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APPROPRIATE NOMINAL STRESSES FOR USE WITH ASME CODE PRESSURE-LOADING STRESS INDICES FOR NOZZLES

E. C. Rodabaugh

Work funded by the Nuclear Regulatory Commission
Under Interagency Agreement 40-495-75

June, 1976

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STRESS INDICES FOR NOZZLES

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TABLE OF CONTENTS

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TABLE OF CONTENTS (Continued)

FOREWORD

The work reported here was performed for the Oak Ridge National Laboratory at Battelle-Columbus Laboratories under Union Carbide Corporation, Nuclear Division, Subcontract No. 2913 as part of the ORNL Design Criteria for Piping and Nozzles Program, S. E. Moore, Manager. This program is funded by the Office of Nuclear Regulatory Research of the U. S. Nuclear Regulatory Commission, Division of Reactor Safety Research (RSR) as part of a cooperative effort with industry to develop and verify analytical methods for assessing the safety of nuclear pressure-vessel and piping-system design. The cooperative effort is coordinated through the Design Division, Pressure Vessel Research Committee, Welding Research Council. The cognizant RSR-NRC project engineer is E. K. Lynn.

The study of nominal stresses and stress indices described in this report is part of a continuing study of design rules for nozzles in pressure vessels being coordinated by the PVRC Subcommittee on Reinforced Openings and External Loadings, W. L. Greenstreet, Chairman. Results from these studies are used by appropriate ASME Code groups in drafting new and improved design rules.

Other phase reports in this series are:

No.	Title and Date of Issue or Status
117-1	"Elastic Stresses in Nozzles in Pressure Vessels with Internal Pressure Loading", April, 1969.
117-2	"Additional Data on Elastic Stresses in Nozzles in Pressure Vessels with Internal Pressure Loading", December, 1971.
117-3	"Proposed Alternate Rules for Use in ASME Codes", August, 1969.
117-4	"Comparison of Finite-Element Analysis and Experimental Stresses for a Nozzle in a Spherical Shell, Model N-1A", August, 1969.
117-5	"Review of Service Experience and Test Data on Openings in Pressure Vessels with Nonintegral Reinforcing", March, 1970 (WRC Bulletin 166, October, 1971).

117-6R "Elastic Stresses in Nozzles in Spherical Pressure Vessels with Pressure and Moment Loading", August, 1972.

117-7 "Inside Versus Outside Reinforcing of Nozzles in Spherical Shells with Pressure Loading", January, 1974.

117-8 "Applicability of Axisymmetric Geometry Analytical Methods to Nozzles in Cylindrical Shells with Internal Pressure Loading", July, 1973.

117-9 "Elastic Stresses at Reinforced Nozzles in Spherical Shells with Pressure and Moment Loading", January, 1976.

117-10 "Preliminary Evaluation of Closely Spaced Reinforced Openings in a Cylindrical Pressure Vessel Under Internal Pressure Loading", June 30, 1975.

117-11 "Exploratory Study of the Feasibility of Developing Stress Indices for Thermal Gradients in Pressure Vessel Nozzles", May 12, 1975.

117-12 "Evaluation of Available Data on Limit Moments of Branch Connections", Draft dated May 1, 1975. Anticipated completion in FY 1976A.

INTRODUCTION

Stress indices for nozzles in pressure vessels are given in Tables NB-3338.2(c)-1 and NB-3339.7-1 for paragraphs NB-3338 and NB-3339 of the Code*, respectively. Table NB-3686.1-1, in the Piping Design portion of the Code, is almost the same as Table NB-3338.2(c)-1. These three tables are included herein as Tables 1, 2, and 3.

The intended use of the stress index tables is indicated on NB-3331(b): i.e., if the rules of NB-3222.4(d) are not met, and, therefore, a fatigue analysis is required by the Code, the requirements of NB-3222.4(b), "Peak Stress Intensity", may be evaluated by application of the stress index method of NB-3338 or NB-3339. For a fatigue analysis, we wish to obtain the stress range for a given pressure range.

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* The term "Code" used herein refers to the ASME Boiler and Pressure Vessel Code, Section III-division 1--1974 edition, "Nuclear Power Plant Components". References to portions thereof are identified as in the Code.

TABLE 1. STRESS INDICES FROM NB-3338

TABLE NB-3338.2(c)-1
STRESS INDICES FOR NOZZLES

Nozzles in Spherical Shells and Formed Heads		
Stress	Inside Corner	Outside Corner
σ_n	2.0	2.0
σ_t	−0.2	2.0
σ_r	$-2t_n/R$	0
S	2.2	2.0

Nozzles in Cylindrical Shells				
	Longitudinal Plane		Transverse Plane	
Stress	Inside	Outside	Inside	Outside
σ_n	3.1	1.2	1.0	2.1
σ_t	−0.2	1.0	−0.2	2.6
σ_r	$-t_n/R$	0	$-t_n/R$	0
S	3.3	1.2	1.2	2.6

σ_t = the stress component in the plane of the section under consideration and parallel to the boundary of the section

σ_n = the stress component normal to the plane of the section (ordinarily the circumferential stress around the hole in the shell)

σ_r = the stress component normal to the boundary of the section

* S_m = the stress intensity (combined stress) at the point under consideration

* Presumably this should be S.

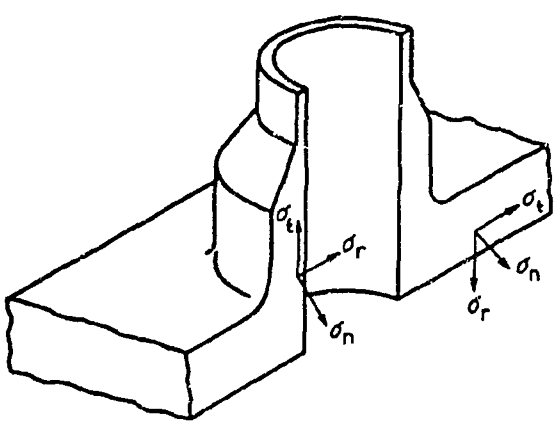


FIG. NB-3338.2-1 DIRECTION OF STRESS COMPONENTS

TABLE 2. STRESS INDICES FROM NB-3339

TABLE NB-3339.7-1
STRESS INDICES FOR INTERNAL PRESSURE LOADING

(a) Nozzles in Spherical Shells and Spherical Heads

Stress	Inside	Outside
σ_n	2.0 (d/D)	2.0 (d/D)
σ_t	0.2	2.0 (d/D)
σ_r	$4T/(D+T)$	0
σ	larger of 2.2 (d/D) or 2.0 + $[4T/(D+T)]$ (d/D)	2.0 (d/D)

(b) Nozzles in Cylindrical Shells

Stress	Longitudinal Plane		Transverse Plane	
	Inside	Outside	Inside	Outside
σ_n	3.1	1.2	1.0	2.1
σ_t	0.2	1.0	−0.2	2.6
σ_r	$-2T/(D+T)$	0	$-2T/(D+T)$	0
$\bar{\sigma}$	3.3	1.2	1.2	2.6

(a) The term stress index, as used herein, is defined as the numerical ratio of the stress components, σ_t, σ_n, and σ_r under consideration to the computed stress, S.

(b) The symbols for the stress components are shown in Fig. NB-3338.2-1, and are defined as follows:

$S = P(D+T)/4T$ for nozzles in spherical vessels or heads

$S = P(D+T)/2T$ for nozzles in cylindrical vessels

σ_t = the stress component in the plane of the section under consideration and parallel to the boundary of the section

σ_n = the stress component normal to the plane of the section (ordinarily the circumferential stress around the hole in the shell)

σ_r = the stress component normal to the boundary of the section

* σ = the stress intensity (combined stress) at the point under consideration

* Should be $\bar{\sigma}$.

D = inside diameter, in corroded condition, of cylindrical vessel, spherical vessel or spherical head, in.

T = wall thickness of vessel or head, computed by the equation given in NB-3324.1 for cylindrical vessels, by NB-3324.2 for spherical vessels or spherical heads, in.

P is not defined. It should be changed to P_o (in the definition of S) and defined as

P_o = range of operating pressure in cycle under consideration.

TABLE 3. STRESS INDICES FROM NB-3686

TABLE NB-3686.1-1
BRANCH CONNECTIONS WITH RESTRICTIONS
GIVEN IN NB-3686, INTERNAL PRESSURE

(a) Branch Connections in Pipe

Stress	Stress Index, i			
	Longitudinal Plane		Transverse Plane	
	Inside	Outside	Inside	Outside
σ_n	3.1	1.2	1.0	2.1
σ_t	−0.2	1.0	−0.2	2.6
σ_r	$-t_r/R_r$ *	0	$-t_r/R_r$ *	0
σ	3.3	1.2	1.2	2.6

* Should be R_m

(b) Branch Connections in Formed Heads

Stress	Stress Index, i	
	Inside Corner	Outside Corner
σ_n	2.0	2.0
σ_t	−0.2	2.0
σ_r	$-2t_h/R_h$	0
σ	2.2	2.0

See Figure NB-3684.1-1 for definitions at stress components, σ_n, σ_t, σ_r, and σ.

R_m = mean radius of run pipe, in.
R_h = mean radius of formed head in the vicinity of the branch connection, in.
t_r = minimum required thickness of run pipe, calculated as a plain cylinder
t_h = minimum required thickness of formed head, calculated as a spherical shell of inside radius, R_h
P = internal pressure, psi

NB-3686.3 Stresses from Stress Indices

(a) For branch connections in pipe, multiply stress indices by:

$$\frac{PR_r}{t_r} \; *$$

* Should be R_m.

(b) For branch connections in formed heads, multiply stress indices by:

$$\frac{PR_h}{2t_h}$$

P should be changed to P_o (three places) and defined as

P_o = range of operating pressure in cycle under consideration.

A problem in interpretation and use of the stress indices arises when part or all of the Code-required reinforcement is obtained by shell-thickening; i.e., by making the actual thickness t_a sufficiently larger than the required thickness t_r so that all reinforcement is available in the shell. The Code includes two portions (NB-3338 and NB-3683.2) which use the actual shell thickness t_a for computing the nominal-stress-multiplier; two other portions (NB-3339 and NB-3686.3) which use the required minimum wall thickness t_r. It appears, from available data, that using t_a can be significantly unconservative for prediction of maximum stresses; whereas using t_r is always conservative in predicting maximum stresses but may be overly conservative.

It must be emphasized that the stress indices represent a gross simplification of a highly complex problem. The problem involves nozzles and openings in any kind of shell (cylindrical, spherical, ellipsoidal, etc) with D/t up to 100 and d/D up to 0.5. The load reinforcement can be of any shape and, as one bound, there need not be any local reinforcement. Variation of these parameters (D/t, d/D, shape and amount of local reinforcement) lead to variations in the stress per unit pressure at all locations and directions. The stress indices are intended to represent maximum stresses at various locations and directions due to internal pressure for any and all of the infinite variety of parameters.

In the present report, the background of the stress index method is briefly discussed. The reinforcement requirements are crucial to an understanding of the applicability of the stress indices and are briefly reviewed. The report then presents data relevant to the maximum stress index and a discussion of that data.

The report deals principally with one bound of the overall problem; the maximum stress for nozzles or openings with <u>no</u> local reinforcement. It is not clear what additional work, if any, is needed to solve the more general problem. This aspect is discussed in the Recommendations section of this report. The information available suggests that the present Code rules should be changed and specific recommendations are made even though, as a result of further work, these recommendations may subsequently need to be modified.

NOMENCLATURE

D = inside diameter of shell

d = inside diameter of opening in a shell

D_m = mean diameter of shell

I = a stress index

P_d = design pressure

P_o = pressure range for cycle under consideration

r_1, r_2, = radii; see Figure 1

$\rho = (d/D)\sqrt{D/t_a}$

S_m = allowable stress intensity

S_n = nominal stress, $\sigma = I \times S_n$

$s/S = (d/D)\ (t_a/t_n)$ for nozzles in cylindrical shells
$\quad = 2(d/D)\ (t_a/t_n)$ for nozzles in spherical shells

σ = stress range due to pressure range

t_a = actual (nominal) wall thickness of shell

t_n = actual (nominal) wall thickness of nozzle

t_r = required minimum wall thickness of shell

Additional definitions are given in Tables 1, 2, 3, and 4. These Code definitions are not always consistent with each other and the Report uses the definitions listed above.

The Symbol I, as used in this report, is equivalent to σ_n/S_n, σ_t/S_n, σ_r/S_n or $\bar{\sigma}/S_n$, where S_n is the nominal stress, σ_n, σ_t and σ_r are stress components, as defined in Table 1 herein and $\bar{\sigma}$ is the stress intensity, identified in various portions of the Code by the symbol $S(S_m?)$, $\bar{\sigma}$ and σ. The appropriate definition of the nominal stress S_n is the main subject of this report.

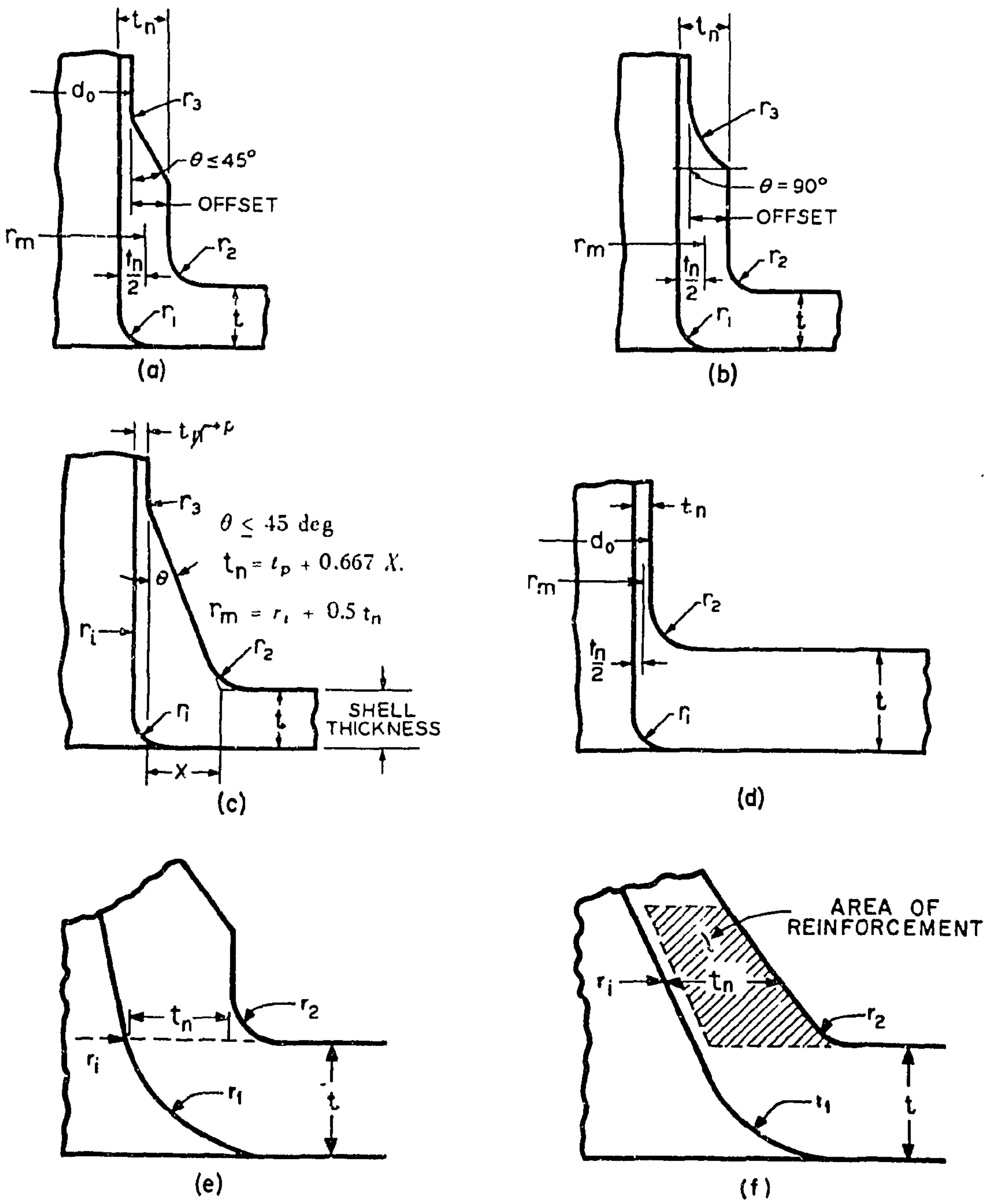


FIG. NB-3338.2-2 NOZZLE DIMENSIONS

FIGURE 1. NOZZLE DIMENSIONS

BACKGROUND OF THE STRESS INDEX METHOD

Definition of a Stress Index

A stress index is a nondimensional ratio, σ/S_n, where σ is an actual stress magnitude and S_n is a nominal stress. If the stress index is identified by the symbol I, then the actual stress is given by:

$$\sigma = I \times S_n \tag{1}$$

Values of I can be established for:

(a) A class of geometrical configuration; e.g., nozzles in cylindrical shells

(b) A range of parameters for that class of geometry; e.g., $D/t \leq 100$, $d/D \leq 0.5$ for nozzles in cylindrical shells

(c) Some particular loading; e.g., internal pressure.

The values of I can be established either by tests, theoretical analysis or combinations thereof. Once this is done, the user of the stress indices can quickly and economically calculate the value of σ for any particular geometry within the range of parameters and loading covered. However, it should be noted that the user of the stress indices must be provided not only with values of the stress index, I, but also a quantitative definition of the nominal stress, S_n.

The definition of S_n is an arbitrary choice in the process of establishing values of I; other than it must have dimensions of stress and (for elastic analysis) must be proportional to the load. However, the definition of S_n used in establishing the values of I must be used in subsequent calculations of σ by equation (1). It might appear that this aspect is so obvious that it hardly merits pointing out. However, as indicated by the following discussion of "Code History", it appears that S_n was incorrectly defined when the stress index method was first introduced into the Code in 1963, and that, at present, the appropriate definition of S_n is still not established. Indeed, it is the question of an appropriate definition of S_n which is the main topic of this report.

Code History

Table NB-3338.2(c)-1 was introduced in the Code in the 1963 edition as Article I-6. The numerical values of the indices at that time were almost identical to those now in Table NB-3338.2(c)-1; Table 1 herein: these values are discussed later herein. The text of I-611, explaining the use of the indices, read as follows:

> "I-611 Stress Index - The term stress index, as used herein, is defined as the numerical ratio of the stress components, σ_t, σ_n and σ_r under consideration to the allowable stress value S_m for the vessel material."

In view of the definition of a stress index, it is obvious that the definition given in I-611 is meaningless. Note that S_m depends upon the material and temperature and in no way on the pressure range. Accordingly, the definition leads to the "nonsense" equation:

$$\sigma = I\ S_m$$

where σ = stress range due to an applied pressure range

I = stress index

S_m = allowable stress intensity.

It is pertinent to note that S_m can be used as a component of the nominal stress to give the equation:

$$\sigma = I\ \frac{P_o}{P_d}\ S_m \qquad (2)$$

where P_o = pressure range for cycle under consideration

P_d = design pressure

This error was continued in the 1965 edition, but, in the 1968 edition of the Code, the words "allowable stress value S_m for the vessel material" were replaced with: "computed membrane stress intensity in the unpenetrated and unreinforced vessel material". While there are some interpretation problems with the new words, they, at least, imply* that the stress range σ is proportional to the applied pressure range, P_o.

---

* The implication follows from the title of NB-3338: "Pressure Stresses in Openings for Fatigue Evaluation".

The 1971 edition of the Code moved what was I-6 into the body of the Code as NB-3338; otherwise, there was no major change from the 1968 edition. The 1971 edition included, for the first time, a section on "Piping Design" and, in particular, Table NB-3686.1-1; Table 3 herein.

The 1974 edition of the Code included for the first time, NB-3339, "Alternative Rules for Nozzle Design" and, in particular, Table NB-3339.7-1; Table 2 herein.

In the 1974 edition, NB-3338.2(a) gives a definition of the term stress index as follows:

> "(a) The term stress index, as used herein, is defined as the numerical ratio of the stress components, σ_t, σ_n, and σ_r under consideration to the computed membrane stress intensity in the unpenetrated and unreinforced vessel material. When the thickness of the vessel wall is increased over that required to the extent provided hereinafter, the values of r_1 and r_2 in Fig. NB-3338.2-2 shall be referred to the thickened section."

The word "unreinforced" could be interpreted* to mean that the minimum required shell thickness is to be used in calculating stress ranges by the stress index method. Accordingly, the following equations would apply:

$$\sigma = I \frac{P_o D_m}{2\ t_r}\ , \text{ for cylindrical shells} \qquad (3a)$$

$$\sigma = I \frac{P_o D_m}{4\ t_r}\ , \text{ for spherical shells} \qquad (3b)$$

where σ = stress range** due to pressure range

I = stress indices given in Table 1.

P_o = range of pressure for cycle under consideration

D_m = mean diameter of shell

t_r = required minimum wall thickness of cylindrical shell.

---

* The interpretation must also include the assumptions that (a) the membrane stress is produced by internal pressure, (b) for cylindrical shells, the hoop membrane stress is the maximum membrane stress, and (c) the membrane stress intensity is equal to the maximum membrane stress.

** The stress index method is intended for use in a fatigue analysis, hence it is in the <u>range</u> of stresses due to a <u>range</u> of loads that is significant.

In June of 1975, NB-3338.2(a) was revised* to read as follows:

"(a) The term stress index, as used herein, is defined as the numerical ratio of the stress components, σ_t, σ_n, and σ_r under consideration to the computed membrane stress in the unpenetrated vessel material; however, the material which increases the thickness of a vessel wall locally at the nozzle shall not be included in the calculations of these stress components. When the thickness of the vessel wall is increased over that required to the extent provided hereinafter, the values of r_1 and r_2 in Fig. NB-3338.2-2 shall be referred to the thickened section."

The deletion of the word "unreinforced" and the additional clause: "however, the material which increases......" are intended to make clear that, in calculating stress ranges by the stress index method, the actual (nominal) wall thickness (not minimum required) is to be used. Accordingly, the following equation would apply:

$$\sigma = I \frac{P_o D_m}{2\ t_a}\text{, for cylindrical shells} \quad (4a)$$

$$\sigma = I \frac{P_o D_m}{4\ t_a}\text{, for spherical shells} \quad (4b)$$

where t_a = actual shell wall thickness, other symbols are defined under equation (3).

It should be noted that equations (3) and (4) give the same answer if $t_r = t_a$. However, if all required reinforcing is obtained by making $t_a > t_r$, equation (3) will give higher values of σ than equation (4); usually, by a factor of two or more. Intermediate conditions occur in which <u>part</u> of the reinforcing is obtained by making $t_a > t_r$; here also, equation (3) will give higher stress ranges than equation (4).

NB-3339 defines the term stress index as follows:

"(a) The term stress index, as used herein, is defined as the numerical ratio of the stress components, σ_t, σ_n, and σ_r under consideration to the computed stress, S."

The value of S, using the nomenclature of this report, is defined in NB-3339.7 as $P\ D_m/2\ t_r$ or $P\ D_m/4\ t_r$ for cylindrical or spherical shells, respectively. However, P is not defined. If it were defined as P_o, the pressure range in

---

* See Winter 1975 Addenda to the Code.

the cycle under consideration, then the stress index method as defined in NB-3339 would be identical to equation (3).

NB-3686.3 states that stresses are to be obtained by multiplying the indices by $P\ D_m/2\ t_r$ or $P\ D_m/4\ t_r$ for pipe or formed heads, respectively. Accordingly, it is identical to equation (3) except that P is simply defined as internal pressure and does not explicitly introduce the concept of stress range due to load range for a fatigue analysis.

Table NB-3683.2-1 (not included in this report) also introduces and defines a stress index which is related to the indices of Tables 1, 2, and 3. The indices for pressure loading of "Branch connections per NB-3643: are $C_1 = 2.0$, $K_1 = 1.7$. The product, $C_1K_1 = 2 \times 1.7 = 3.4$ is intended to correspond to the maximum stress intensity index shown in Tables 1, 2, or 3 herein; i.e., 3.3. The stress is obtained from the stress index by multiplying the index by $P_oD_o/2\ t_n$, where P_o = range of pressure, D_o/t_n = outside diameter-to-nominal wall thickness ratio of run (shell) or branch; whichever gives the larger value of D_o/t_n. Noting that the nominal thickness is equivalent to the actual shell thickness, the stress index method of Table NB-3683.2-1 is essentially (disregarding difference between D_o and D_m) the same as equation (4a)

There are a number of limitations on the nozzles to which the stress indices are applicable. These are given in NB-3338.2(d) and NB-3339.1; included herein as Table 4. The referenced Figure NB-3338.2-2 is included herein as Figure 1. These limitations are essentially the same as shown in the 1963 edition with perhaps one significant exception; in 1963, the D/t limit for cylindrical shells was 50 and was increased to 100 in the 1968 edition. Similar limitations are given in NB-3686.1 and Table NB-3683.2-1.

In summary of the above, the Code history indicates considerable confusion concerning the stress index method and, at present, we have two parts of the Code (NB-3338 and NB-3683.2) which use the <u>actual</u> wall thickness for computing the nominal stress, S_n; two other parts (NB-3339 and NB-3686.3) which use the <u>required</u> minimum wall thickness.

TABLE 4. LIMITATIONS OF APPLICABILITY OF STRESS INDICES IN TABLES NB-3338.2(c)-1 and NB-3339.7-1

NB-3338.2

(d) The indices of Table NB-3338.2(c)-1 apply when the conditions stipulated in *(1)* through (6) exist.

(1) The opening is for a circular nozzle whose axis is normal to the vessel wall. If the axis of the nozzle makes an angle φ with the normal to the vessel wall and if $d/D \leq 0.15$, an estimate of the σ_n index on the inside may be obtained from one of the following formulas:

For hillside connections in spheres or cylinders

$$K_2 = K_1 \ (1 + 2 \sin^2 \varphi)$$

For lateral connections in cylinders

$$K_2 = K_1 \ [1 + (\tan \varphi)^{4/3}]$$

where

K_1 = the σ_n inside stress index of Table NB-3338.2(c)-1 for a radial connection

K_2 = the estimated σ_n inside stress index for the nonradial connection

(2) The arc distance measured between the centerlines of adjacent nozzles along the inside surface of the shell is not less than three times the sum of their inside radii for openings in a head or along the longitudinal axis of a shell and is not less than 2 times the sum of their radii for openings along the circumference of a cylindrical shell.

(3) The dimensional ratios are not greater than the following:

Ratio	*Cylinder*	*Sphere*
$\dfrac{\text{Inside shell diameter}}{\text{Shell thickness}} = \dfrac{D}{t}$	100	100
$\dfrac{\text{Inside nozzle diameter}}{\text{Inside shell diameter}} = \dfrac{d}{D}$	0.50	0.50

In the case of cylindrical shells, the total nozzle reinforcement area on the transverse axis of the connections including any outside of the reinforcement limits, shall not exceed 200% of that required for the longitudinal axis (compared to 50% permitted by Fig. NB-3332.2-1) unless a tapered transition section is incorporated into the reinforcement and the shell, meeting the requirements of NB-3361.

(4) The inside corner radius, r_1 (Fig. NB-3338.2-2), is between 10% and 100% of the shell thickness, t.

(5) The outer corner radius, r_2 (Fig. NB-3338.2-2), is large enough to provide a smooth transition between the nozzles and the shell. In addition, for opening diameters greater than 1½ times shell thickness in cylindrical shells and 2:1 ellipsoidal heads and greater than 3 shell thicknesses in spherical shells, the value of r_2 shall be not less than ½ the thickness of the shell or nozzle wall, whichever is greater.

(6) The radius, r_3 (Fig. NB-3338.2-2), is not less than the greater of the following:

(a) 0.002 θd_o where d_o is the outside diameter of the nozzle and is as shown in Fig. NB-3338.2-2, and the angle θ is expressed in degrees;

(b) $2(\sin \theta)^3$ times offset for the configuration shown in Figs. NB-3338.2-2(a) and (b).

NB-3339 Alternative Rules for Nozzle Design

Subject to the limitations stipulated in NB-3339.1, the requirements of this paragraph constitute an acceptable alternative to the rules of NB-3332 through NB-3336 and NB-3338.

NB-3339.1 Limitations. These alternative rules are applicable only to nozzles in vessels within the limitations stipulated in (a) through (e) below.

(a) The nozzle is circular in cross section and its axis is normal to the vessel or head surface.

(b) The nozzle and reinforcing (if required) are welded integrally into the vessel with full penetration welds. Details such as those shown in Figs. NB-4244(a)-1, NB-4244(b)-1 and NB-4244(c)-1 are acceptable. However, fillet welds shall be finished to a radius in accordance with Fig. NB-3339.1-1.

(c) The edge of the opening is at least $2.5\sqrt{RT}$ from the nearest edge of any other opening.

(d) The material used in the nozzle, reinforcing, and vessel adjacent to the nozzle shall have a ratio of UTS/YS of not less than 1.5 where

UTS = specified minimum ultimate tensile strength

YS = specified minimum yield strength

(e) The following dimensional limitations are met

Typo error correction

	Nozzles in Cylindrical Vessels	*Nozzles in Spherical Vessels or Heads*
D/T	10 to 200	10 to 250
d/D	0.33 max	0.5 max
$d/\sqrt{RT}$	1.56 max	. . .

t is presumably the nominal (actual) shell thickness.

T = required shell thickness

R = D/2.

Numerical Values of Code Indices

As remarked earlier, the numerical values of indices shown in the present (1974 edition) Table NB-3338.2(c)-1 are almost identical to those introduced in the 1963 Code. The only exception is for σ_r, inside corner. In the 1963 edition, these were correctly shown as -t/R for cylindrical shells and -2t/R for spherical shells, if t and R (not defined) were appropriately taken as the shell thickness and shell mean radius, respectively, and either equation (3) or (4) were used. This follows because the range of σ_r on the inside surface can only be $-P_o$. The value of I must be such that we get that answer from equation (3) or (4); i.e.

$$\sigma_r = -P_o = -\frac{t}{R} \times \frac{P_o \times 2 \times R}{2t} \text{, for cylindrical shells}$$

$$\sigma_r = -P_o = -\frac{2t}{R} \times \frac{P_o \times 2 \times R}{4t} \text{, for spherical shells}$$

In the above, t can be either t_a or t_r, if used consistently with the definitions of S_n. In 1971, however, the index for σ_r, inside corner, was changed to $-t_n/R$ and $-2\ t_n/R$, where t_n would presumably be interpreted by looking at Figure NB-3338.2-2 (Figure 1 herein) as the <u>nozzle</u> thickness. This is obviously incorrect and seems to be further evidence of confusion as to just what the value of S_n in the stress index method should be. It may be noted that Tables NB-3339.7-1 and NB-3686.1-1 (Tables 2 and 3 herein) show a correct value of σ_r, inside corner, which is consistent with the use of required wall thickness in calculating S_n.

The basis for the original selection of the values of stress indices is not known to the author. At the time (~1961/1962), at least preliminary results from tests reported in References (1), (2), (3), and (4) were available.

The introduction of a single set of stress indices to cover the entire parameter range of $D/t_a < 100$ and $d/D < 0.5$, and for all of the infinite variety of detailed shapes of reinforcement permitted by the Code rules, is obviously a gross simplification of the actual stresses that can exist.

The tables of stress indices indicate maximum stress locations and directions, and relative magnitudes of stresses at other locations/directions. For example, the indices of Table 1 indicate that for nozzles in cylindrical shells, the maximum stress occurs in the longitudinal plane, inside surface, normal direction and that the stress in the transverse plane, inside surface, tangential direction is -0.2/3.1 times the maximum stress. Test data shows later herein indicate that this relationship is not always valid and can be significantly inaccurate.

In this report, we do not further address the question of the relative magnitude of the indices. Rather, we address the question of whether the indices, and the definition of the nominal stress, give a conservative evaluation of the maximum stress range due to a given pressure range. Specifically, we will be examining the equations:

$$\sigma = 3.1 \frac{P_o D_m}{2t_r} \text{, for cylindrical shells} \tag{5a}$$

$$\sigma = 2.0 \frac{P_o D_m}{4t_r} \text{, for spherical shells} \tag{5b}$$

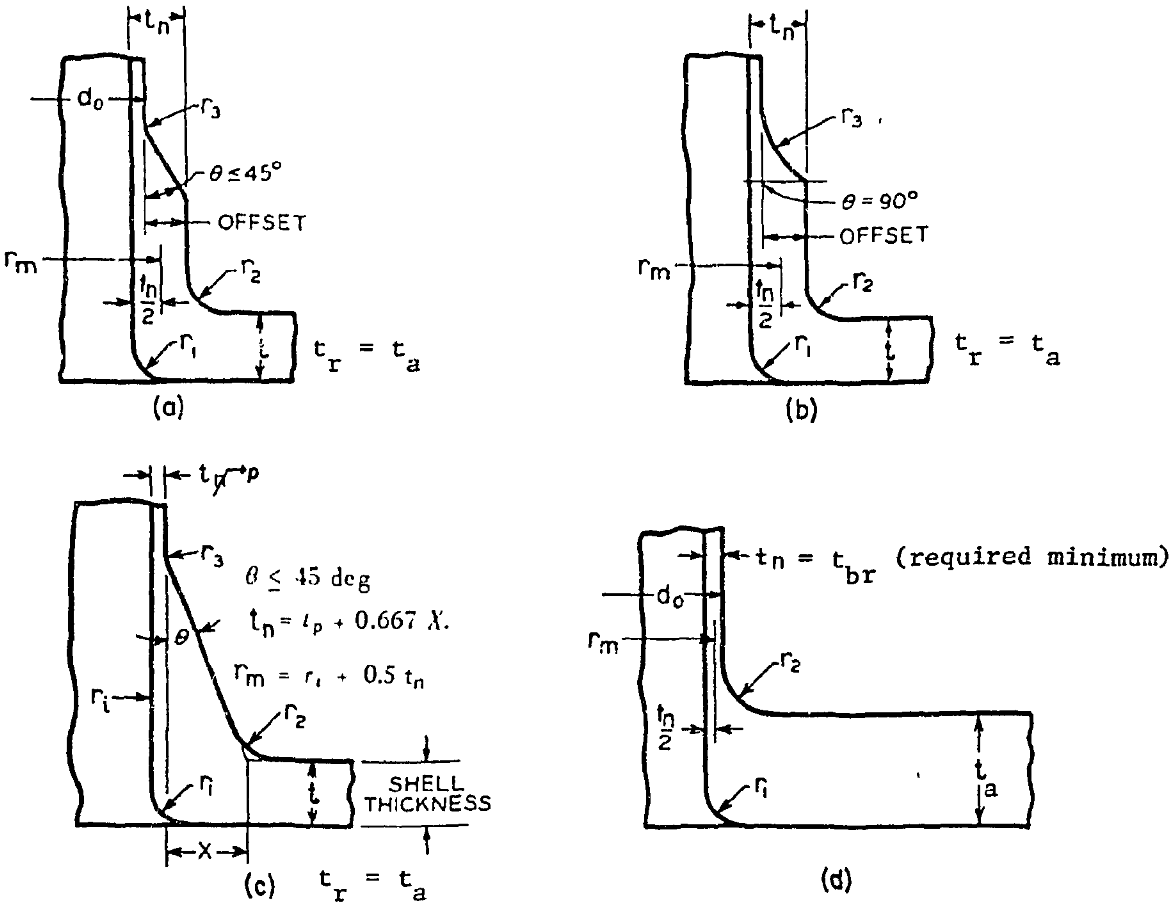
$$\sigma = 3.1 \frac{P_o D_m}{2t_a} \text{, for cylindrical shells} \tag{6a}$$

$$\sigma = 2.0 \frac{P_o D_m}{4\ t_a} \text{, for spherical shells} \tag{6b}$$

REINFORCEMENT REQUIREMENTS

The stress index method is applicable only to nozzles reinforced in accordance with Code rules. Figure 2 is a reproduction of part of Figure 1 with some changes to illustrate the following discussion. Figures 2(a), (b), and (c), show nozzles which are "locally" reinforced, with $t_r = t_a$. Figure 2(d) shows a nozzle which is reinforced to meet Code rules by increasing the shell thickness $t_r < t_a$. The nozzle thickness is assumed to be only the required minimum thickness; i.e., $t_n = t_{br}$. There are, of course, an infinite variety of reinforcement shapes and combinations of local reinforcement combined with shell thickening ($t_r < t_a$); all of which may be used to meet the Code reinforcement requirements. However, for the present discussion, only the bounding case where the reinforcement is obtained entirely from shell thickening [Figure 2(d)] will be considered.

In the following, test-derived stresses and theoretically-derived stresses will be cited which show that, for nozzles in a cylindrical shell, like Figure 2(d), if the parameter $\rho = (d/D)\sqrt{D/t_a}$ is less than about 0.6, equation (6a) using t_a is conservative and equation (5a) using t_r is excessively conservative. Above this parameter limit, equation (5a) remains conservative but equation (6a) may be unconservative; by factors of 3.0 or more. For nozzles in spherical shells, equation (5b) using t_r is essentially always conservative but equation (6b) using t_a is conservative only for values of ρ of about 0.05 or less. At the upper limit of applicability of Table NB-3338.2(c)-1 ($\rho = 5.0$), equation (6b) can be unconservative by factors of up to 5.5.



$t_a > t_r$ such that all required reinforcement is obtained by excess shell thickness.

FIGURE 2. ILLUSTRATION OF CONCEPTUAL DIFFERENCE BETWEEN NOZZLES WITH "LOCAL" REINFORCING, (a), (b), AND (c) AND NOZZLES REINFORCED BY EXCESS SHELL THICKNESS, (d)

TEST DATA

Numerous model tests have been conducted under PVRC auspices in which stresses at nozzles in pressure vessels have been measured. Much of the data is for models with some amount of local reinforcing; these results are not directly relevant to the present report which is concerned with nozzles that are reinforced by shell thickening; $t_a > t_r$. Tables 5 and 6 summarize data for models with relatively little local reinforcing, and:

(a) D/t_a and d/D within the range of applicability of the indices,

(b) r_1/t_a and r_2/t_a within the restrictions of applicability of the indices (with minor deviations of r_2/t_a for three models)

An exception to (b) occurs for the ORNL-1 model. This model was constructed with "sharp corners"; i.e., with radii r_1 and r_2 made as small as feasible. The indices cited in Table 5 for ORNL-1 are for the highest measured stresses at the cited locations and directions. The center of the strain gage closest to the inside corner was 0.165" from the corner; the center of the strain gage closest to the outside corner was 0.075" from the outside corner. These distances, while small, are significant fractions of the shell wall thickness of 0.10". Rows of gages were placed at each location so that it is possible to extrapolate to the corners. In Reference (6), such an extrapolation lead to an estimated maximum actual stress index of 13.3. This stress occurred on the outside surface of the nozzle in the longitudinal plane, normal direction. This is also the location, surface, and direction of the maximum measured stress index of 7.85 shown in Table 5. However, this model did not have a fillet radius. If a fillet radius of r_2 = 0.5 t_a, had been used, it is estimated that the maximum actual stress would be representable by an index of about 8 to 10.

Table 5, for nozzles in cylindrical shells, indicates that models with $\rho = (d/D)\sqrt{D/t_a}$ of 0.69 and less had maximum stress indices less than 3.1. This indicates that equation (6a) (which uses t_a to calculate S_n) is conservative within this parameter range. For larger values of ρ, equation (6a) is unconservative; for ORNL-1 by a factor of at least 7.85/3.1 = 2.5.

TABLE 5. TEST DATA[a], NOZZLES IN CYLINDRICAL SHELLS

Model Number	$\rho = \frac{d}{D}\sqrt{\frac{D}{t_a}}$	$\frac{D}{t_a}$	$\frac{d}{D}$	$\frac{s}{S}$[c]	$\frac{r_1}{t_a}$	$\frac{r_2}{t_a}$	Stress Index[b], I							
							Longitudinal Plane				Transverse Plane			
							Inside		Outside		Inside		Outside	
							σ_n	σ_t	σ_n	σ_t	σ_n	σ_t	σ_n	σ_t
(d)	< 5.00	<100	<0.5	--	0.1/1.0	>0.5	3.1	-0.2	1.2	1.0	1.0	-0.2	2.1	2.6
C-1A	0.17	11.9	0.05	1.04	0.55	0.55	2.45	0.05	1.09	0.65	1.02	-0.28	1.07	0.94
C-2A	0.45	12.1	0.13	0.97	0.56	0.56	2.80	0.03	0.99	0.65	1.00	-0.32	1.37	1.01
C-3C	0.47	5.5	0.20	2.02	0.56	0.56	3.09	-0.15	2.06	1.10	1.97	-0.47	2.52	1.16
C-3A	0.69	11.9	0.20	1.01	0.55	0.55	3.0	0.06	1.22	0.63	1.02	-0.58	1.95	1.30
C-5C	1.18	5.6	0.50	1.97	0.56	0.56	4.1	-0.21	2.11	1.05	1.94	-1.01	3.72	2.30
WC-2AQ	1.29	99.1	0.13	0.93	0.19	0.50	3.61	-1.78	4.75	3.67	1.20	-1.77	2.15	1.60
E-4B	1.75	12.2	0.50	0.99	0.56	0.68	3.50	-0.04	1.90	1.25	1.16	-1.57	2.91	2.59
E-4E	1.75	12.3	0.50	0.98	0.34	0.69	3.80	-0.07	1.89	1.13	--	-1.63	2.66	2.51
E-4	1.77	12.5	0.50	0.98	0.58	0.46	3.50	-0.09	1.98	1.44	--	-1.77	2.91	2.75
ORNL-1 (e)	4.95	98.0	0.50	1.00	(e)	(e)	5.00	-2.88	7.85	4.28	1.20	-2.02	2.74	2.36

(a) Model WC-2AQ is from Reference (5), ORNL-1 from Reference (6); all other models from Reference (3).

(b) Except for the first row, these are stress indices from test data where σ_n or σ_t = I (P_o D_m/2 t_a).

(c) $s/S = (d/D)(t_a/t_n)$.

(d) This row gives dimensional limits of applicability of stress indices [NB-3338.2(d)] and stress indices from Table NB-3338.2(c)-1.

(e) See text.

TABLE 6. TEST DATA[a], NOZZLES IN SPHERICAL SHELLS

Model Number	$\frac{d}{D}\sqrt{\frac{D}{t_a}}$	$\frac{D}{t_a}$	$\frac{d}{D}$	$\frac{s}{S}$[c]	$\frac{r_1}{t_a}$	$\frac{r_2}{t_a}$	Stress Index[b], I — Inside σ_n	Inside σ_t	Outside σ_n	Outside σ_t
(d)	< 5.00	<100	<0.5	--	0.1/1.0	>0.5	2.0	-0.2	2.0	2.0
S-1AB	0.25	24.0	0.05	1.00	0.25	0.56	2.17	-0.02	1.51	1.07
N-8G	0.62	24.5	0.13	1.00	0.68	0.68	2.04	-0.38	1.96	1.29
S-3CB	0.68	11.6	0.20	1.98	0.25	0.48	2.62	-0.25	2.90	1.70
N-5B	0.82	16.8	0.20	1.38	0.40	0.72	2.23	-0.35	2.70	1.55
N-1A	1.00	25.0	0.20	1.13	0.58	0.58	2.01	-0.78	2.72	1.84
N-1AA	1.00	25.0	0.20	0.98	0.57	0.57	2.04	-0.77	2.55	1.74
S-5AW	1.50	9.0	0.50	1.10	0.55	0.55	1.68	-0.41	2.29	1.93
S-5C	1.70	11.5	0.50	2.00	0.25	0.55	2.45	-0.66	3.62	2.60
N-9B	1.86	24.0	0.38	0.99	0.49	0.49	1.81	-0.96	3.00	2.74
N-9C	1.86	24.0	0.38	1.02	0.66	0.66	1.85	-0.83	2.75	2.40
WN-50B	1.91	50.2	0.27	1.10	0.25	0.80	2.83	-1.62	3.92	3.06
S-5A	2.45	24.0	0.50	1.00	0.25	0.81	2.00	-0.62	2.87	2.50
S-5AZ	4.18	70.0	0.50	1.02	0.25	0.78	2.34	-1.78	4.90	4.20

(a) Model WN-50B is from Reference (4); all other models are from Reference (3).
(b) Except for the first row, these are indices from test data where σ_n or σ_t = I $(P_o D_m/4\ t_a)$.
(c) $s/S = 2(d/D)(t_a/t_n)$.
(d) This row gives dimensional limits of applicability of stress indices [NB-3338.2(d)] and stress indices from Table NB-3338.2(c)-1.

Table 6, for nozzles in spherical shells, indicates that all models had maximum stress indices greater than 2.0. This indicates that equation (6b) is never conservative. For model S-5AZ, equation (6b) is unconservative by a factor of 4.9/2.0 = 2.45.

It is significant to note that all test models listed in Tables 5 and 6 are fully reinforced in accordance with Code requirements for some value of t_r. For models with ρ less than about 1.0, the ratio t_r/t_a is about one-half.* For models with ρ greater than about 1.0, the 2/3 area within $(r + 0.5\sqrt{R_m t_a})$ controls, and the value of t_r/t_a is approximately given by:

$$\frac{t_r}{t_a} = \frac{1}{1 + \rho} \tag{7}$$

It follows from the t_r/t_a ratios that equation (5) which uses t_r to calculate S_n, is always conservative. By using equation (5), in effect, the test-derived stress indices are multiplied by t_r/t_a for comparison with Code index values. Since t_r/t_a cannot be greater* than 0.5, all indices in Table 5 are less than 3.1 and all test indices in Table 6 are less than 2.0 with two exceptions: ORNL-1 in Table 5 and S-5AZ in Table 6. However, in both of these exceptions, equation (7) applies and t_r/t_a is equal to 0.17 for ORNL-1 and to 0.19 for S-5AZ. The test indices, with these t_r/t_a ratios, are then conservative using equation (5); i.e., 0.17 x 7.85 = 1.33 which is well below 3.1 and 0.19 x 4.90 = 0.93 which is well below 2.0. In general, equation (5) tends to be excessively conservative when applied to nozzles reinforced entirely by an increase in vessel wall thickness; $t_a > 2\ t_r$.

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* Ignoring the area of the fillet radius.

CALCULATED STRESSES

We need to look at calculated stress data because the test data give only a few isolated points where equation (6) is or is not conservative. Further, the test models do not include models which completely bound the possibilities included in the use of equation (6). Such a bound consists of an opening in a cylindrical or spherical shell. Note that an opening in a shell can be fully reinforced in accordance with Code rules by making t_a appropriately greater than t_r. Because equation (6) uses t_a in calculating S_n, equation (6) is applicable to any opening in a shell up to the limit $d/D = 0.5$. Accordingly, the calculated stresses at openings in cylindrical shells or spherical shells provides a significant bound on the applicability of equation (6). Calculations for nozzles are also included herein; the models are such that the nozzle wall gives relatively little local reinforcement.

Openings or Nozzles in Cylindrical Shells

Figure 3 shows calculated maximum membrane and membrane plus bending stresses for a hole in a cylindrical shell. The theory is that of Van Dyke[7]. Reference (8) gives about the same results. the theory indicates that stresses are a function of the single parameter: ρ. The stress corresponds to σ_n on the inside surface, longitudinal plane. Figure 3 indicates that equation (6a) is conservative for any value of ρ up to about 0.6, but is unconservative for values of ρ greater than about 0.6.

Calculated stresses for nozzles in cylindrical shells are also shown in Figure 3. These stresses were taken from Reference (9) and were calculated using Eringen's analysis. The plotted stress is specifically σ_n, inside corner, longitudinal plane. Eringen's analysis is a "two-piece" shell theory and cannot be used to evaluate the effect of the fillet radius required by the Code; i.e., $r_2 \geq 0.5\ t_a$. Inclusion of this fillet radius might reduce the maximum calculated stresses but probably only to a minor extent.

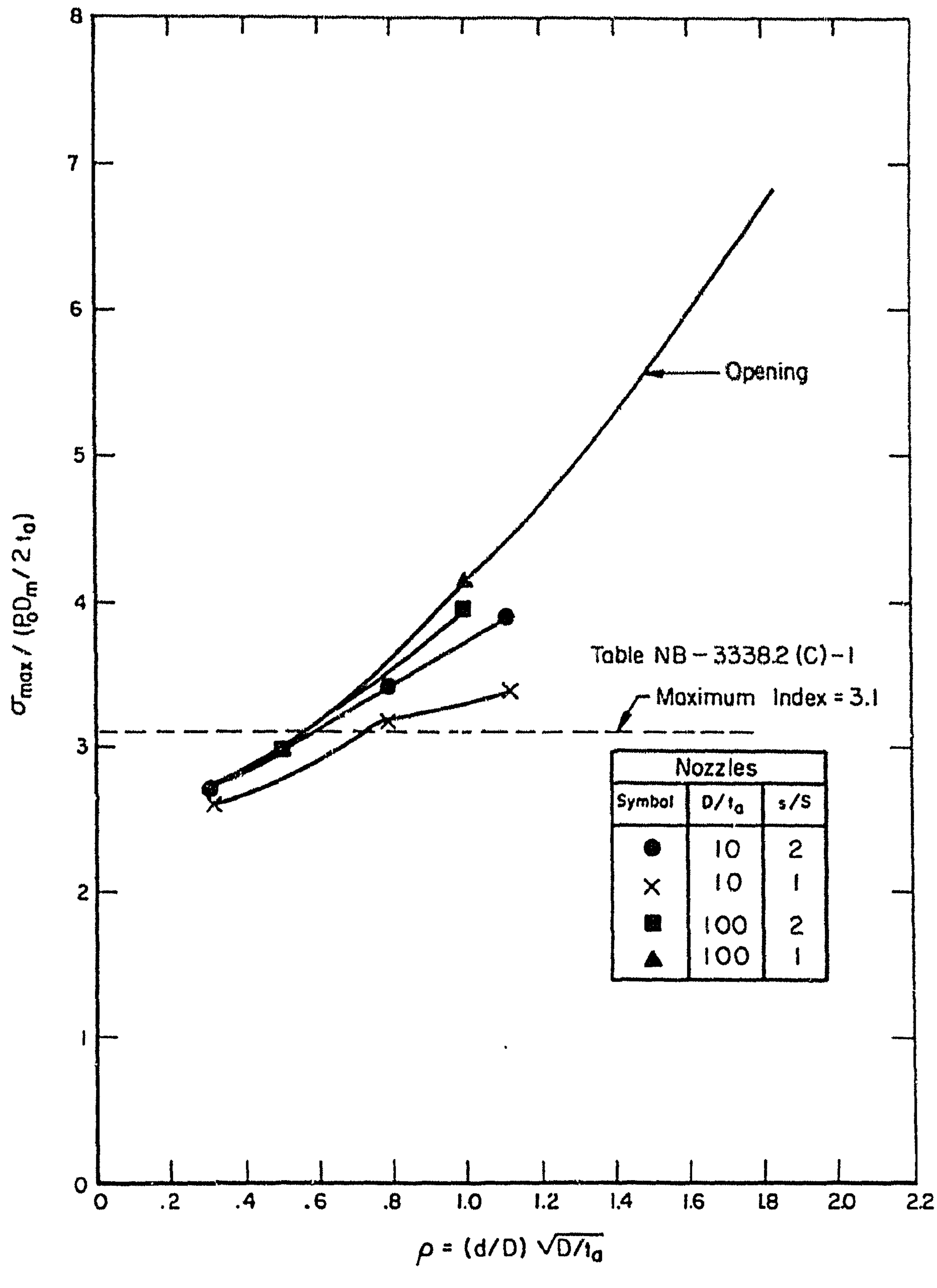

FIGURE 3. CALCULATED MAXIMUM STRESS AT OPENINGS OR NOZZLES IN CYLINDRICAL SHELLS

Openings or Nozzles in Spherical Shells

Figure 4 shows calculated maximum stresses in the spherical shell at an opening in the spherical shell and for nozzles in spherical shells. The results were obtained* from Figure 2 of Reference (10). Figure 4 uses the same scales as Figure 3 so that a visual comparision of the stresses at openings/ nozzles in cylindrical shells can be made with the stresses at openings/nozzles in spherical shells.

Figure 5 shows the calculated maximum stresses out to $\rho = 5.0$; the limit of coverage of NB-3338.

The calculated stresses shown in Figures 4 and 5 for nozzles are based on a discontinuity-type analysis of an intersection of a cylindrical shell with a spherical shell. The midsurfaces of the two parts are assumed to be rigidly joined. The calculated stresses shown are maximum stresses in the spherical shell, <u>not</u> in the cylindrical shell nozzle. The calculated stresses in the nozzles are usually higher than those in the spherical shell; for some parameters, several times as high.

Figure 6 shows maximum calculated stresses for nozzles in spherical shells with a fillet radius as required by NB-3338; i.e., $r_2 = 0.5\ t_a$. Table 7 gives maximum stresses for both inside and outside surfaces and for the normal and tangential directions; the highest of these for each model is plotted in Figure 6. These stresses were calculated with the computer program MOLSA[13]. The modeling details are described in Reference (12). Checks of calculated stresses using this computer program and modeling technique have generally indicated resonable agreement with test data.

Figure 6 also shows the calculated maximum stresses from Figure 2 of Reference (10). Except for small values of ρ, the analysis of Reference (10) is not unduly conservative as judged by the more exact MOLSA analysis, including $r_2 = 0.5\ t_a$. Indeed, for large values of ρ and large D/t_a, the MOLSA analysis gives higher stresses than obtained from Reference (10). This

* This graph is also published in Reference (11). Reference (9) gives comparisons of Reference (10) with other computer program results (using the same boundary conditions) and found only minor differences.

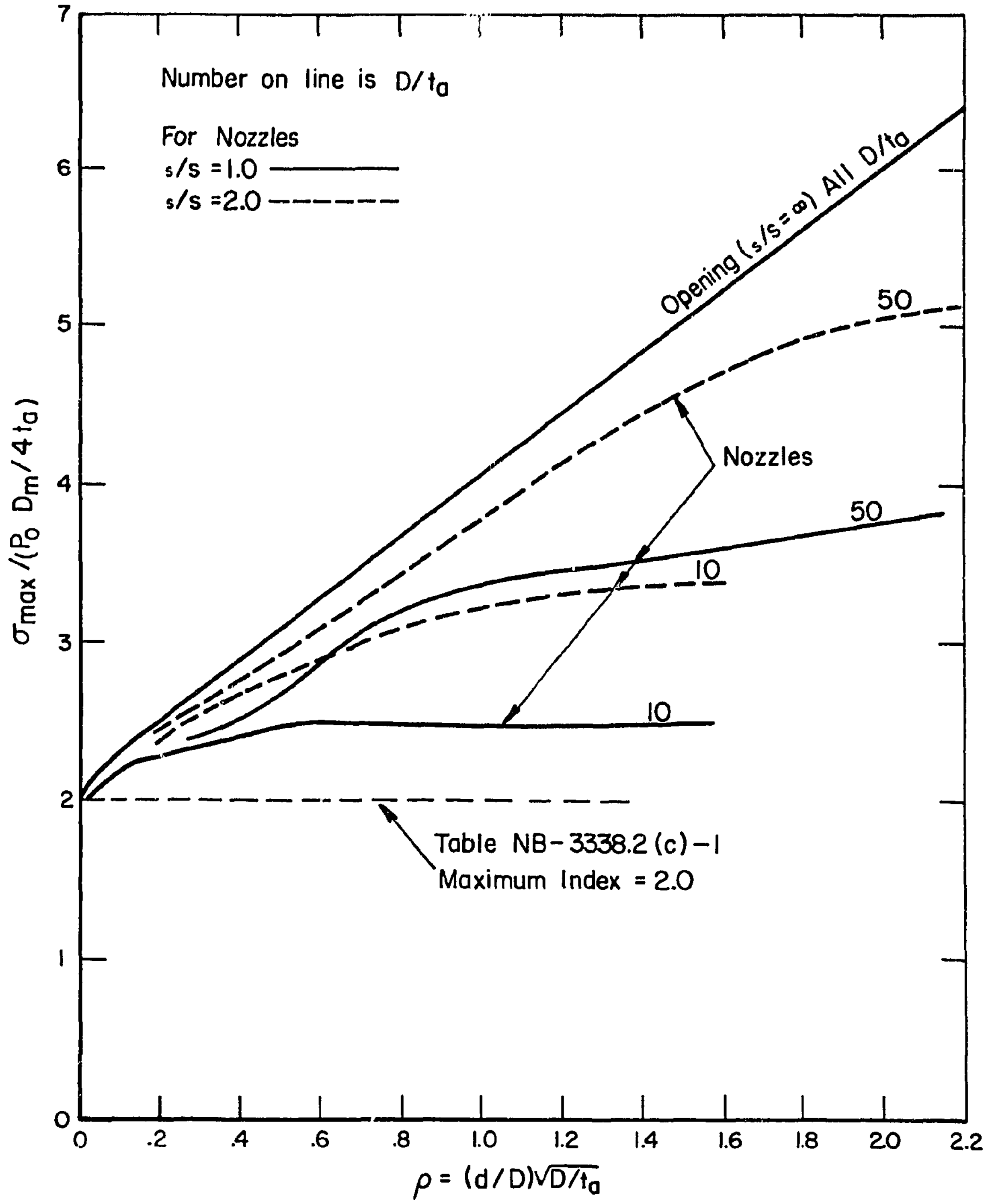

FIGURE 4. CALCULATED MAXIMUM STRESS IN SPHERICAL SHELL AT OPENING OR NOZZLE, DATA FROM REFERENCE (10), TO ρ = 2.2

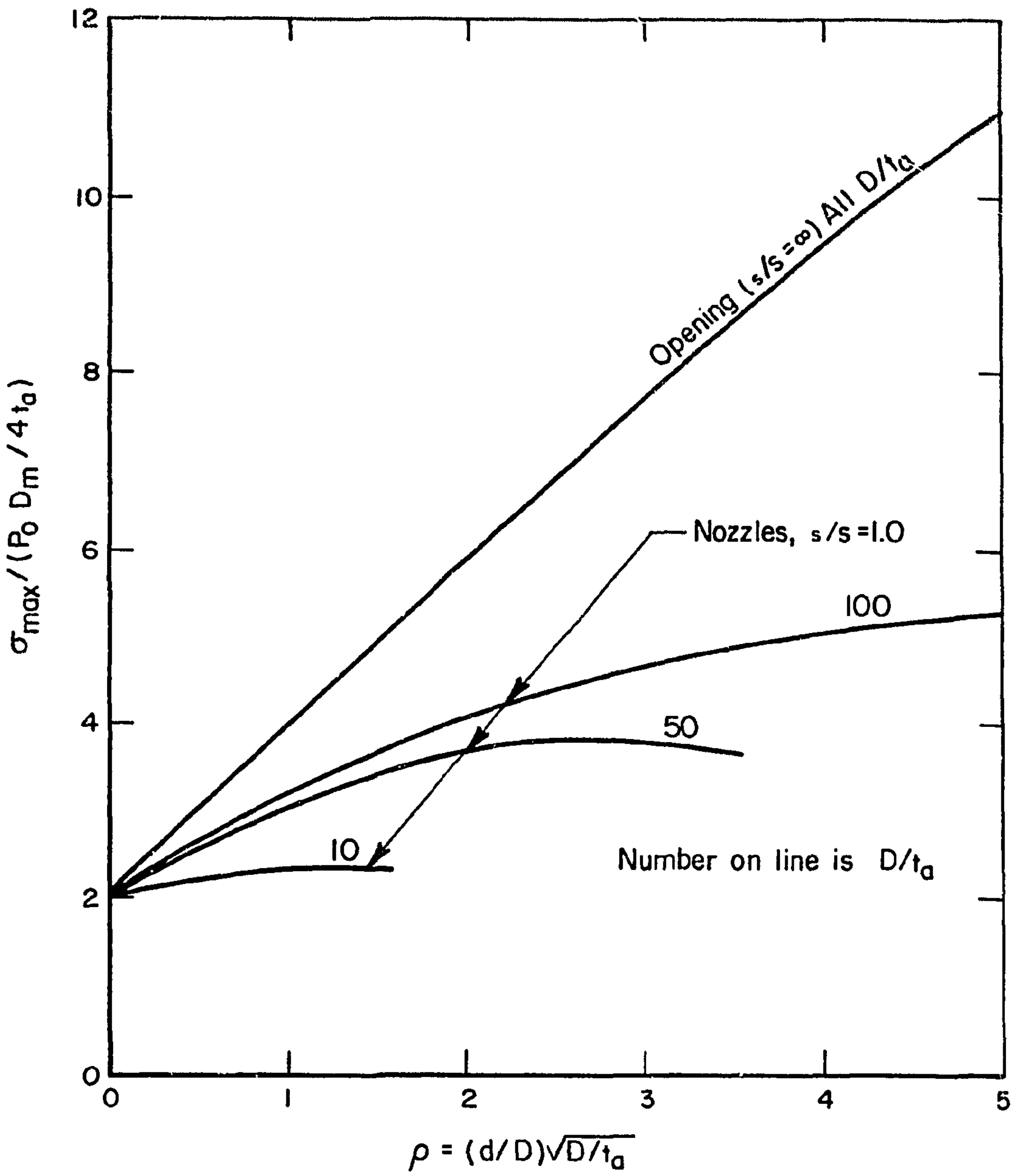

FIGURE 5. CALCULATED MAXIMUM STRESS IN SPHERICAL SHELL AT OPENING OR NOZZLE, DATA FROM REFERENCE (10), to ρ = 5.0

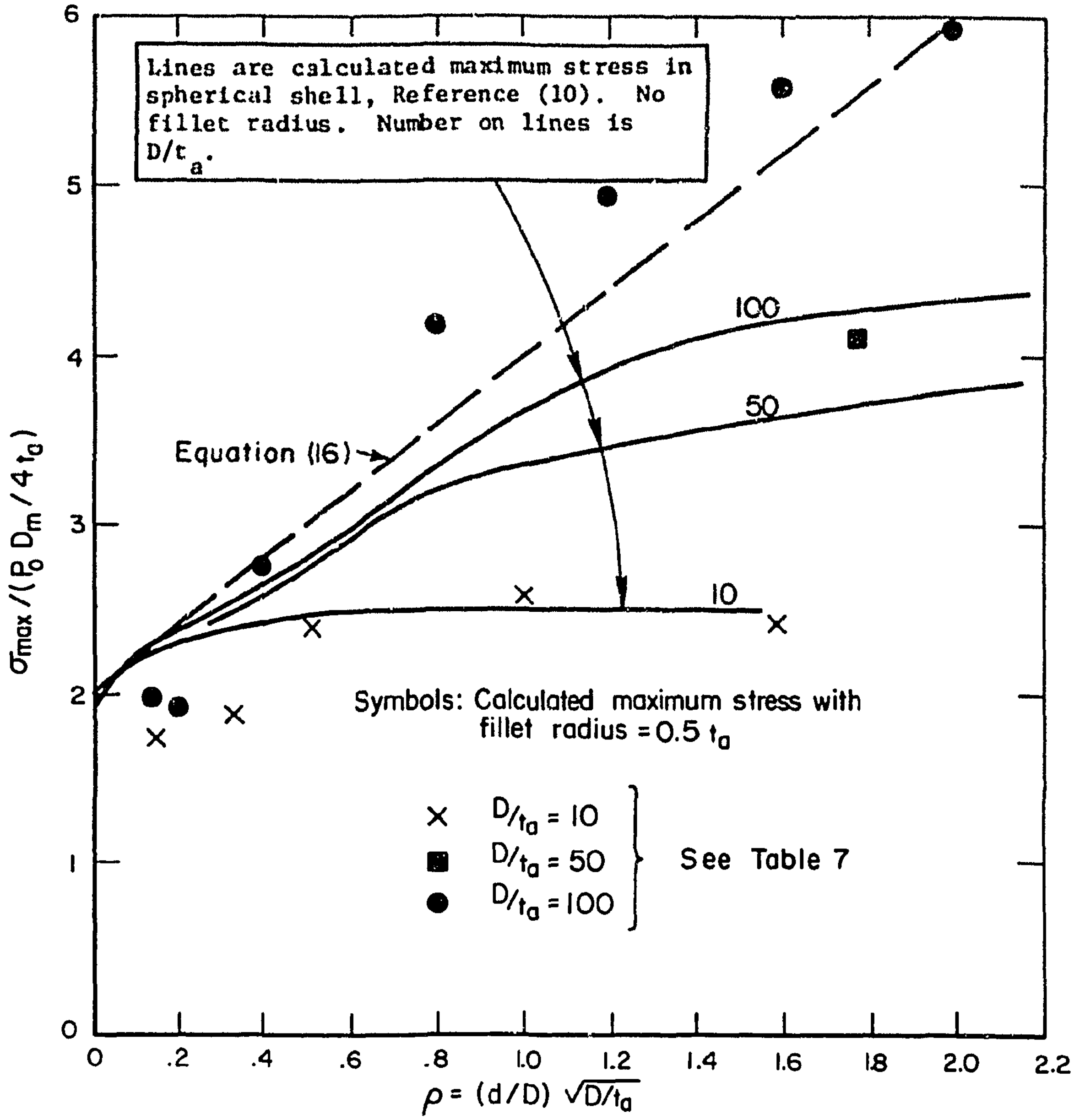

FIGURE 6. COMPARISON OF CALCULATED MAXIMUM STRESS AT NOZZLE IN A SPHERICAL SHELL, s/S = 1.0, FOR NOZZLES WITH AND WITHOUT FILLET RADII.

TABLE 7. CALCULATED STRESSES USING COMPUTER PROGRAM MOLSA[13] NOZZLES IN SPHERICAL SHELLS WITH $r_2 = 0.5\ t_a$

Model[c] Number	$\frac{D}{t_a}$	$\frac{d}{D}$	$\frac{d}{D}\sqrt{\frac{D}{t_a}}$	Maximum Stress Index: Inside σ_n	Inside σ_t[a]	Outside σ_n	Outside σ_t
(b)	10	0.0447	0.1414	1.76	-0.13	1.51	1.00
2	10	0.08	0.253	1.81	-0.20	1.86	1.00
3	10	0.16	0.506	1.91	-0.44	2.38	1.53
4	10	0.32	1.012	1.81	-0.83	2.57	2.02
5	10	0.50	1.581	1.67	-0.74	2.40	2.04
12	50	0.25	1.768	2.16	-2.89	4.09	4.01
1	100	0.01414	0.1414	1.94	-0.26	1.54	1.00
6	100	0.02	0.2	1.90	-0.27	1.82	1.00
7	100	0.04	0.4	2.14	-0.80	2.76	1.60
8	100	0.08	0.8	2.46	-2.58	4.18	3.22
9	100	0.12	1.2	2.54	-3.83	4.94	4.89
10	100	0.16	1.6	2.52	-4.53	5.29	5.61
11	100	0.20	2.0	2.55	-4.82	5.41	5.93
13	100	0.50	5.0	2.52	-3.23	4.67	4.31

(a) This column shows the highest negative stress. It is not the highest σ_t-stress on the inside surface. A tensile σ_t-stress of 1.00 exists in the sphere remote from the nozzle.

(b) These values are taken from Reference (12).

(c) s/S = 1.00 for all models.

seemingly anomaly (the fillet radius <u>increases</u> the maximum stress) is readily explained by looking at the detailed stresses in, for example, the model in Table 7 with $D/t_a = 100$, $d/D = 0.2$ where those detailed stresses are calculated with MOLSA, but using the same model as Reference (10). The stresses at the nozzle-to-sphere juncture are

	Stress Index			
	Inside		Outside	
	σ_n	σ_t	σ_n	σ_t
Sphere	2.44	-1.39	(4.17)	1.87
Nozzle	1.60	-6.67	5.91	7.69

The value given in Reference (10) is the maximum stress in the sphere and corresponds, within the accuracy of reading the graph, to the 4.17 index shown in parenthesis. But the <u>maximum</u> stress index is 7.69 for σ_t on the outside surface of the nozzle. As shown in Table 7, model No. 11, the fillet <u>reduces</u> this stress index from 7.69 for the crude "two-piece" model to 5.93 for the more refined model which includes a fillet radius of $r_2 = 0.5\ t_a$. The detailed results from MOLSA show that the maximum stress occurs in the fillet radius close to where it joins the nozzle.

Model 13 of Table 7 is not included in Figure 6 because it is way off scale ($\rho = 5.0$). However, it is an important model in that it shows that the maximum stress, in nozzles with $s/S = 1.0$ and $r_2 = 0.5\ t_a$, can <u>decrease</u> with increasing ρ.

DISCUSSION

Test and Theory Correlations

The calculated stresses shown in Figure 3 indicate that Equation (6a) should be conservative for any nozzle in a cylindrical shell with $\rho \leq 0.6$. In Table 5, which gives test data for nozzles in cylindrical shells, we note that the limitation of $\rho \leq 0.6$ does indeed eliminate these models with a stress index greater than 3.1. (It also eliminates the border line case, Model C-3A, with index of 3.0).

The calculated stresses shown in Figures 4, 5, 6, and Table 7 indicate that Equation (6b) is not conservative for any nozzle in a spherical shell, except nozzles with s/S = 1.0 (or smaller) and for ρ of about 0.2 to 0.4 (depending upon D/t_a). In Table 6, which gives test data for nozzles in spherical shells, we note that none of the models have a maximum stress index equal to or less than 2.00.

Significance of Calculated Stress Ranges

Equation (7) shows that, for large values of ρ, the ratio of t_r/t_a becomes small; e.g., for d/D = 0.5, D/t_a = 100, the value of t_r/t_a is 1/6. The question arises: Even though the stress given by Equation (6a) may be unconservative by a factor of perhaps 3, does it make any significant difference in the fatigue analysis? That is, will the pressure range be so low that stress range due to cycles of pressure are insignificant.

To examine the question, we assume that S_m = 30,000 psi. This is not the highest value of S_m given in the Code, but is on the high side and will serve to illustrate the point of this discussion. Let us further assume that the maximum operating pressure is 0.9 times the design pressure and we wish to evaluate the fatigue life for cycles of pressure from zero to 0.9 times the design pressure and back to zero. The pressure range is then

$$P_o = \frac{2S_m t_r}{D_m} \times 0.9 \qquad (8)$$

The maximum stress range by Equation (6a) is

$$\sigma_{max} = 3.1 \times \frac{P_o D_m}{2t_a} \quad . \tag{9}$$

Substituting P_o from Equation (8) into Equation (9), and with S_m = 30,000 psi, we obtain

$$\sigma_{max} = 3.1 \times 0.9 \times 30{,}000 \times t_r/t_a = 83700\ t_r/t_a \quad . \tag{10}$$

The value of t_r/t_a is approximately 0.5 for $\rho < 1.00$ and is given by Equation (7) for larger values of ρ. Now, the test data and theory presented herein suggest that Equation (9) is unconservative for values of $\rho > 0.6$ and that, instead of using the index of 3.1, we should use values ranging from 3.1 for $\rho = 0.6$ up to somewhere between 8 and 12 for $\rho = 5.0$ and for nozzles with $s/S \leq 1.0$. For the present discussion, we will assume that at $\rho = 5.0$, the "correct" maximum stress index is 3 x 3.1 = 9.3 and that the "correct" index increases linearly with ρ. With these assumptions and estimates, the "correct" stress range is given by

$$\sigma_{max} = I_e \times 0.9 \times 30{,}000 \times t_r/t_a \quad , \tag{11}$$

where

$$I_e = 3.1 \text{ for } \rho < 0.6$$

$$I_e = 3.1 + \frac{6.2}{4.4}(\rho - 0.6) \text{ for } \rho > 0.6 \quad .$$

Table 8 shows the stress ranges obtained from Equation (10) and (11). We note that, for $\rho = 5.0$, Equation (10) grossly underpredicts the stress range, (i.e.; 13950 psi vs 41850 psi), but the question is whether the stress range of 41,850 psi is significant in a fatigue analysis. If pressure

TABLE 8. MAGNITUDE OF CALCULATED STRESS RANGES FOR S_m = 30,000 PSI, PRESSURE = 0.9 OF DESIGN PRESSURE

$\rho =$ $(d/D)\sqrt{D/t_a}$	I_e (a)	t_r/t_a (b)	σ_{max}, psi(c)	
			$I = 3.1$	$I = I_e$
0.6	3.10	0.5	41,850	41,850
1.0	3.66	0.5	41,850	49,460
1.5	4.37	0.4	33,480	47,180
2.0	5.07	0.333	27,900	45,650
2.5	5.78	0.286	23,910	44,570
3.0	6.48	0.250	20,920	43,750
3.5	7.19	0.222	18,600	43,120
4.0	7.89	0.200	16,740	42,610
4.5	8.60	0.182	15,220	42,200
5.0	9.30	0.167	13,950	41,850

(a) $I_e = 3.1 + \frac{6.2}{4.4}(\rho - 0.6)$

(b) t_r/t_a 0.5 for $\rho < 1.0$

$t_r/t_a = 1/(1 + \rho)$ for $\rho > 1.0$

(c) $\sigma_{max} = I \times \frac{P_o D_m}{2t_a}$

$P_o = (2S_m t_r/D_m) \times 0.9$

S_m = 30,000 psi

is the <u>only</u> significant source of cyclic stresses, the answer is: No. This answer is apparent when one enters the Code fatigue design curves (Figure I-9.1 and I-9.2) with a stress amplitude of $S_A = 41{,}800/2 = 20{,}900$ psi. We see that we have permissible design cycles (0 to 0.9 P_d to 0) of 80,000 and up. Further, for any reasonable operating pressure cycle (e.g., $P_o < 0.2\ P_d$), we would have permissible design cycles $>10^6$ (which, the Code says, can be considered as giving zero usage fraction). Accordingly, it is apparent that a stress range of 41,800 psi, due to a pressure cycle from 0 to 0.9 P_D to 0, is of no practical significance (i.e., gives a very small fatigue usage fraction).

It is not apparent to the author that we can dismiss large inaccuracies in the stress-index method by the preceding argument. First, we note that the preceding argument can be applied to the entire gamut of stress indices shown in Tables 1, 2, and 3. If the argument is valid, the logical step would be to remove the stress-index tables from the Code and simply say in the Code that stress ranges due to pressure variations can be ignored.

However, the Code requires that stress ranges due to other loadings must be combined with that due to pressure. This is specifically pointed out in NB-3338.2(c) by the sentence: "In the evaluation of stresses in or adjacent to vessel openings and connections, it is often necessary to consider the effect of stresses due to external loadings or thermal stresses." Presumably, the stress indices are being used for this purpose. If so, it is not at all apparent that an error in the calculated stress range due to pressure of 13,950 psi versus 41,850 psi may not be significant.

Possible Code Revisions

Nozzles or Openings in Cylindrical Shells

The data presented herein indicate that the revision to NB-3338.2 (Sectional Committee III meeting, 6/19/75) and the resulting Equation (6a) herein is conservative for $\rho \leq 0.6$. The revision serves to reduce the conservatism which results from using Equation (5a). However, the data presented herein also indicate that the revision and resulting Equation (6a) herein can be unconservative for $\rho > 0.6$.

One possible revision would consist of adding to NB-3338.2(d)(3),* the limitation

$$(d/D)\sqrt{D/t_a} \leq 0.6 \quad .$$

This could, of course, be stretched up to 0.7 or 0.8 with no great amount of unconservatism. However, for $\rho = 0.6$, the maximum values of d/D, for various values of D/t_a, are

D/t_a	5	10	20	30	40	60	80	100
d/D	0.268	0.190	.134	0.110	0.095	0.077	0.067	0.060 .

If these d/D ratios are sufficient to cover almost all nozles in vessels under NB-3300, the simple addition of the limitation $\rho \geq 0.6$ would be a viable solution to the problem for cylindrical shells.

It should be noted that the Code user always has the option of making his own stress analysis and, if the stress indices were limited in application to $\rho \leq 0.6$, he would have to do so for $\rho > 0.6$.

If it is deemed appropriate to provide stress indices for $\rho > 0.6$, a possible revision might consist of the following:

(a) For: $0.6 < \rho < 5.0$, nozzles with $t_n/t_a \geq d/D$

$$\sigma_{max} = [3.1 + \frac{6.4}{4.4}(\rho - 0.6)]\frac{P_o D_m}{2t_a} \tag{12}$$

or

$$\sigma_{max} = 3.1\frac{P_o D_m}{2t_r} \tag{13}$$

(b) For: $0.6 < \rho < 2.0$, openings and nozzles with $t_n/t_a < d/D$

$$\sigma_{max} = [3.1 + \frac{4.0}{1.4}(\rho - 0.6)]\frac{P_o D_m}{2t_a} \tag{14}$$

or

$$\sigma_{max} = 3.1\frac{P_o D_m}{2t_r} \quad . \tag{15}$$

* A similar revision would be needed in NB-3683.

Equation (12) is based on data presented herein (particularly, ORNL Model No. 1) which indicate that the maximum stress index (as a factor times $P_oD_m/2t_a$) will be about 3 x 3.1 for ρ = 5.0. A linear interpolation is used between I = 3.1 at ρ = 0.6 and I = 9.3 at ρ = 5.0. Equation (14) is based on Figure 3 and a straight line between I = 3.1 at ρ = 0.6 and I = 7.1 at ρ = 2.0.

Equations (12) and (14) would presumably be used for nozzles reinforced by excess shell thickness, i.e., like Figure 2(d). Equations (13) and (15) would presumably be used for nozzles with local reinforcement, i.e, the Figures 2(a), 2(b), and 2(c).

Nozzles or Openings in Spherical Shells

Figure 4 indicates that, for openings in spherical shells, the stress index of 2.0 is valid only for very small values of ρ. Figure 2 from Reference (10) is reproduced here as Figure 7 to show more clearly the stress index at small value of ρ. Even at the Code limit for unreinforced openings (ρ = 0.1414, $(r/R)\sqrt{R/t_a}$ = 0.1), the stress index according to Figure 7 is about 2.4.

Figure 6 indicates that, for nozzles in spherical shells with $s/S \leq 1.0$ and $r_2 \geq 0.5\ t_a$, there is some small range of ρ for which the index of 2.0 is valid; i.e., up to about 0.4 for D/t_a = 10 or up to about 0.25 for D/t_a = 100.

An optimum method of handling this complex situation is not apparent to the author. A possible Code revision might be

$$\sigma_{max} = 2.0\ (\rho + 1)\ \frac{P_o D_m}{4t_a} \quad , \qquad (16)$$

or

$$\sigma_{max} = 2.0\ \frac{P_o D_m}{4t_r} \qquad (17)$$

Equation (16) is an equation for the straight line portion of "openings" in Figure 5. For openings, it may be slightly unconservative around ρ = 0.1414 and slightly conservative at ρ = 5.0. Figure 6 indicates it would be highly conservative for nozzles with D/t_a = 10, r_2 = 0.5 t_a, but somewhat unconservative

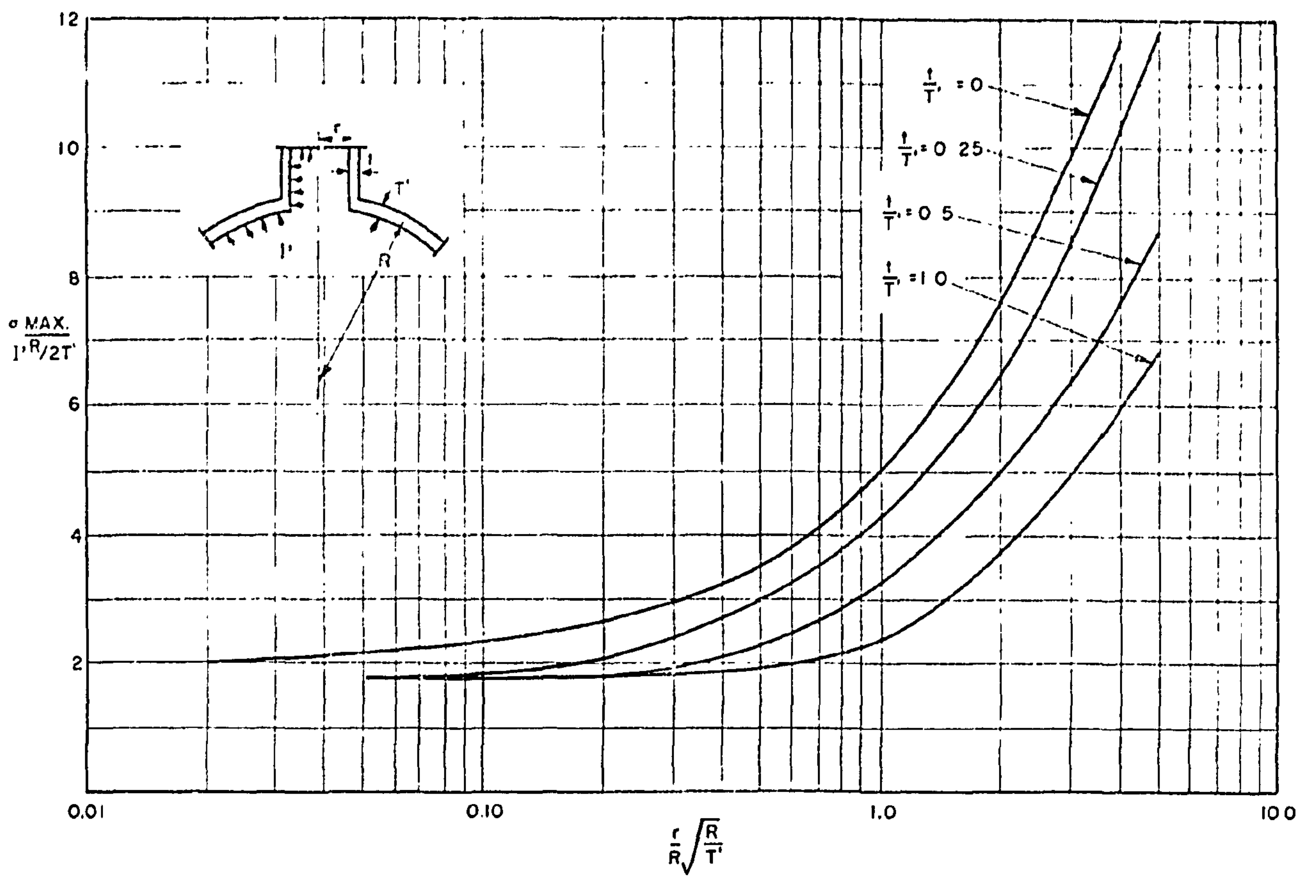

Fig. 2—Maximum stress in sphere for internal pressure (flush nozzles)

FIGURE 7. FIGURE 2 FROM REFERENCE (10)

for nozzles with $D/t_a = 100$, $r_2 \approx 0.5\ t_a$ and ρ between 0.4 and 1.9. Equation (16) would presumably be used for nozzles reinforced by excess shell thickness, i.e., like Figure 2(d). Equation (17) would presumably be used for nozzles with local reinforcement, i.e., like Figures 2(a), 2(b), and 2(c). Reference (12) presents data that indicates Equation (17) is conservative for such nozzles. According to Figure 7, it could be slightly unconservative for openings not requiring reinforcing, e.g., 2.4 versus 2.0 at $\rho = 0.1414$.

RECOMMENDATIONS

We have not yet addressed the question of whether it is better to use t_a or t_r in calculating S_n. The advantage in using t_r is that the present stress indices are conservative; although not necessarily accurate. If we use t_a in calculating S_n, and all reinforcement is obtained by shell thickening, then the present stress indices are not conservative for nozzles in spherical shells or for nozzles in cylindrical shells with $\rho > 0.6$. The use of either t_a or t_r in calculating S_n leads to certain paradoxes. These paradoxes arise because we are trying to represent a very complex problem by a few simple numbers.

A paradox in using t_a is illustrated in Figure 8(a) and ties back to an undefined term in the June, 1975 revision to NB-3338.2(a); i.e., what is meant by "locally" in the clause*: "however, the material which increases the thickness of a vessel wall locally at the nozzle shall not be included in the calculation of these stress components". Figure 8(a) may be considered as showing the longitudinal plane of a nozzle in a cylindrical shell with the shell-course thickened to provide all required reinforcement in the shell wall. The length, L_c represents the Code-undefined boundary between what is local and what is not local. The paradox then is that as the course length, L, is increased and passes through the critical length, L_c, the stress index method gives a step change in the computed stress by the ratio of t_a/t_r; a ratio which is not less than about 2 and can be 3 or 4 or 5. One might guess that the intent of "locally" is that L_c is such that any increase in course length L beyond L_c would not significantly change the stresses in the viscinity of the nozzle. If so, the paradox is even stronger because we would obtain the decrease in calculated stresses by the ratio of t_r/t_a by adding metal where it presumably does not change the stress at all.

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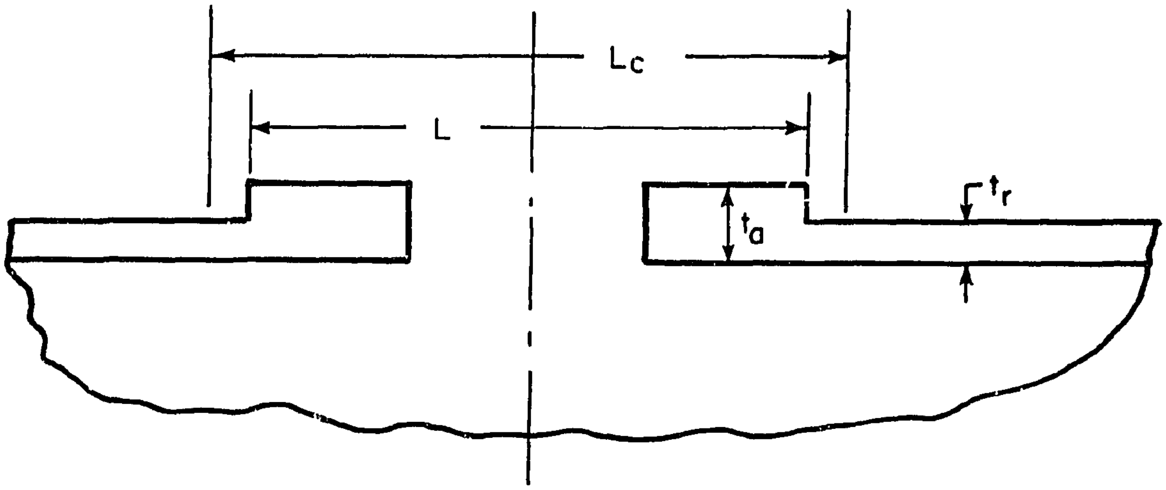
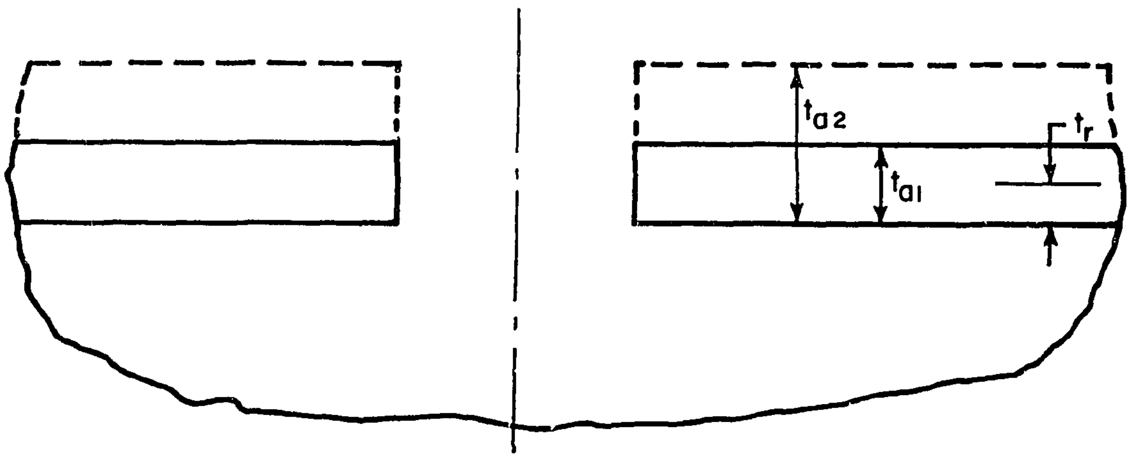
* The author believes that the intent of the sentence would be more evident if the words "these stress components" were replaced by "the membrane stress".



(a) For $L < L_c$, $\sigma = I \times P_o D_m / 2\ t_r$

for $L > L_c$, $\sigma = I \times P_o D_m / 2\ t_a$

Step change in σ by factor t_a / t_r



(b) $\sigma = I \times P_o D_m / 2\ t_r$. Increasing wall thickness from t_{a1} to t_{a2} does not change calculated value of σ.

FIGURE 8: ILLUSTRATION OF PARADOXES USING EITHER t_r OR t_a IN CALCULATING VALUE OF S_n.

A paradox in using t_r is illustrated in Figure 8(b). We start with the assumption that $t_a = t_{a1}$ is sufficiently larger than t_r so that all required reinforcement is in the shell. We obtain a set of stresses by multiplying the indices by $P_o D_m / 2t_r$. Now, we visualize doubling the shell thickness so that $t_a = t_{a2} = 2\, t_{a1}$. However, we still obtain the same set of stresses because t_r has not changed. The paradox is that, by looking at Figure 3, we would expect the stress to decrease by more than 50 percent since ρ has decreased (because t_a has increased) and t_a in the nominal stress $S_n = P_o\, D_m/2\, t_a$ is now $2\, t_{a1}$ rather than t_{a1}.

The second of the above two paradoxes apparently has impressed users of the Code, hence, it appears necessary to use t_a in calculating the nominal stress. The associated penalty is that the present stress indices, when used with $S_n = P_o\, D_m/2\, t_a$, may not be conservative and restrictions need to be placed on their use. In the following, we will assume that the Code will change to the use of t_a rather than t_r in calculating S_n; however, it is not apparent to the author that such a change is necessarily the better choice.

Some possible Code revisions are outlined in this report. These consist of suggestions that:

(a) For nozzles in cylindrical shells, the applicability of the stress indices should be limited to 0.6 and/or

(b) Equations such as (12) through (17) should be incorporated in the Code.

Suggestion (a) is complete in the sense that the stress index tables would not be changed. However, it does not solve the problem of nozzles in spherical shells for which the indices can be unconservative for all values of ρ.

Suggestion (b) is not complete in the sense that it gives only the maximum stress indices, regardless of location or direction. Note that the stress indices in Tables 1, 2, or 3 indicate that the maximum stress for nozzles in cylindrical shells occurs in the longitudinal plane, inside, n-direction. This location/direction is not always the maximum; even the very limited test data of Table 5 shows two contrary examples. Note also, in Table 5, the relationship between the Code index I = -0.2 and test data for ORNL-1, I = -2.88.

To obtain conservative but reasonably accurate indices would probably require developing equations like equation (12) for each of the four significant* indices for nozzles in spherical shells and each of the eight significant* indices for nozzles in cylindrical shells. These twelve equations would have to be incorporated into three places in the Code if we followed the present pattern.

However, a complete solution is even more complex. Note that equation (12) was developed for nozzles with <u>all</u> reinforcement obtained by shell thickening and equation (13) was developed for nozzles with <u>all</u> reinforcement of a local nature and $t_r = t_a$. These represent only the bounds of the total problem for which $1.0 \leq t_a/t_r \leq \infty$.

In order to establish a complete set of indices (even for the bounds) we would need significantly more data than is presently available; and both the additional data and presently available data would have to be reviewed and analyzed in greater depth. Before embarking on such a substantial effort, it is deemed pertinent to ask if it is worth doing.

The first question we ask is: Are there really many nozzles in pressure vessels or piping which are reinforced solely by shell thickening? If by shell thickening we visualize increasing the thickness of an entire vessel from, say t_r = 5" to t_a = 10", the answer is probably: No. Similarly, if we visualize an entire piping system thickness being increased from, say t_r = 2.5" to t_a = 5.0", the answer is probably: No. However, if a hemispherical head contains several nozzles, it might be economically attractive to make $t_a = 2\ t_r$ so as to obtain all of the required reinforcement by shell

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* The significant indices are those for σ_n and σ_t. The indices for σ_r only convey (or should convey) the information that $\sigma_r = -P_o$ on the inside surface, zero on the outside surface. The indices for the stress intensity are immediately derivable from the σ_n and σ_t indices and the obvious fact that $\sigma_r = -P_o$ on the inside, $\sigma_r = 0$ on the outside.

thickening. Similarly, if there are several nozzles in a row around the circumference of a pressure vessel, a thicker shell course might be inserted to obtain all the reinforcement by shell thickening. For piping with a fairly closely spaced row of nozzles, it is common practice to use a "header" where the header thickness is increased as necessary to obtain all reinforcement by shell thickening.

In addition to the above, there is another reason why one may encounter t_a sufficiently greater than t_r so that all required reinforcement is part of the shell. This can occur when the design pressure is relatively low and the required thickness for pressure is inadequate for other loadings and the thickness must be increased. In piping, the standardization of wall thicknesses may also lead to t_a sufficiently greater than t_r so that all required reinforcement is part of the shell. Consider, for example, a 24" pipe made of A-106 Grade B material for a design pressure of 200 psi and design temperature of 100 F with S_m = 20,000 psi. The required wall thickness is about 0.12". However, 0.12" wall A-106 Grade B pipe is not readily available and, unless ordered in many-ton quantities, would probably cost more than 24" std. wt. pipe for which t_a = 0.375" nominal, 0.328" minimum. A 4" branch connection can be placed in this pipe and all required reinforcement will be available by shell thickening.

The author cannot answer the first question in a quantitative sense; but it appears that nozzles reinforced by shell thickening, while perhaps representing only a small fraction of all nozzles, are sufficient in number to warrant developing stress indices applicable thereto.

The second question we ask is: Does anyone ever use the complete set of stress indices? If there is no use (or very little use) of the entire set of stress indices (4 indices for spherical shells, 8 indices for cylindrical shells), there would appear to be *little incentive to produce* a new, and much more complex, complete set of indices.

The third question we ask is: Does anyone ever use the maximum index in the stress index tables or, equivalently, does anyone ever use the C_1 and K_1 indices for "Branch connections per NB-3643" given in Table NB-3683.2-1?

The author would speculate that we might get mostly "nos" to the second question but many "yeses" to the third question. If this speculation is correct, it would seem appropriate to recommend a revision to the C_1 and K_1 index in Table NB-3683.2-1 at this time. However, there is a potentially controversial question involved which needs to be resolved; i.e., how is the stress obtained by the use of the stress indices tables (Tables 1, 2, or 3 herein) to be categorized?

The question of categorization immediately arises when we consider possible revisions to the C_1 and K_1 indices. The C_1 index represents the primary-plus-secondary stress intensity range; S_n in piping terminology, (S_n' here to distinguish it from S_n = nominal stress used elsewhere in this report) and $P_L + P_b + P_e + Q$ in pressure vessel terminology. The C_1K_1 product represents the primary-plus-secondary-plus-peak stress intensity range; S_p in piping terminology, $P_L + P_b + P_e + Q + F$ in pressure vessel terminology. If we were to increase these indices, in analogy to equation (12), should C_1 be increased with K_1 remaining at 1.7, or should C_1 remain at 2.0 and K_1 be increased, or should both C_1 and K_1 be increased?

Categorization is not directly addressed by other Code portions containing stress indices, but we note in NB-3331(b) that NB-3222.2 is considered as satisfied if it is shown that S_n' due to thermal gradients is less than 1.5 S_m. Because the requirement of NB-3222.2 is that $S_n' \leq 3S_m$, this would imply* that S_n' due to any operating pressure range, is less than $1.5S_m$. It should be noted that any changes in the stress index tables do not change this implication. However, this seems to be in conflict with the basic definitions of NB-3213.9, "Secondary Stress" and NB-3213.11, "Peak Stress"; when these definitions are applied to, for example, Model No. 11 of Table 7. The values of σ_t, inside and outside, represent a primary membrane stress plus a linear thru-the-wall bending stress at a gross structural discontinuity. The definition in NB-3213.9 suggests that this entire stress should be categorized as S_n'; i.e., $S_n' = S_p' = 5.93\ P_oD_m/2\ t_a$. This is not necessarily contradictory to the assumption that $S_n' < 1.5\ S_m$. For Model No. 11 of Table 7, $\rho = 2$ and by equation (7), $t_r/t_a = 1/3$. If $P_o = (3/4)\ P_d$, then $S_n' = (5.93)/3 \times (3/4)$

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* There is a paradox in NB-3331(b) in that it seemingly places no limits whatsoever on external loads. Even if the nozzle supported a relief valve, and the thrust on the relief valve were sufficiently high to tear the nozzle out of the vessel the first time it pops, NB-3331(b) would still imply that $S_n < 3\ S_m$ and, indeed, all requirements except the fatigue life evaluation would be satisfied.

$S_m = 1.5\ S_m$. However, P_o is not necessarily limited to (3/4) P_d and there may well be other nozzles where a significant inconsistency may exist. Accordingly, it may be necessary not only to develop new stress indices but also develop guidance as to the appropriate categorization of the stresses obtained from the stress indices.

In view of the preceding discussion, specific recommendations are:

(1) An effort should be undertaken to establish whether the stress indices tables (Tables 1, 2, and 3 herein) are being used, whether the complete set or only the maximum index is used and, if used, how the stresses derived therefrom are categorized.

(2) The Code should be revised as shown in Appendix A.

The recommended Code changes shown in Appendix A are those needed to require the use of the actual wall thickness t_a in calculating the nominal stress S_n in all four portions of the Code concerned with the use of stress indices. Rules are provided which are intended to insure that the stress indices will not be grossly unconservative when used with S_n based on t_a. These rules are:

(a) For nozzles in cylindrical shells with reinforcement entirely or in part from shell thickening, the indices are valid only for $\rho < 0.6$.

(b) For nozzles in spherical shells with reinforcement entirely or in part from shell thickening, the indices are not valid.

(c) For nozzles in either cylindrical or spherical shells, the indices are valid if the area cut out by the opening, $d\ t_a$, is replaced by reinforcement within the boundaries specified by the Code.

The basis of rules (a) and (b) are given in the report; e.g., see Figures 3 and 6. Rule (c) is based on the concept that, if reinforcement for the actual cut-out area is provided, then the stresses obtained by $\sigma = I\ (P_o\ D_m/A\ t_a)$, where A = 2 or 4, are entirely equivalent to a geometry where the reinforcement area is equal to $d\ t_r$ and $\sigma = I\ (P_o\ D_m/A\ t_r)$. Reference (12) gives data which indicates that the stresses for nozzles in spherical shells obtained by $\sigma = I\ (P_o\ D_m/4\ t_r)$, with reinforcement of $d\ t_r$, are generally conservative and never grossly unconservative. A comparable study to Reference (12) for nozzles in cylindrical shells does not exist; however, such data, as is available, indicates that stresses for nozzles in cylindrical shells obtained by $\sigma = I\ (P_o\ D_m/2\ t_r)$ with reinforcement of $d\ t_r$, are also generally conservative and never grossly unconservative.

The recommended Code changes shown in Appendix A are restricted to those changes necessary to insure that stresses will not be grossly underpredicted. There are many additional changes which are necessary or desirable; e.g., corrections of typographical errors, better definitions and improved consistency in terminology between various Code portions.

The contents of NB-3338 present problems which are more than simply editorial. The revised (Winter, 1975 Addenda) reads

> NB-3338.2 Revise subparagraph (a) to read:
>
> (a) The term stress index, as used herein, is defined as the numerical ratio of the stress components, σ_t, σ_n, and σ_r under consideration to the computed membrane stress in the unpenetrated vessel material; however, the material which increases the thickness of a vessel wall locally at the nozzle shall not be included in the calculation of these stress components. When the thickness of the vessel wall is increased over that required to the extent provided hereinafter, the values of r_1 and r_2 in Fig. NB-3338.2-2 shall be referred to the thickened section.

The more-than-editorial problem in NB-3338.2 lies in the definition of the word "locally". As discussed earlier in this report, any definition would probably lead to a step-change in calculated stresses in a paradoxical manner. A possible solution might consist of stress indices which are a function of t_r/t_a. There is another problem in NB-3338.2 in that the user may ask "Which computed membrane stress?" The computed membrane stress is presumably due to pressure loading P_o but, except in a spherical shell, there are an infinity of different computed membrane stresses. In a cylindrical shell, the computed membrane stresses are bounded by $P_oD_m/2\ t_a$ in the circumferential direction and $P_oD_i/4\ t_a$ in the axial direction. The intent, presumably, is to use the maximum computed membrane stress. However, note that Table NB-3338.2(c)-1 includes "formed heads". Which membrane stress is to be used for a nozzle located in the knuckle region of a tori-spherical head where the maximum absolute value of the membrane stress may be negative? The more-than-editorial problem, however, is whether or not the stress indices are applicable (for any definition of the calculated membrane stress) for a nozzle in a formed head other than a spherical head.

If it is established that the stress index tables are being widely used, then further work should be undertaken to provide a better basis for such indices. At such time, it would be desirable to strive for better definitions and consistency in the three portions of the Code concerned with the use of the stress index tables.

SUMMARY

(1) A review of the Code history indicates that considerable confusion has existed and still exists concerning the appropriate value of the nominal stress to be used as a multiplier of the stress indices for pressure loading given in Code tables.

(2) Two definitions of the nominal stress are currently included in the Code; one which uses the actual (nominal) shell-wall thickness, the other uses the minimum required shell-wall thickness. This leads to a significant difference in calculated stress ranges due to pressure loading if the required reinforcement is provided by excess shell-wall thickness.

(3) Test data and theoretical data presented herein indicate that, within the present range of parameters covered by the stress indices for pressure loading, the use of actual shell-wall thickness can be unconservative; in extreme cases by factors of 3 or more for cylindrical shells or 5 or more for spherical shells.

(4) The test data also shows parameter ranges where the use of actual shell-wall thickness is conservative while the use of required wall thickness is excessively conservative. Code revisions have been suggested which permit the use of actual shell-wall thickness, where it is conservative. The revisions also show ways in which the actual shell-wall thickness can be used, with an increased stress index, so as to give conservative calculated stress ranges, but not as excessively conservative as sometimes occurs using the required wall thickness.

(5) This report deals only with maximum stress ranges due to pressure ranges. Tests and theoretical data presented herein show that the Code indices can be significantly inaccurate in relative magnitudes. This is independent of the choice of the nominal-stress-multiplier of the indices.

(6) Recommendations are given in Appendix A for interim revisions to the Code. These provide improved consistency in that all four portions of the Code concerned with stress indices for pressure loading would then use t_a for calculating S_n. Appropriate limits on the applicability of the stress indices are given in Appendix A.

(7) The suggestions in (4) above are not included in the recommended Code revisions because they apply only to the maximum stress index: not to the complete set of indices now given in the Code. Development of a complete set of indices would require substantially more work and the resulting formulas would probably result in a complex set of stress index equations in place of the present 12 (significant) numbers. It has not been established that the complete set of indices are, in fact, being used. It is recommended that the extent of use of the stress index tables be established to see if it is worth while to develop a new and complete set of stress indices.

ACKNOWLEDGEMENTS

The author wishes to express appreciation to members of the Pressure Vessel Research Council Subcommittee on Reinforced Openings and External Loadings for their review of a draft of this report. In particular, the comments of W. L. Greenstreet, Oak Ridge National Laboratory, and of R. W. Schneider, Gulf and Western, Energy Products Group, Bonney Forge Division, contributed significantly to the report.

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APPENDIX A*

Recommended Interim Revisions to the ASME Boiler and Pressure Vessel Code Section III, Subsection NB

1. NB-3338.2(d)

Change (6) to (7) in the first sentence.

Add a new (7) as follows.

(7) The stress indices are not applicable where any of the required reinforcement is obtained from excess shell thickness unless:

(a) The opening is in a cylindrical shell with $d/\sqrt{Dt}$ less than 0.6, or

(b) The reinforcement area calculated in accordance with NB-3332, but not including any area provided by excess shell thickness, is not less than d t F, where t = minimum (considering tolerances) wall thickness of the shell.

2. NB-3339.1

Change (e) to (f) in the first sentence.

Add a new (f) as follows.

(f) The stress indices given in NB-3339.7 are not applicable where any of the required reinforcement is obtained from excess shell thickness unless:

(f1) The opening is in a cylindrical shell with $d/\sqrt{D\ T_s}$ less than 0.6, or

(f2) The reinforcement area, not included any area provided by excess shell thickness, meets the requirements of NB-3339.3 using T_s in place of T.

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* Symbols used herein are those used in various Code portions. The reader of this report must refer to the Code for definitions of the symbols.

3. NB-3339.7(b) Revise as follows

(b) The symbols for the stress components are shown in Figure NB-3338.2-1, and are defined as follows:

$S = P_o\ (D + T_s)/4\ T_s$ for nozzles in spherical vessels or heads

$S = P_o\ (D + T_s)/\ 2\ T_s$ for nozzles in cylindrical vessels

P_o = range of pressure in the cycle under consideration

σ_t = (Remainder without change)

4. Table NB-3683.2-1, footnote (3). Revise by adding the following sentence.

The stress indices for pressure loading (B_1, C_1 and K_1) are not applicable where any of the required reinforcement is obtained by excess thickness of the run pipe wall unless:

(1) the value of $d/\sqrt{D\ t}$ is less than 0.6, where d = diameter of opening, D = inside diameter of run pipe, t = nominal wall thickness of run pipe, or

(2) the reinforcement area calculated in accordance with NB-3643 but not including the area A_3 of Figures NB-3643.3(a), is at least equal to d t, where d = diameter of opening, t = minimum(considering tolerances) wall thickness of the run pipe.

5. NB-3686.1

Change (h) to (i) in the first sentence
Add a new (i) as follows.

(i) The stress indices are not applicable where any of the required reinforcement is obtained from excess shell thickness unless:

(i1) The opening is in a cylindrical shell with $r_m'/\sqrt{R_m T_r}$ less than 0.6, or

(i2) The reinforcement area calculated in accordance with NB-3643, but not including the area A_3 of Figures NB-3643.3(a), is not less than $d\ T_r'$, where d = diameter of opening, T_r' = minimum (considering tolerances) wall thickness of the run pipe.

5. NB-3683.3. Revise as follows

(a) For branch connections in straight pipe, multiply stress indices by:

$$\frac{P\ R_m}{T_r}$$

(b) For branch connections in formed heads, multiply stress indices by:

$$\frac{P\ R_h}{2\ T_h}$$