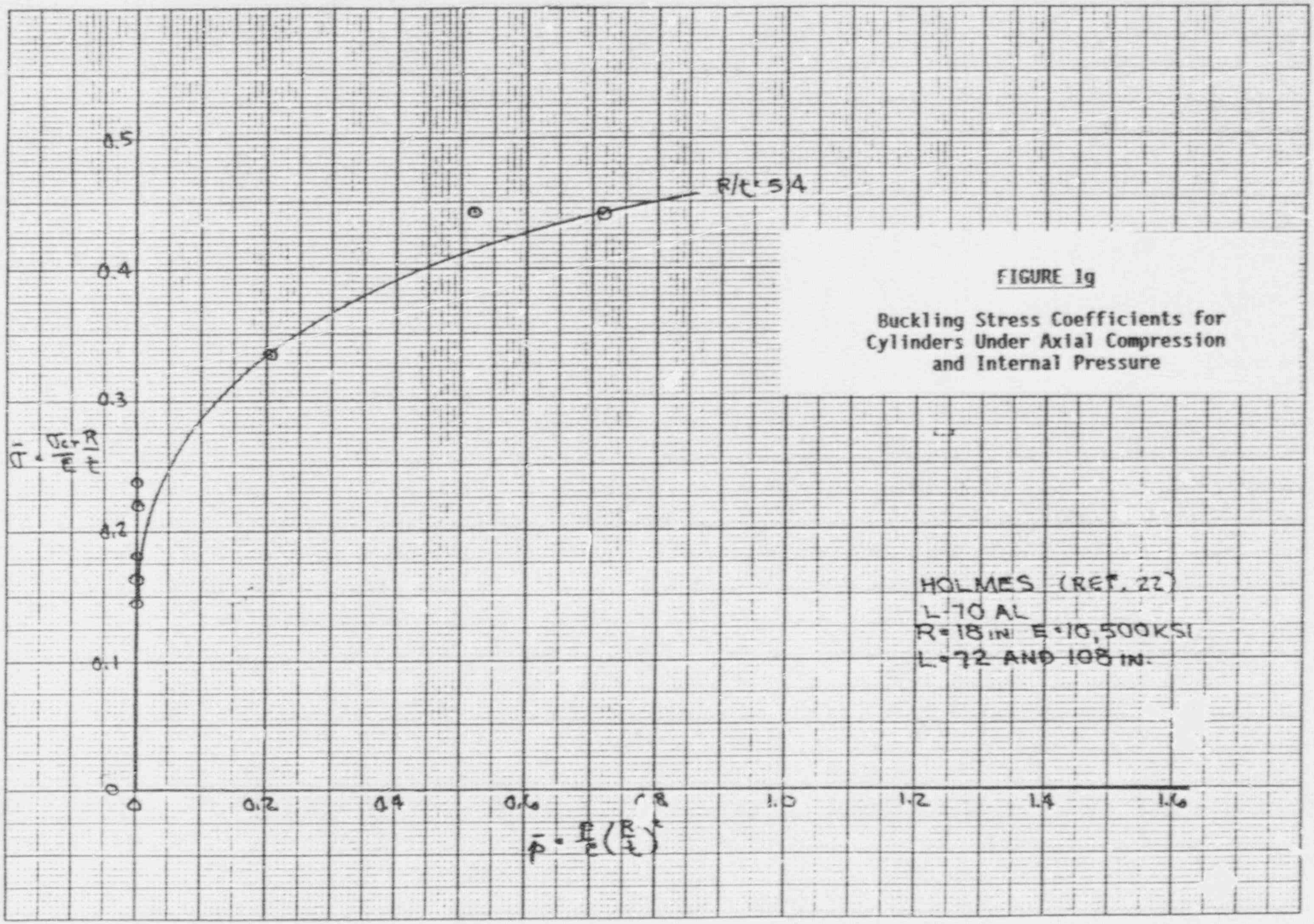

FIGURE 1g

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

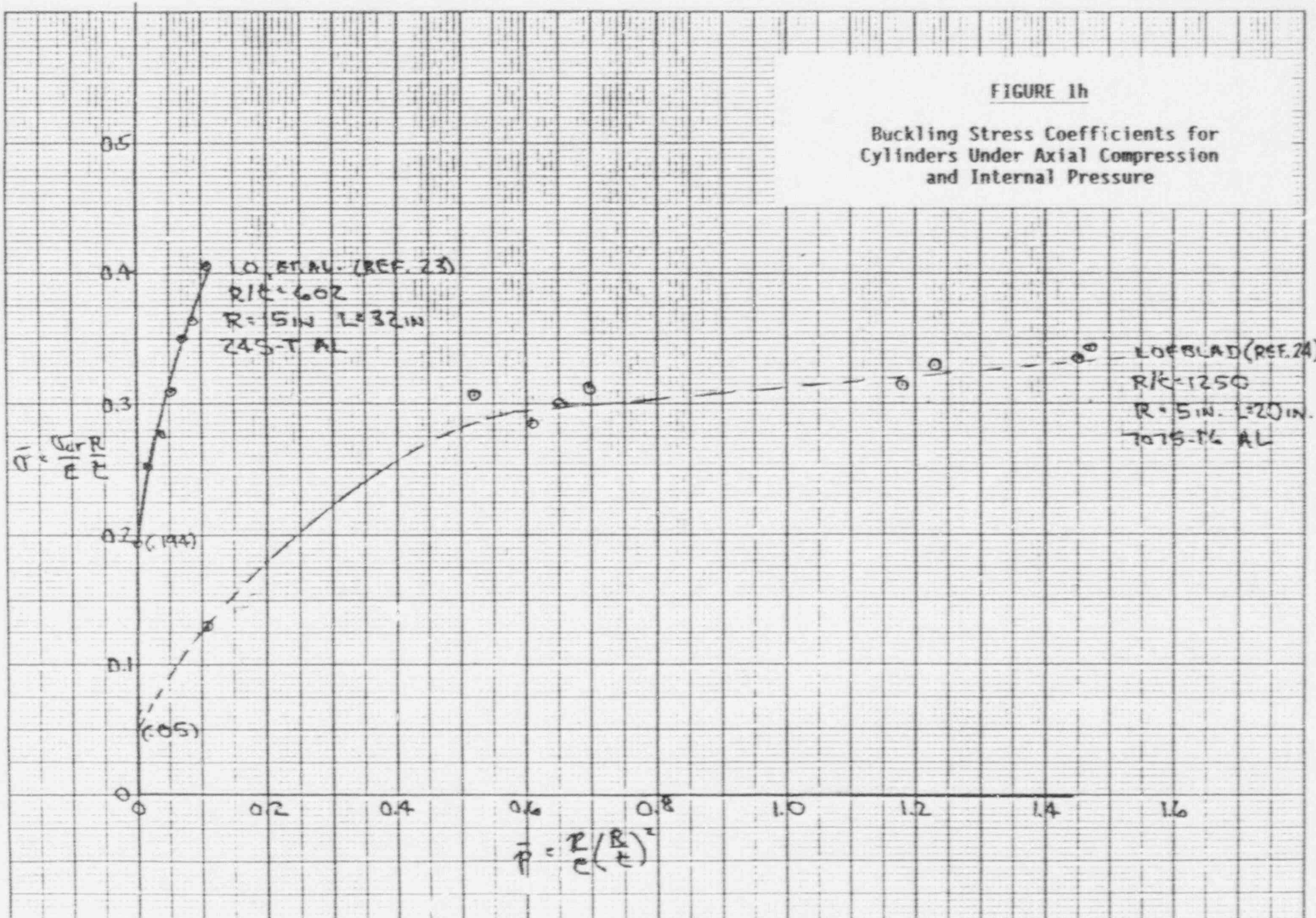

FIGURE 1h

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

FIGURE 11

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

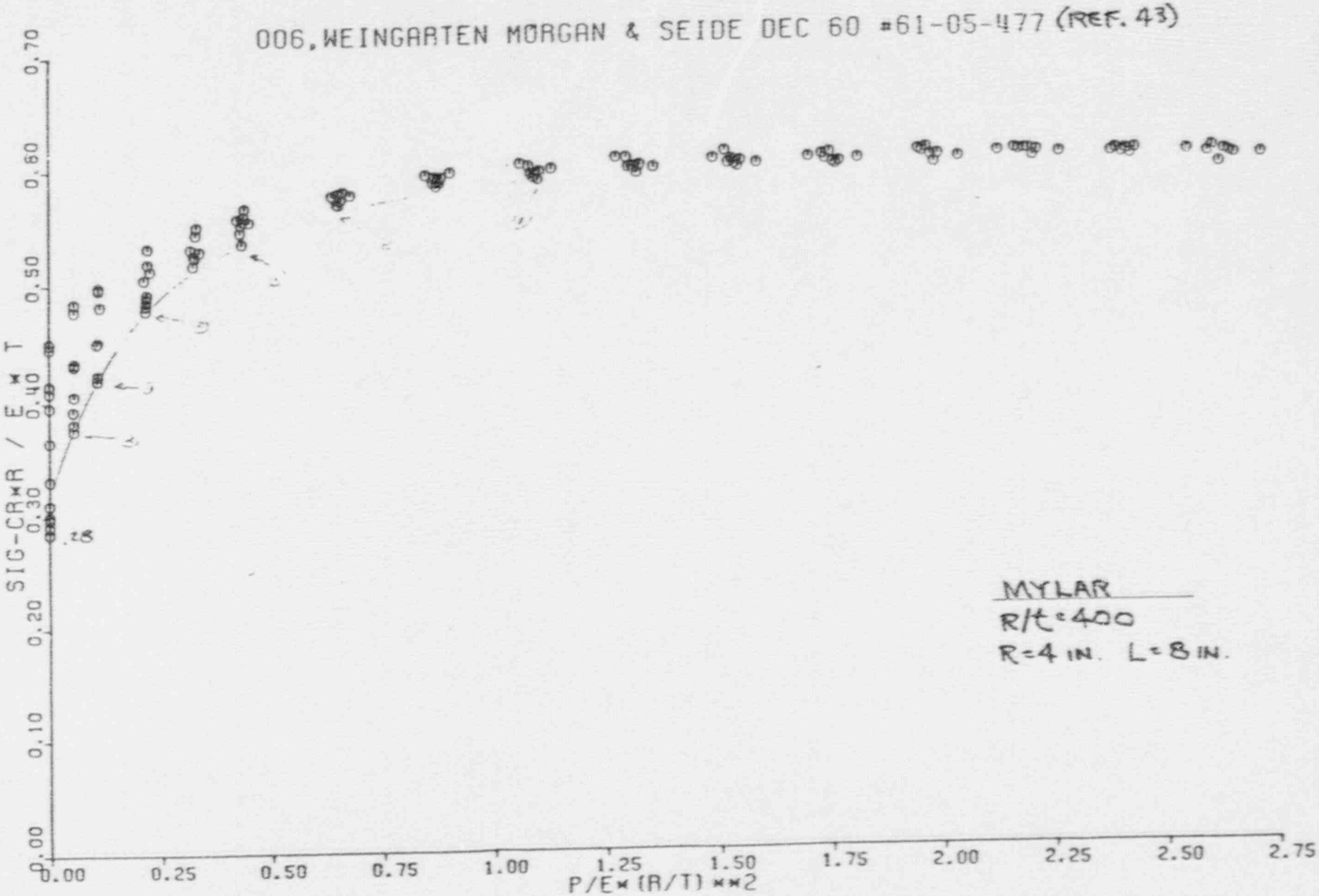

FIGURE 1j

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

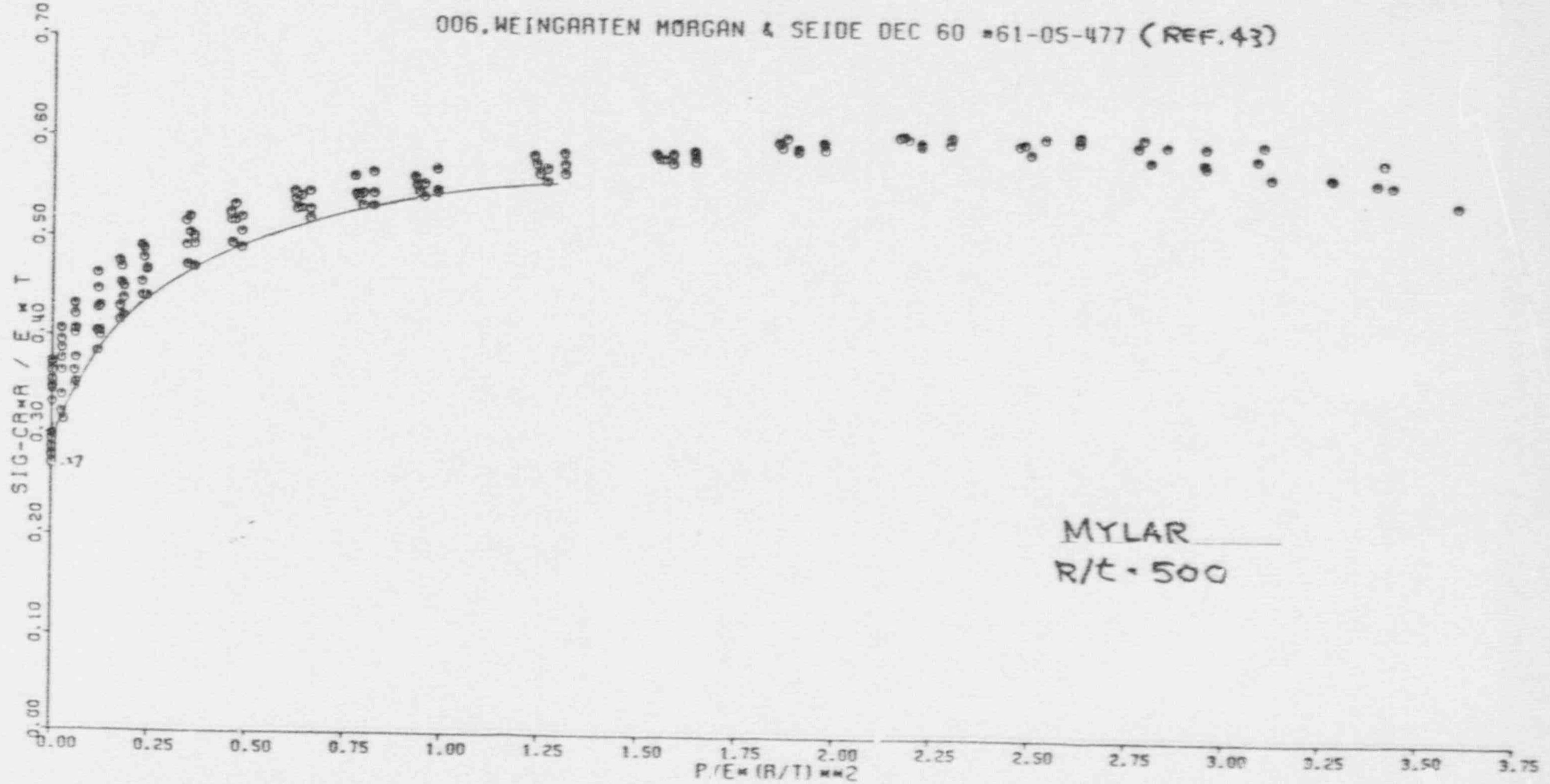

FIGURE 1k

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

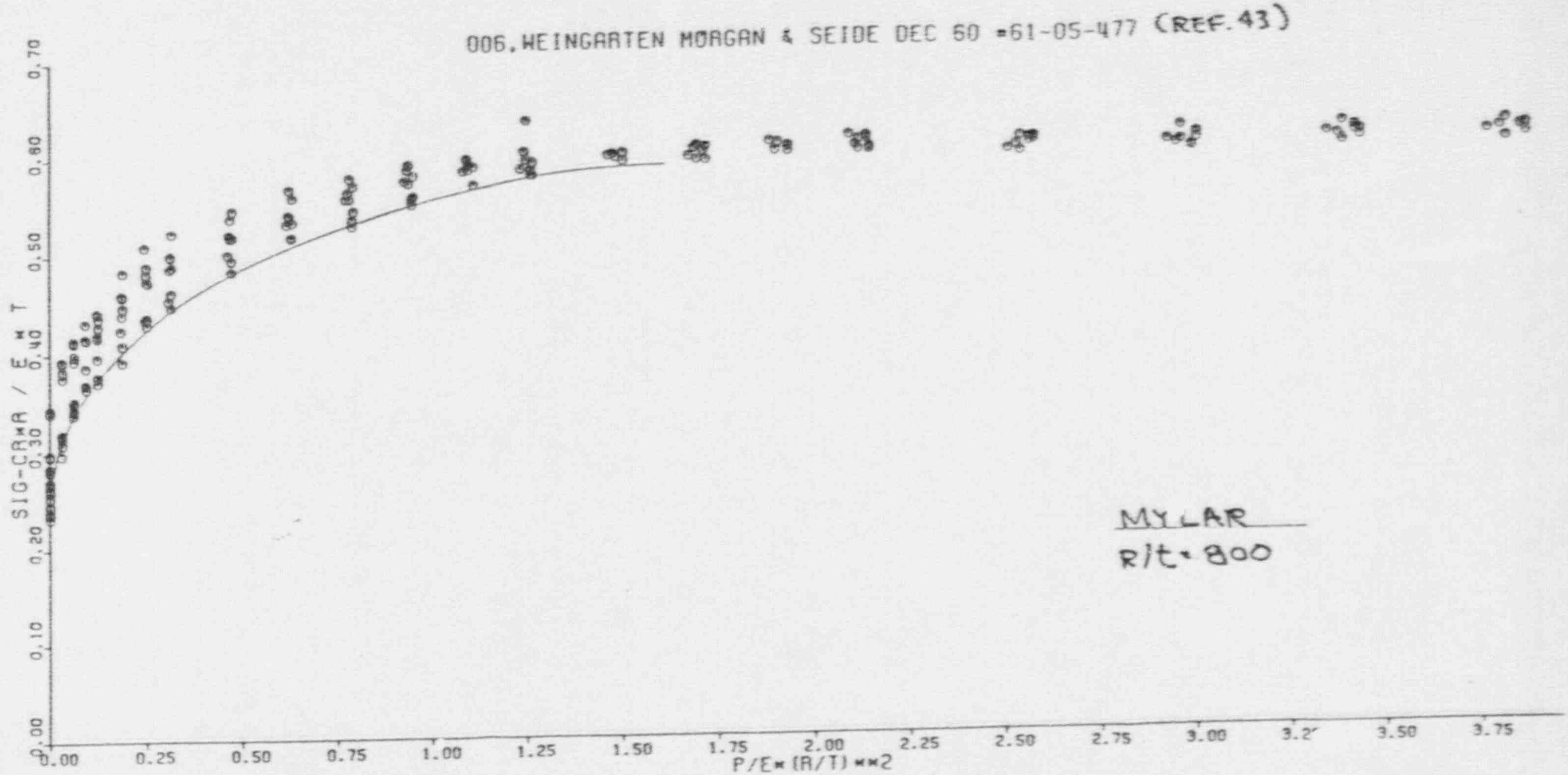

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FIGURE 11

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

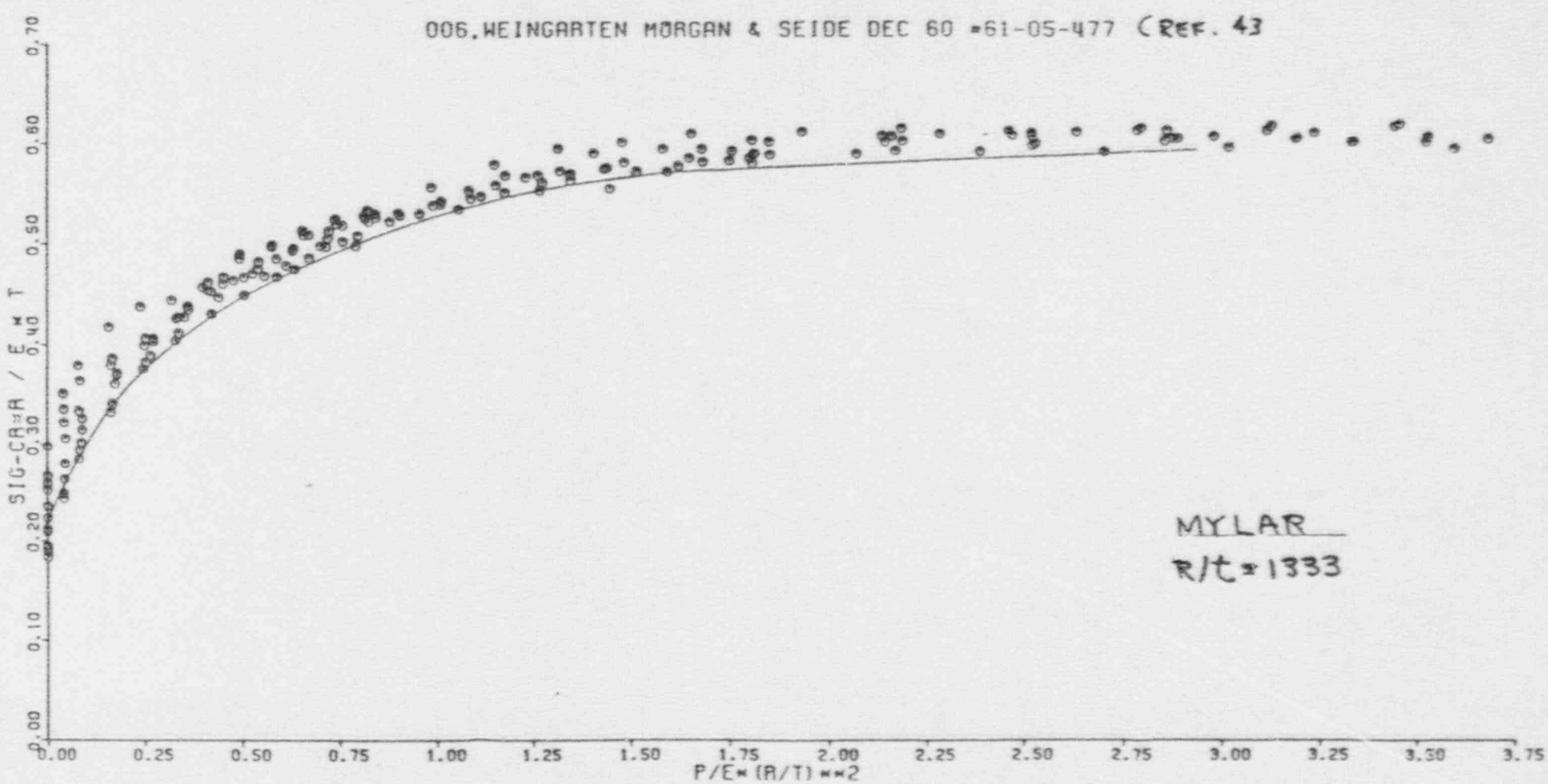


FIGURE 1m

Buckling Stress Coefficients for Cylinders Under Axial Compression and Internal Pressure

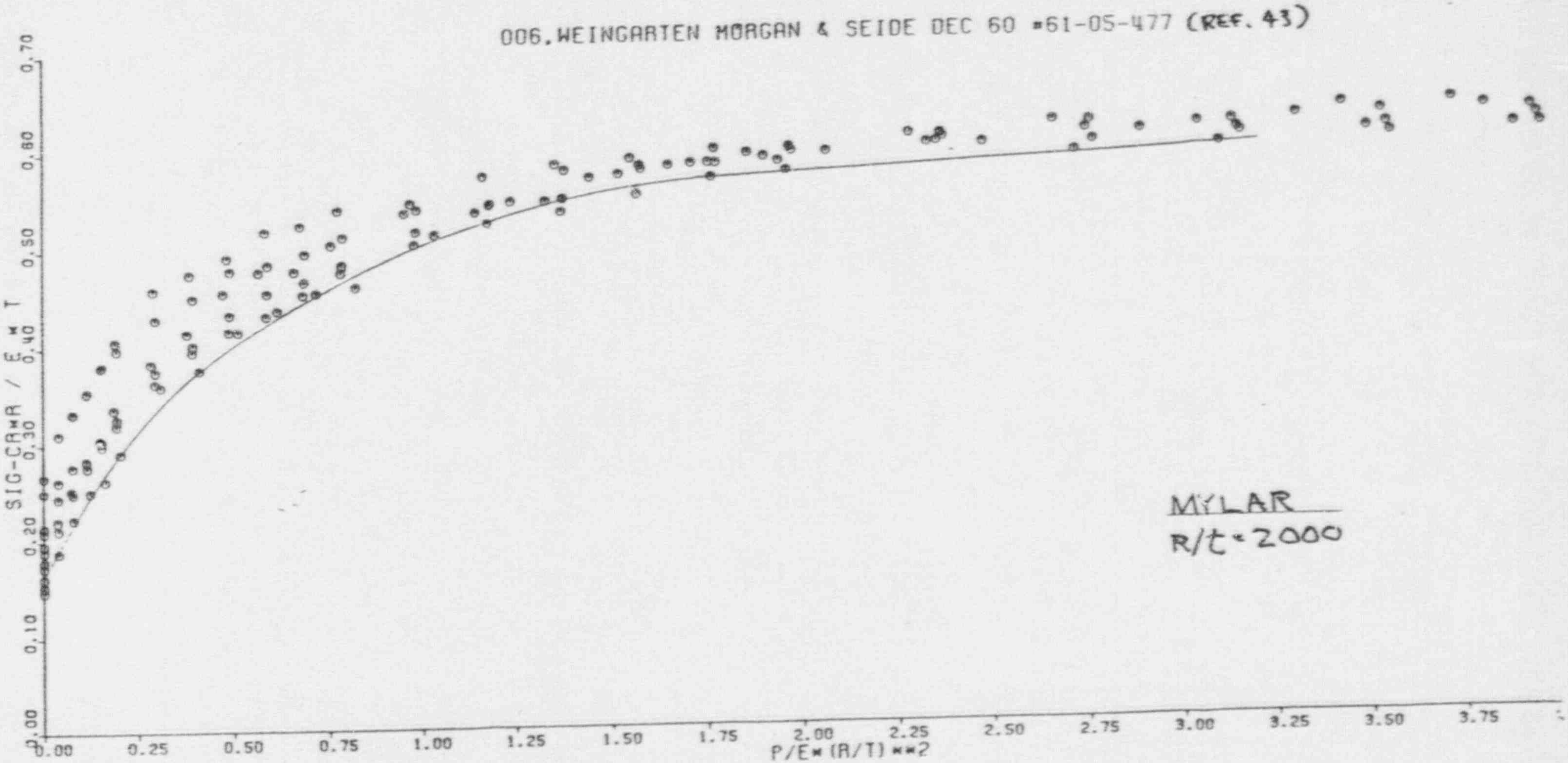
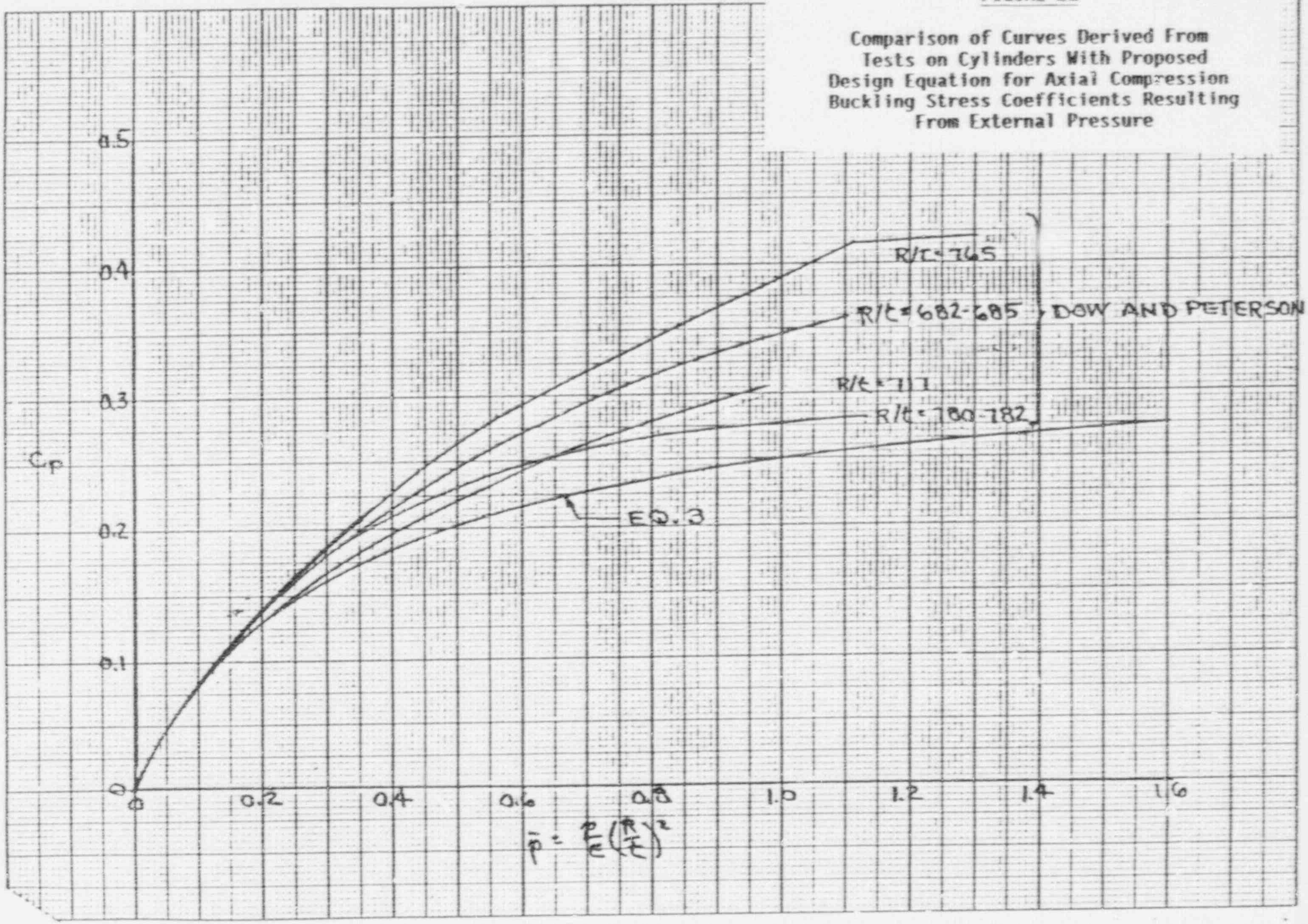


FIGURE 2a

Comparison of Curves Derived From Tests on Cylinders With Proposed Design Equation for Axial Compression Buckling Stress Coefficients Resulting From External Pressure



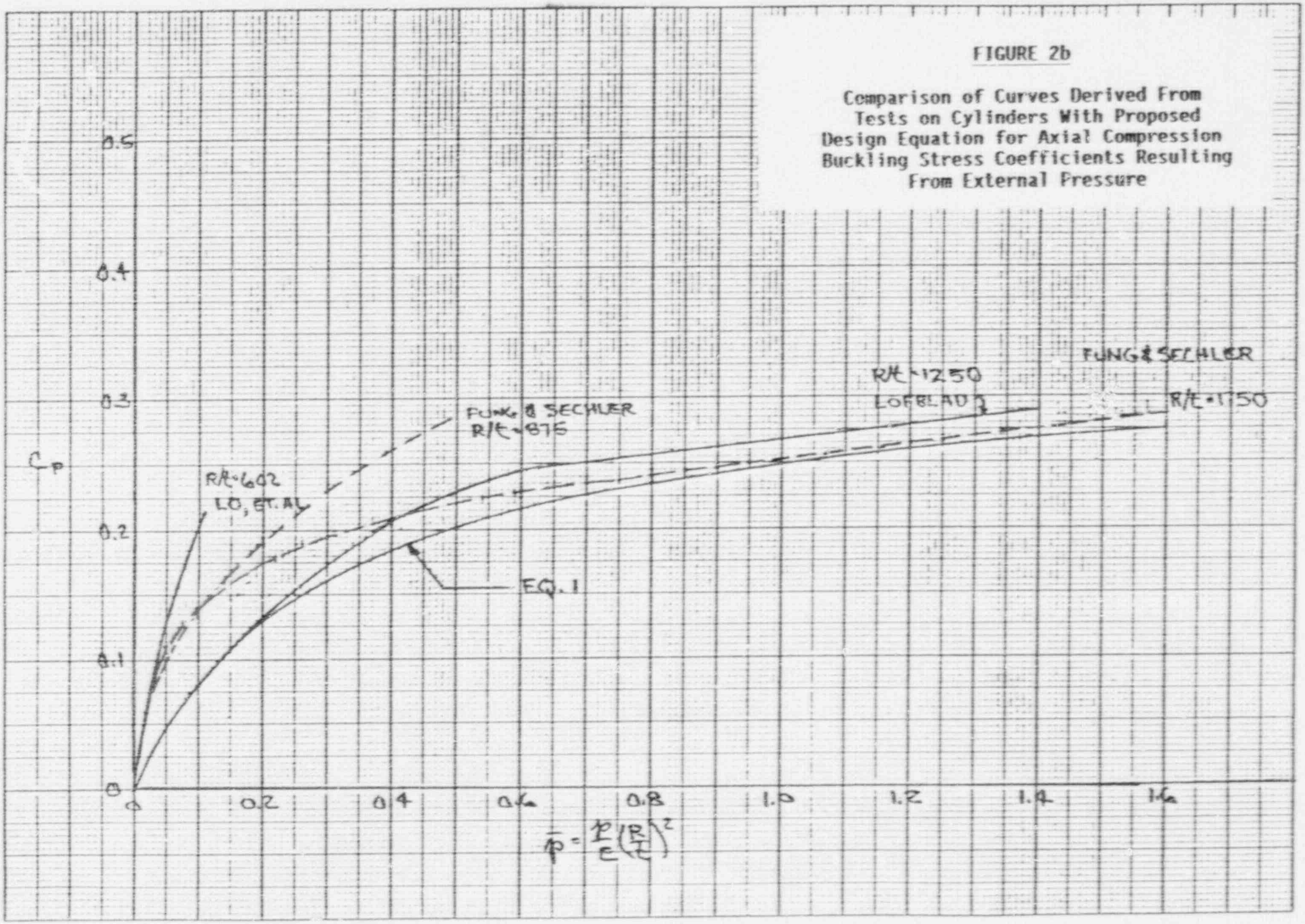


FIGURE 2b

Comparison of Curves Derived From Tests on Cylinders With Proposed Design Equation for Axial Compression Buckling Stress Coefficients Resulting From External Pressure

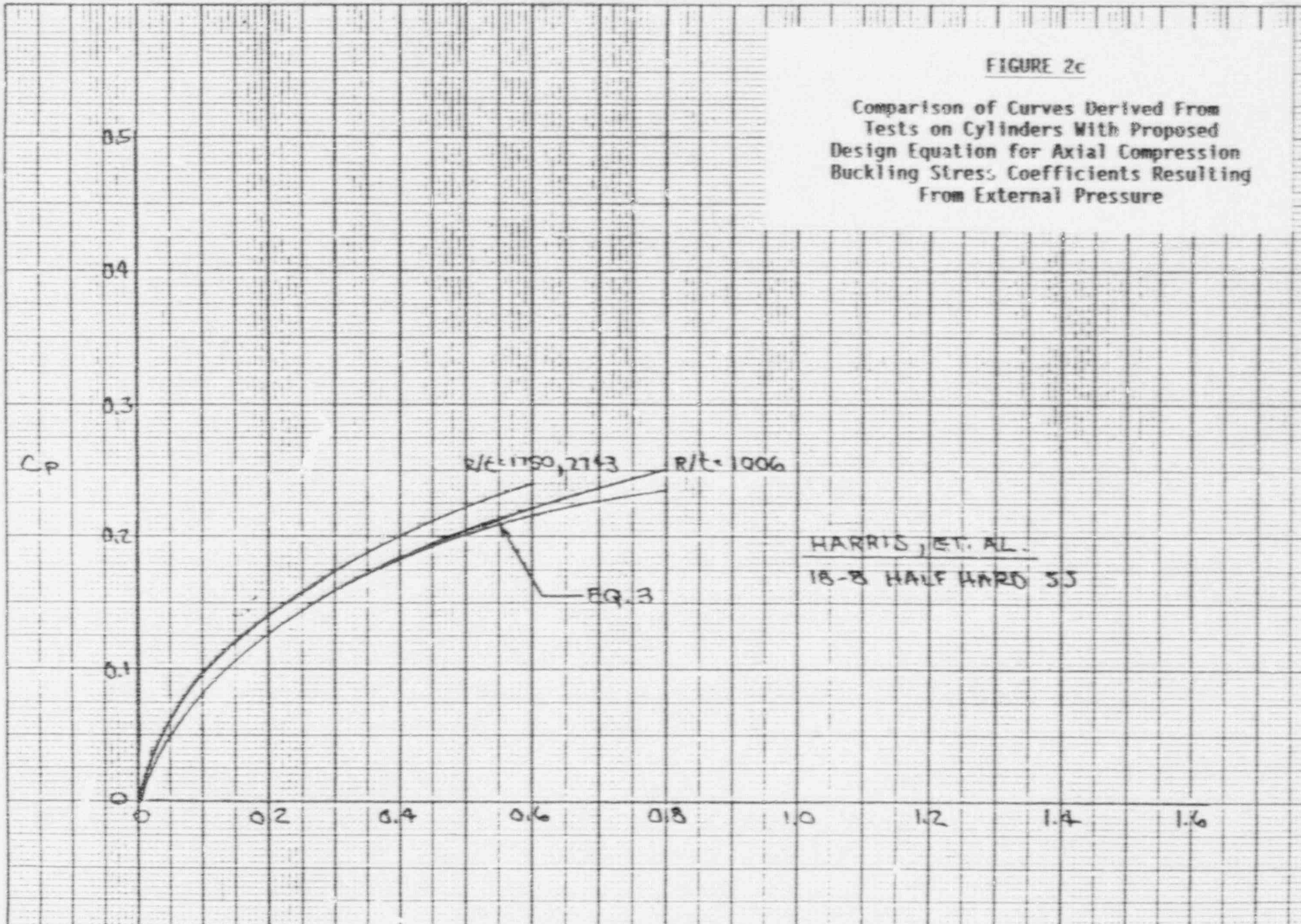


FIGURE 2c

Comparison of Curves Derived From Tests on Cylinders With Proposed Design Equation for Axial Compression Buckling Stress Coefficients Resulting From External Pressure

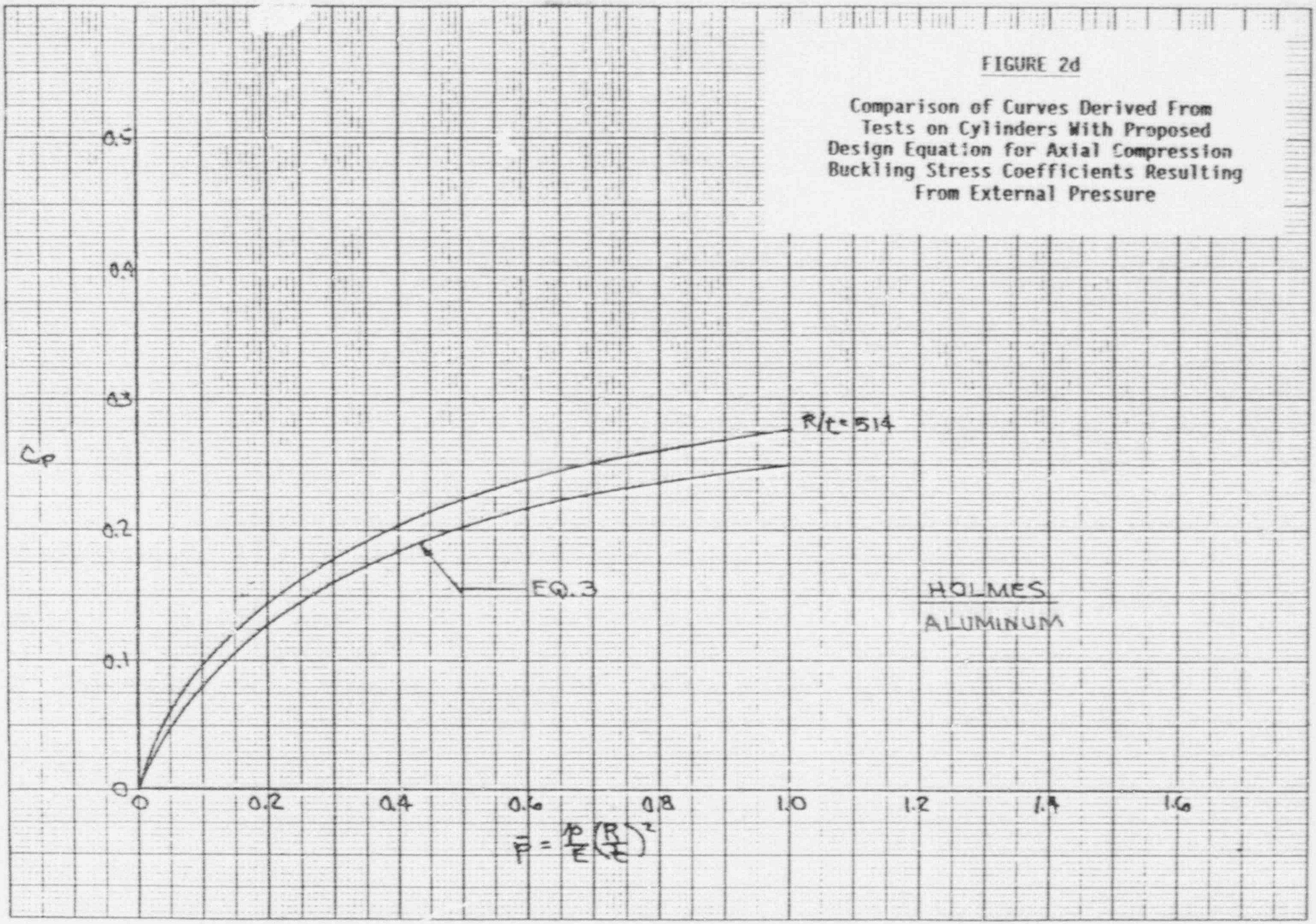


FIGURE 2d

Comparison of Curves Derived From Tests on Cylinders With Proposed Design Equation for Axial Compression Buckling Stress Coefficients Resulting From External Pressure

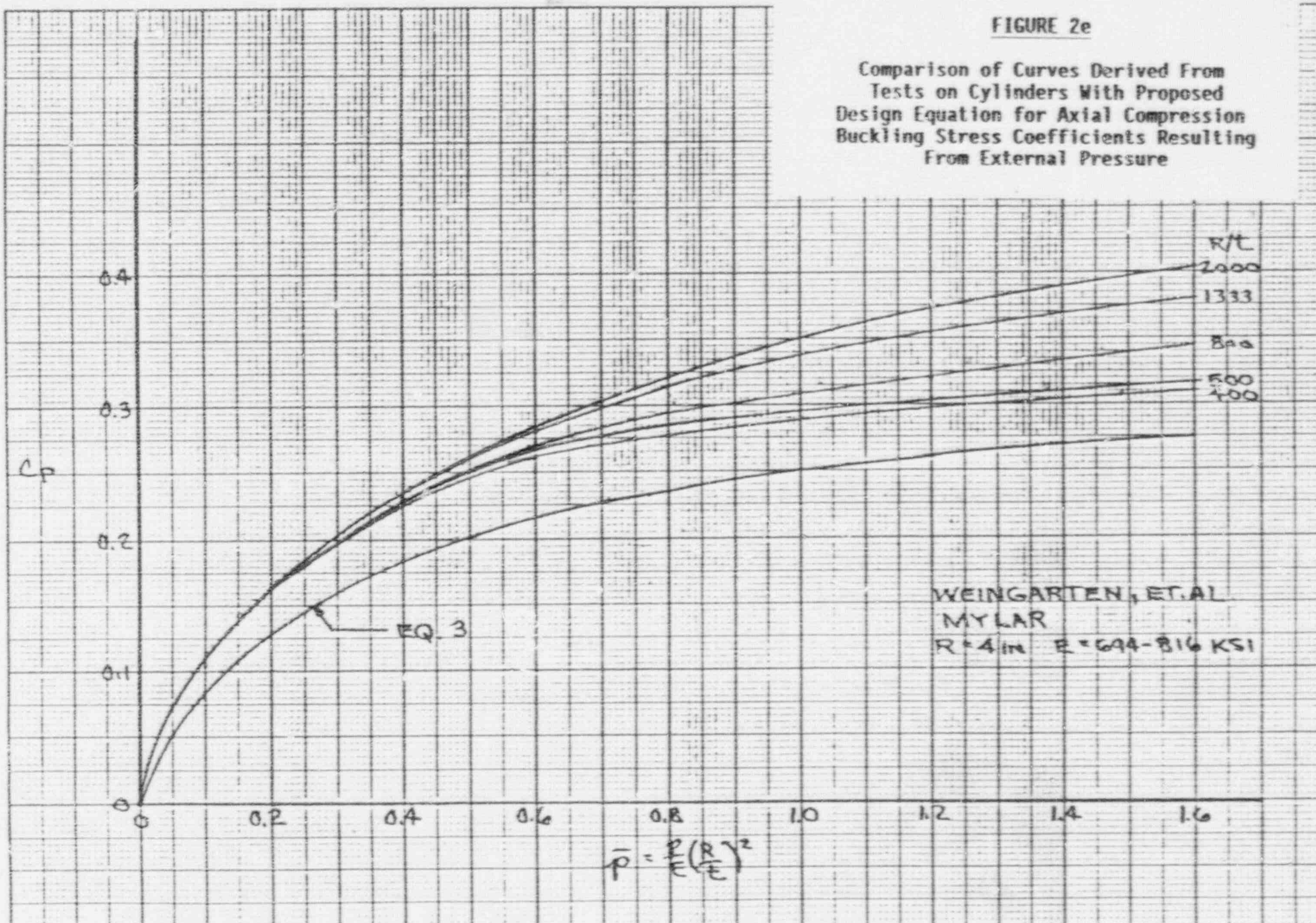
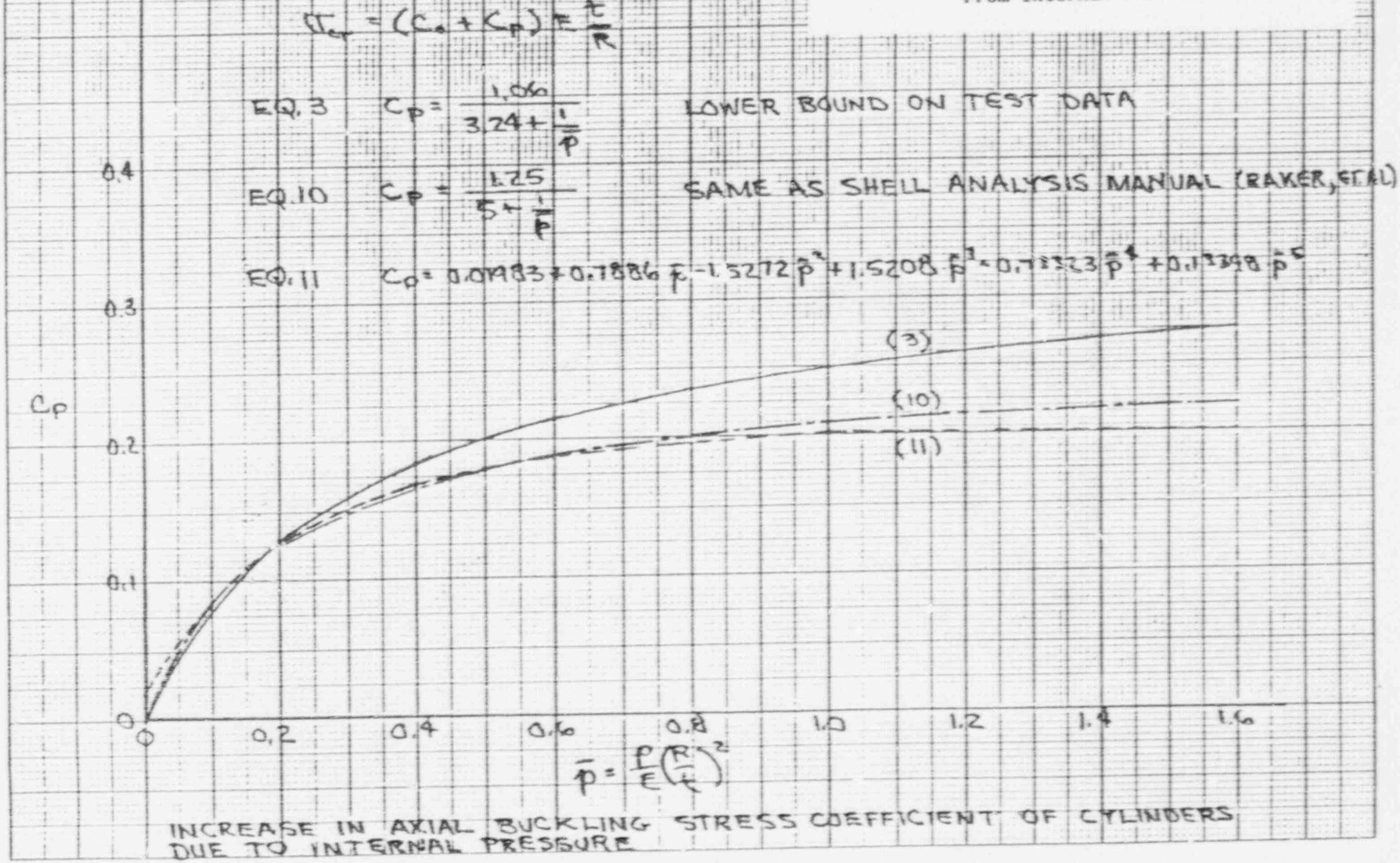


FIGURE 2e

Comparison of Curves Derived From Tests on Cylinders With Proposed Design Equation for Axial Compression Buckling Stress Coefficients Resulting From External Pressure

FIGURE 3

Comparison of Design Equations for
Axial Compression Buckling Stress
Coefficients for Cylinders Resulting
From Internal Pressure



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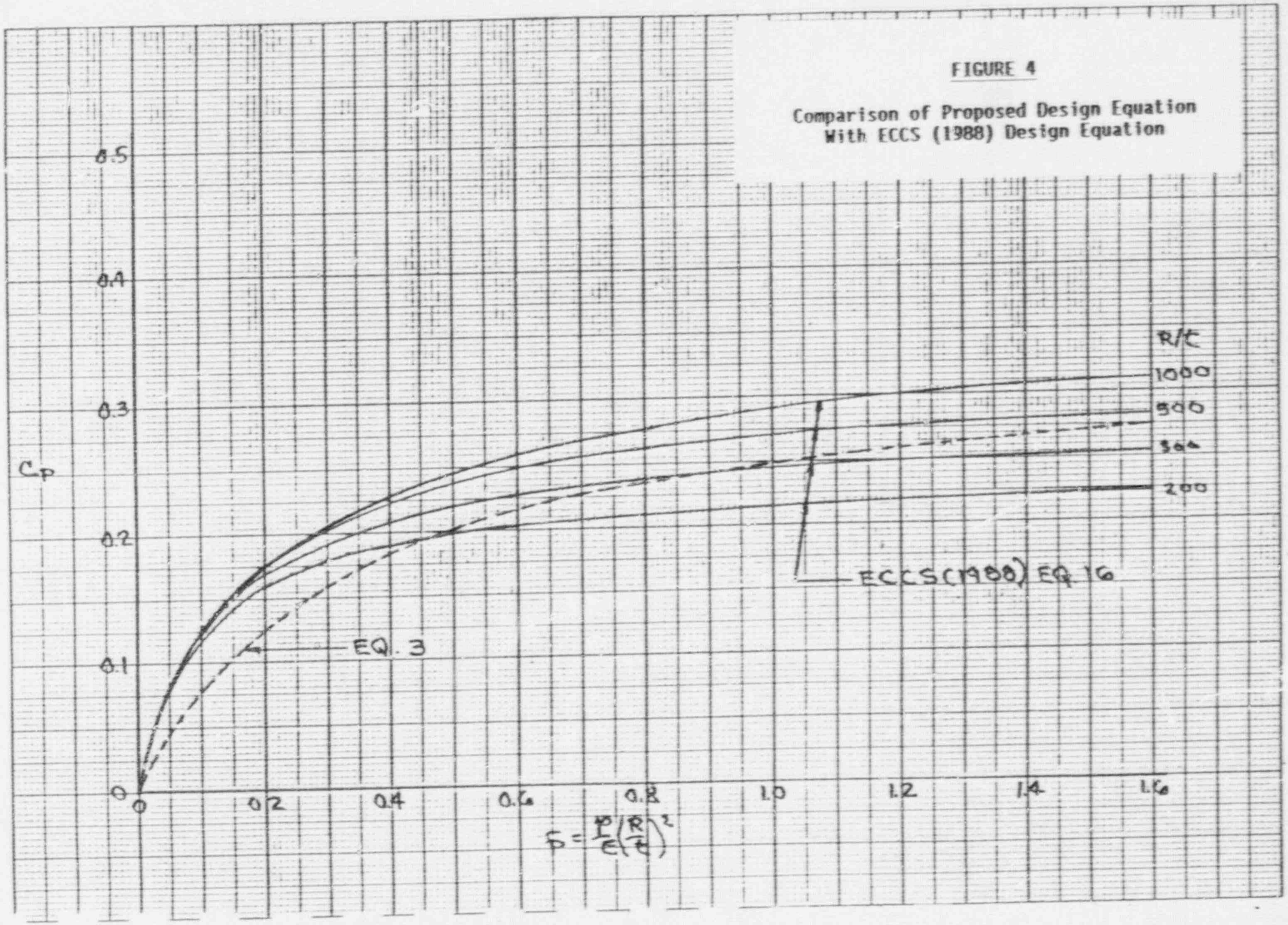

FIGURE 4

Comparison of Proposed Design Equation With ECCS (1988) Design Equation

7. Page 2-4, Section 2.4

This section states that Reference 2-6 was used to calculate the plasticity reduction factor for the meridional direction elastic buckling stress. Since this approach apparently has not been incorporated into Code Case N-284, the sensitivity of the results should be studied by using other methods which address this effect.

Response:

Using the approach outlined in Code Case N-284, (Ref. 2-1 of the GE Report) the calculated value of the plasticity reduction factor is 1.0, i.e., no reduction need be made in the calculated value of elastic buckling stress. The approach outlined in Reference 2-6 takes a more conservative approach resulting in lower plasticity reduction factors. This more conservative approach is the one used in the analysis (Ref. 2-6).

8. <u>Page 3-3, second paragraph</u>

For the stability analysis the stiffness for the sandbed was assumed to be 366 psi/inch and no sensitivity studies are reported. As described in Question 5, the results of the stability analysis with the sand removed should be provided.

<u>Response:</u>

GE Report index 9-4 entitled "An ASME Section VIII Evaluation of Oyster Creek Drywell for Without Sand Case Part 2 Stability Analysis" submitted March 4, 1991 demonstrates Code compliance without sand. Therefore, sensitivity of buckling capacity to variations of the sand spring stiffness is not a concern.

9. <u>Page 3-6, Section 3.5.[illegible]</u>

The first sentence states that "the [illegible] psi external pressure load for the refueling case is applied to the external faces of all of the drywell and vent shell elements. Unless it can be demonstrated that this pressure actually is present at all times during normal operation and refueling, the effect on the buckling analysis results of assuming no external pressure for these two load cases should be reported. Furthermore, is it possible to have an external pressure greater than 2 psi on the drywell shell? If so, an enveloping pressure case should be considered in the analysis.

<u>Response</u>:

The 2 psi external pressure results in a compressive stress in the shell in the sandbed region of PR/2T = (2 * 420)/(2* 0.736) = 570 psi which is small compared to the total applied from other sources. The 2 psi is the plant design basis and it is the enveloping pressure. Regardless of the cause of differential pressure between the reactor building and the drywell, the maximum differential pressure is 1.0 psid. There are two 20-inch vacuum breakers between the reactor building and the torus. These are set to to open at 0.5 psid. Further, there are 14 18-inch vacuum breakers between the torus and the drywell. These are also set to open at 0.5 psid. The maximum differential pressure is arrived at by adding these two setpoints.

Part III - General

10. Justification for the use of ASME Section III, Subsection NE has been provided to evaluate the Oyster Creek Steel drywell; taking into consideration design, materials, fabrication, inspection and testing with exception of the comments indicated above, the justification appears to be reasonable. Since the present-day quality assurance and quality control requirements for the design and construction of nuclear power were in the formative stage at the time when the Oyster Creek Plant was designed and constructed, indicate what quality assurance and quality control programs were implemented for the Oyster Creek drywell. Indicate if documentation of the program is available.

Response:

The quality assurance and quality control programs implemented for the Oyster Creek drywell are available and documented in the original (docketed) Oyster Creek FDSAR Amendments 15, 27, and 39.

Amendment 15, Section IV, provides the containment shell fabrication, erection and testing history. Section VI provides details on the fabrication and installation of the containment penetrations and nozzles.

Amendment 27 provides the Plant Quality Assurance Report with Amendment 39 providing expanded information on organizational structures and responsibility for quality efforts by organizational components and subcontractors. Amendment 39 describes Burns & Roe's subcontractor responsibilities a part of which resulted in the issuance of B&R Specification S-2299-4 to Chicago Bridge & Iron for the containment construction. Section II of this specification (copy attached) provides additional specific containment quality requirements consistent with the above mentioned FDSAR Amendments.

SUPPLEMENT TO RESPONSE NO. 10

Oyster Creek Burns and Roe Specification S-2299-4, Section II (Addendum No. 5), dated June 18, 1964.