WESTINGHOUSE ELECTRIC CORPORATION

WAPD-CE-45

ATOMIC POWER DIVISION
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PITTSBURGH 30, PA.

Bureau of Ships Inspection Officer
Westinghouse Atomic Power Division
P. O. Box 1105, Bettis Field
Pittsburgh 30, Pennsylvania

Subject: DESIGN BASIS FOR REACTOR VESSELS

Reference: (a) SSN/551 (551A) Ser. 551A-899, 28 April 1955.
Memorandum issued by Chief, Bureau of Ships

Dear Sir:

This letter transmits WAPD-CE-43 which is the proposed design basis for reactor vessels requested by Mr. P. Clark, Code 1500. It was prepared by the Reactor Vessel Subcommittee of the Reactor Design Committee, made up of the following individuals:

B. F. Langer, Chairman	P. H. Gustafson (Alternate)
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L. Porse (Alternate)	J. F. Gibbons
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J. L. Price (Alternate)	P. H. Seran (Alternate)
V. W. Burton	H. E. Kline (Alternate)

It is proposed that this document be used to accompany all requests for bids on reactor vessels for Naval service. Although differing somewhat in detail, it follows the general principles of the proposed structural design basis for reactor pressure vessels prepared by Code 551 (See Reference a). The allowable membrane stresses, due to pressure loading, are higher than those allowed by the ASME Boiler Code. In order to provide safety at these higher stresses, better inspection is required and more detailed analysis of peak stresses and thermal stresses is specified.

Your approval of this document is requested.

Very truly yours,

B. F. Langer
B. F. Langer
Consulting Engineer
Chairman, Reactor Vessel
Subcommittee

RFL:rs

Enc 1.

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Transmitted by *Betty Forrest*

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DESIGN BASIS FOR REACTOR VESSELS

B. F. Langer

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DESIGN BASIS FOR REACTOR VESSELS

I. PURPOSE AND SCOPE

The purpose of the present document is to specify the design basis which is to be used for reactor pressure vessels for Naval service which are built for Westinghouse Electric Corporation, Bettis Plant. It specifies the types of loading and types of operating conditions which must be considered in evaluating the design, and establishes the limitations which must be imposed on stresses and deformations in order to insure structural reliability. Specific values for loads and operating conditions will be given in the equipment specification for each vessel. Allowable stresses for some materials are given herein, and limits for other materials will be added in subsequent revisions as information becomes available.

For the purposes of this document, the term "reactor vessel" shall be interpreted to include (1) the pressure-containing shell, (2) integral or welded heads, (3) removable heads and closures including flanges and means for fastening and sealing, (4) all openings, reinforcements and nozzles penetrating the shell, heads, or closures, (5) internal structures attached to the vessel such as those required for the control of coolant flow, the protection of the pressure shell from radiation, and the support of the reactor core itself, and (6) external structures attached to the vessel such as those required for handling, operation, servicing and support. The exact point of demarcation between the reactor vessel and the foundation shall be specified for each application.

II. TYPES OF LOADING

In evaluating the structural reliability of a reactor vessel, the effects of the following types of loading should be considered:

- (1) Primary coolant pressure, including transients and differential pressure due to coolant flow.
- (2) External forces at nozzles, foundations and attachments due to:
 - (a) Thermal expansion
 - (b) Hull distortion caused by submergence
 - (c) Weight of reactor vessel or attached equipment
 - (d) Loads during handling and installation

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- (3) Thermal conditions due to internal (gamma) heat generation.
 - (4) Thermal conditions due to steady-state heat flow.
 - (5) Thermal conditions due to normal power changes.
 - (6) Thermal conditions due to abnormal power changes including emergency shut-down and recovery.
 - (7) Thermal conditions due to fastest allowable rate of coolant temperature change.
 - (8) Thermal conditions due to removal of decay heat under emergency conditions (no forced circulation).
 - (9) Thermal effects on components constructed of materials with different thermal expansion coefficients.
 - (10) Mechanical loadings due to shock.

Consideration should be given to the effect of the foregoing types of loading on stress and deformation. The permanent or transient deformations resulting from service conditions must be consistent with safety and operational requirements.

III. TYPES OF STRESS

The maximum shear stress theory will be used as the criterion of strength. It is a little more conservative than the shear energy theory, (Mises criterion), it is easier to apply, and recent experiments indicate that it is more accurate in predicting creep, stress rupture, and yielding of thick-walled cylinders. In order to make the allowable stress limits given herein directly comparable to those given in pressure vessel and piping codes, and also comparable to data obtained from tensile and bending tests, use is made of the "equivalent intensity of combined stress" instead of the maximum shear stress itself. When the maximum shear stress theory is used, the ratio between maximum shear stress and equivalent intensity of combined stress is 0.5. (For the Mises criterion, the maximum value of the corresponding ratio is 0.577). All formulas given herein and all allowable stresses are expressed in terms of the equivalent uniaxial stress, which is merely the algebraic difference between the largest and smallest principal stresses at the point in question.

In considering the structural significance of the loads listed in Section II, it is necessary to distinguish between

- (a) Steady-state loads and cyclic loads,
- (b) Average (membrane) stress and peak stress, and
- (c) Self-relieving and non self-relieving stress.

It is not reasonable to establish a single limiting stress value to cover all of the foregoing categories. It is well known, for example, that repetitive loading can be more damaging than loading which is held constant, and the possibility of failure is dependent on the number of repetitions. Membrane stress* produced by internal pressure must be held to a comparatively low value, whereas a peak stress in a portion of a section can safely be allowed to go higher since the highly stressed material is backed up by other material in regions of lower stress. Thermal stresses are, in general, less damaging than pressure-applied stresses because yielding produces relaxation. An effort has therefore been made to select certain significant groupings of the given load conditions for comparison with allowable stress values. These groupings are as follows:

- (1) Membrane stress due to coolant pressure or any other steady mechanical load. Allowable stress = S_m .
- (2) Peak stress at any location due to pressure or any other steady mechanical load. Allowable stress = S_p .
- (3) Combined steady and alternating peak stress, evaluated in accordance with the Soderberg diagram (see Fig.1) for each specified operating cycle. Limiting value of steady stress component = S_s . Limiting value of alternating stress component = S_v .

Examples of operating cycles which should be investigated are:

- (a) Pressurization and depressurization.
- (b) Heating and cooling of primary coolant at constant pressure.
- (c) Variation of reactor power through full range at constant pressure and temperature of coolant.
- (d) Emergency cooling cycle.

In each cycle investigated, the abscissa plotted on the Soderberg diagram is the mean stress** during the cycle and the ordinate is the difference between the highest stress and the mean stress. When the maximum shear stresses for steady and transient loads do not occur on the same plane, the greatest equivalent intensity of combined stress which occurs under any operating condition shall be limited to a value of $1/2 (S_s + S_v)$.

- (4) Peak stress due to high-impact shock. For allowable stress, see proposed Specification 565366-C.
- (5) Peak stress due to handling and installation loads. Allowable stress = S_p .

* Membrane stress is defined as the average normal stress across a section.

** Mean stress is defined as $1/2$ the sum of the lowest and highest stresses which occur.

IV. ALLOWABLE STRESSES

Material	Minimum Ultimate Strength (S _u)	Minimum Yield Strength .2% (S _y)	S _m	S _p	S _s	S _v
A-302-B (100° to 700°F)	80000	50000	26000	45000	50000	50000
A-212-B (200° to 700°F)	70000	35000	21000	31000	35000	35000
AISI-304 (600°F)	60000	18000	11000	17000	18000	18000
AISI-347 (600°F)	60000	29000	17000	26000	29000	29000

V. GENERAL REQUIREMENTS

- (1) The use of the methods and allowable stresses specified herein is predicated upon the use of inspection and fabrication practices which are more stringent than those called for in the ASME Boiler Code, Sections I and VIII. These requirements are specified in detail in the equipment specification for each vessel, and involve cleanliness, surface finish, welding procedure, acceptable types of welded joints, radiographic and liquid penetrant weld inspection and ultrasonic inspection of base material. Except when superseded by information contained herein or in the applicable equipment specification, the design, fabrication, and testing of reactor vessels shall be in accordance with the latest issue of the ASME Boiler Code, Section I - Power Boilers.
- (2) A detailed stress analysis of all major structural components shall be prepared considering all significant conditions of operation and in sufficient detail to permit independent checking. Peak stresses as specified in Section III shall include:
 - (a) Stress at inside and outside of vessel wall due to internal pressure and thermal effects.
 - (b) Stress at discontinuities, such as head-to-shell junctions, openings, and changes in section.
 - (c) Stresses in flanges, seals, bolts and other elements of closure assemblies.
 - (d) Stresses due to external loads, such as pipe and foundation reactions.
- (3) Effects of stress concentration.

The foregoing is not intended as a complete list of all the stresses which are to be calculated. Special conditions, such as internally or externally mounted equipment, hydraulic forces, etc., may require the calculation of other stresses not specifically listed.

In cases where accurate analytical methods for calculating stresses are not available, the best possible estimates may be made, based on conservative assumptions. Consideration should be given to the use of model tests where known analytical methods are not satisfactory.

- (3) The intent of this structural design basis is to emphasize the goal of "uniformly proportioned design" in order that proper safety can be achieved and operational requirements satisfied. It must be recognized that achievement of this goal requires close attention to all design and fabrication details. For example:
- (a) Design details of structural components should provide a practicable minimum of local structural discontinuities.
 - (b) Bolts and other members subjected to shock loads should be designed for a practicable maximum of energy absorption capacity.
 - (c) Since bolts loaded in shear and shear pins are inherently incapable of absorbing substantial energy by plastic deformation, their use should be avoided unless it can be shown that yielding will not occur under the most extreme non-lethal shock conditions.
 - (d) Abrupt changes in contour due to forming, fit-up, etc., should be avoided.
 - (e) Where unequal thicknesses are involved in vessels, a maximum slope of 1:4 is recommended.
- (4) During normal operation, the primary system is subject to cyclic variation of both pressure and temperature. The design pressure, specified in the applicable equipment specification is at least as high as (generally 5 to 10% higher than) the maximum estimated surge, and fluctuations in pressure during normal operation are small (less than 15%). Therefore, the design pressure shall be used in stress calculations and the effect of fluctuations below this value may be ignored.
- (5) Weld efficiencies shall be taken as 100% provided the following conditions are met:
- (a) Full weld penetration.
 - (b) Weld ground smooth and continuous with 1/16" maximum reinforcement.
 - (c) Full radiographic and magnetic powder or penetrant oil inspection of completed weld plus penetrant oil inspection of first pass.

- (6) When austenitic cladding is used on ferritic base metal, no structural strength shall be attributed to the cladding. Stresses due solely to differences in thermal coefficients of expansion between the cladding and the base metal shall not be considered.
- (7) Stresses due to vibration of vessel components have not been specifically listed in this design basis, on the assumption that serious vibrations cannot be tolerated. Conditions conducive to resonant or self-induced vibration must be eliminated.
- (8) It is recognized that, in some cases, special considerations such as infrequency of load application, special information on material properties or results of component tests, will make calculated stresses acceptable which are higher than those specified herein. In such cases a detailed study of the component, with explanation of the special considerations, may be submitted to the cognizant engineer, Bettis Plant, with a request for a waiver.

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FIGURE I

Soderberg Diagram

A P P E N D I X

Basic Formulas for Stress Calculation

- Let
- S = equivalent intensity of combined stress (lb. per in.²)
 - D = inside diameter of pressure vessel (in.)
 - P = internal pressure (lb. per in.²)
 - T = temperature relative to inner wall (°F)
 - E = modulus of elasticity (lb./in.²)
 - a = thickness of vessel wall (in.)
 - x = distance from inner wall (in.)
 - β = linear absorption coefficient ($\frac{1}{\text{in.}}$)
 - k = thermal conductivity (BTU/hr. in. °F)
 - q = heat generation per unit volume (BTU/hr. in.³)
 - α = linear coefficient of thermal expansion (in./in.°F)
 - ν = Poisson's ratio (dimensionless).
 - ΔT = temperature decrease at inner wall
 - t = time for temperature change (hr.)
 - h = film coefficient (BTU/ft.²hr.°F)
 - ϵ = diffusivity (ft.²/hr.)
 - $\tau = \frac{\epsilon t}{a^2}$

Subscripts refer to location in wall thickness:

- o means inside
- x means any location
- a means outside

1. Membrane stress due to internal pressure (Section III - (1))

$$S = \frac{P(D + a)}{2a} \text{ ----- (1)}$$

S must be less than S_m .

2. Peak stress at inside of vessel wall due to internal pressure (Section III-(2))

$$S = \frac{2P \left(\frac{D}{2} + a \right)^2}{a(D + a)} \text{ ----- (2)}$$

S must be less than S_p .

3. Thermal stress due to internal heat generation

CASE I

Inner wall cooled to zero temperature ($T_o = 0$)

Outer wall perfectly insulated.

Heat generation varies in accordance with $q = q_o e^{-\beta x}$

The temperature at any point in the wall is

$$T_x = \frac{q_o}{k\beta^2} \left[1 - e^{-\beta x} - \beta x e^{-\beta a} \right]$$

The maximum temperature is at the outer wall and is

$$T_a = \frac{q_o}{k\beta^2} \left[1 - (\beta a + 1) e^{-\beta a} \right]$$

The stress at any point in the wall is

$$S_x = \frac{E\alpha q_o}{2(1-\mu)k\beta^2} \left[-\frac{2}{\beta a} + 2e^{-\beta x} + \left(\frac{2}{\beta a} - (\beta a + 2\beta x) \right) e^{-\beta a} \right]$$

The maximum stress is at the inner wall and is

$$S_o = \frac{E\alpha q_o}{2(1-\mu)k\beta^2} \left[2 - \frac{2}{\beta a} + \left(\frac{2}{\beta a} + \beta a \right) e^{-\beta a} \right]$$

The stress at the outer wall is

$$S_a = \frac{E\alpha q_o}{2(1-\mu)k\beta^2} \left[-\frac{2}{\beta a} + (\beta a + 2 + \frac{2}{\beta a}) e^{-\beta a} \right]$$

CASE IIInner wall cooled to zero temperature ($T_0 = 0$)Outer wall cooled to known temperature, T_a .(T_a may be either positive or negative)Heat generation varies in accordance with $q = q_0 e^{-\beta x}$

The temperature at any point in the wall is

$$T = \frac{T_a x}{a} + \frac{q_0}{k\beta^2} \left[1 - \frac{x}{a} + \frac{x_0}{a} e^{-\beta a} - e^{-\beta x} \right]$$

The maximum temperature occurs at the location

$$x_{\max.T} = \frac{1}{\beta} \ln \left[\frac{q_0 \beta a}{q_0 (1 - e^{-\beta a}) - k\beta^2 T_a} \right]$$

The value of this maximum temperature is

$$T_{\max.} = \frac{q_0}{k\beta^2} + \frac{1}{\beta a} \left[T_a - \frac{q_0}{k\beta^2} (1 - e^{-\beta a}) \right] \left[1 + \ln \frac{q_0 \beta a}{q_0 (1 - e^{-\beta a}) - k\beta^2 T_a} \right]$$

The stress at any point in the wall is

$$S_x = \frac{E\alpha}{2(1-\mu)} \left[T_a \left(1 - \frac{2x}{a} \right) + \frac{q_0}{k\beta^2} \left(\left(1 + \frac{2}{\beta a} - \frac{2x}{a} \right) (e^{-\beta a} - 1) + 2e^{-\beta x} \right) \right]$$

The stress at the inner wall is

$$S_0 = \frac{E\alpha}{2(1-\mu)} \left[T_a + \frac{q_0}{k\beta^2} \left(1 - \frac{2}{\beta a} + e^{-\beta a} \left(\frac{2}{\beta a} + 1 \right) \right) \right]$$

The stress at the outer wall is

$$S_a = \frac{E\alpha}{2(1-\mu)} \left[-T_a + \frac{q_0}{k\beta^2} \left(1 - \frac{2}{\beta a} + e^{-\beta a} \left(\frac{2}{\beta a} + 1 \right) \right) \right]$$

4. Thermal stress produced by temperature transients.

When a vessel wall is insulated on the outside and is in contact with a fluid on the inside, the stress on the inside due to change in fluid temperature is

$$S = \frac{E \Delta t}{1 - \mu} \times \text{factor (lb./in.}^2\text{)}$$

Values of the stress factor for use in this formula are found in Figure 2.

B. F. Langer
10/18/55

FIGURE 2

STRESS FACTOR FOR THIN WALL