REVISIONS TO DESIGN BASIS FOR REACTOR VESSELS (WAPD-CE-43)

March 13, 1956

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Page 5, Section III (4). Change to read:

- (4) Peak stress due to high-impact shock. For reactor vessels used in submarines, see the latest revision of E-Spec. 565366, and for surface vessels, see the latest revision of E-Spec. 565913.

Page 6, Section IV. Retain title, but change this entire section to read as follows:

The curve sheets beginning with Fig. 3 show allowable values of S_m , S_p , S_s , and S_v for the following materials:

MATERIAL	MINIMUM MECHANICAL PROPERTIES AT ROOM TEMPERATURE	
	0.2% Yield Stress (psi)	Tensile Strength (psi)
Normalized Mn-Mo Low Alloy Steel ASTM Spec. A-302 Grade B. Plates SA-182 Grade F6. Forgings.	57000	80000
C-Si Steel ASTM Spec. A-212 Grade B. Plates.	38000	70000
Annealed Austenitic Stainless Steel AISI Type 304, .08C max. Plates, seamless tubes, forgings.	30000	75000
Annealed Austenitic Stainless Steel AISI Type 347, .08C max. Plates, seamless tubes, forgings.	33000	75000

The allowable stresses are based primarily on the .2% yield stress, with S_s and S_v at 100% of this value. Consideration has also been given to the successful experience represented by the ASME Boiler Code. Values of S_m are never below the allowable stresses given in the Boiler Code, nor more than 30% above these values. Information on mechanical properties in the

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400°F to 600°F range is meager, and it is expected that revised curves and additional curves will be issued when more data become available. For example, values of S_v should really be based on biaxial fatigue tests in the plastic range, since it is known that ductile materials can withstand limited numbers of strain cycles above the yield point. In the absence of such data, it is necessary to use the yield stress as an allowable limit and thus call for special justification (see V (8) below) whenever this value is exceeded.

Page 7, Section V (2). Change last paragraph to read:

In cases where accurate analytical methods for calculating stresses are not available, the best possible estimates should be made. The allowable stresses given make allowance for variations in material properties but not for inaccuracies in design. Therefore, conservative assumptions should be used in the analysis. Consideration should be given to the use of model tests where known analytical methods are not satisfactory.

Page 7, Section V (3). Omit paragraph (e) of this section, since this subject is discussed in the proposed addition to the Appendix.

Page 13, Appendix. Add section 5 and list of references.

(5) Discontinuity Stress

The calculation of stresses produced by discontinuities such as head-to-shell junctions, openings, nozzles, and changes in plate thickness cannot be accomplished by substitution in formulas. Numerical methods requiring digital computers or else experimental stress analysis with models is required. The references at the end of this section give some results from this type of work. Curves are available which cover useful ranges of the variables for hemispherical heads, conical heads, and flat heads. Work is in progress under the sponsorship of the Pressure Vessel Research Committee on ellipsoidal heads, reducers, and nozzle reinforcements. Until more of this work has been completed, the following information can be used as a guide. In the following, stress values will be given in the form of an index which is the ratio between the peak stress due to the discontinuity and the circumferential membrane stress in the cylindrical shell. Since, for materials listed in Section IV, $S_p \geq 1.5 S_m$, * the most important thing to know about the stress index is whether or not it exceeds 1.5. If it does not exceed 1.5, the plate thickness calculated from eq. (1) above is satisfactory. If it does exceed 1.5, reinforcement may be necessary.

(a) Heads

- (1) Discontinuity stresses produced by the head-to-shell junction of full hemispherical heads may be ignored. (See Ref. (2)).

* AISI Type 304 is an exception to this if the higher values for S_m are allowed.

- (2) In an ellipsoidal head, if the inside height of the head is $1/2$ the inside radius of the vessel, and the head thickness is equal to the vessel thickness, the stress index is 1.4 (See Ref. (1)), which is satisfactory. Information is not yet available on other head heights, but it is expected that satisfactory thicknesses can be found for heights greater than one-half the vessel radius.

Deviations from the true ellipsoidal shape within 1-1/4 percent of the vessel diameter are allowable, but deviations beyond this value should be avoided, since they can result in very high localized stresses.

- (3) Some information is available on peak stresses in conical and toriconical heads. (See Refs. (1) and (3)). It is possible to design them with proportions which produce satisfactory stress indices.
- (4) Flat heads without transition knuckles have excessively high localized stress and should be avoided. (See Ref.(4)).

(b) Openings and Nozzles

The stress concentration factors for an isolated circular hole in a plate subjected to tension are: *

Type of Tension in Plate	No Reinforcement	Reinforced According to ASME Code, with Reinforcement on Both Sides of Plate.
Uniaxial	3.0	1.72
Bi-axial, $S_1 = S_2$	2.0	1.36
Bi-axial, $S_1 = 1/2 S_2$	2.5	1.54

For multiple holes, the stress concentration factors are lower, but the effect of removing load-carrying material (ligament efficiency) must be considered separately.

Since uniaxial tension does not ordinarily occur in a pressure vessel shell, it is apparent that the peak stress due to pressure can be brought within the required limits. In addition to the ASME Code rules, however, it is important to use a smooth transition from shell to nozzle, with part of the reinforcement added to the shell and part to the nozzle, and a generous fillet radius.

* Values calculated from information in Ref. (5).

Stress in the cylindrical shell due to pipe reactions on the nozzle can be calculated by the method described in Ref. (6).

(c) Changes in Plate Thickness

Transitions from one plate thickness to another will be satisfactory if made in accordance with the following rules:

- (1) Centerlines of plates should coincide. Otherwise, bending stress produced by eccentric loading must be added to peak stress.
- (2) Sides of transition plate must have maximum slope of 1:3, and 1:4 is preferable.
- (3) Fillet between thin plate and transition plate must be smooth and have radius of at least 15% of thickness of thin plate.

REFERENCES FOR SECTION 5 OF APPENDIX:

- (1) "Report on the Design of Pressure Vessel Heads"
Welding Research Supplement, Jan. 1953, p. 31-S.
- (2) "The Stresses in a Pressure Vessel with a Hemispherical Head"
by G. W. Watts and H. A. Lang.
Transactions of A.S.M.E., Paper No 52-SA-19.
- (3) "Stresses in a Pressure Vessel with a Conical Head"
by G. W. Watts and H. A. Lang.
Transactions of the A.S.M.E., April 1952, p. 315.
- (4) "The Stresses in a Pressure Vessel with a Flat Head Closure"
by G. W. Watts and H. A. Lang.
Transactions of the A.S.M.E., Aug. 1952, p. 1083.
- (5) "Stress Concentration Design Factors"
by R. E. Peterson,
John Wiley and Sons, 1953.
- (6) "Stresses from Radial Loads and External Moments in Cylindrical Pressure Vessels", by P. P. Bijlaard.
Welding Journal, Research Supplement, Dec. 1955, p.608-S.

Fig. 5

ALLOWABLE STRESSES

Annealed Austenitic Stainless Steel, .08 C max.
AISI Type 304. Plates, Seamless Tubes, and Forgings.

NOTE: The higher values of S_m may be used for applications where permanent strain of about 0.1% can be tolerated.

Fig. 6

ALLOWABLE STRESSES

Annealed Austenitic Stainless Steel, .08 C max.

AISI Type 347. Plates, Seamless Tubes, and Forgings.

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SIGNATURE

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