

**FORM NPP-1 CERTIFICATE HOLDER'S DATA REPORT FOR FABRICATED
NUCLEAR PIPING SUBASSEMBLIES***
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of ⁽⁵⁶⁾

1. Fabricated and certified by _____⁽¹⁾
(name and address of NPT Certificate Holder)

2. Fabricated for _____⁽²⁾
(name and address of Purchaser)

3. Location of installation _____⁽³⁾
(name and address)

4. Type _____⁽⁶⁾ _____⁽⁷⁾ _____⁽⁸⁾ _____⁽⁹⁾ _____⁽¹⁰⁾
(Certificate Holder's serial no.) (CRN) (drawing no.) (National Bd. no.) (year built)

5. ASME Code, Section III, Division 1: _____⁽¹¹⁾ _____⁽¹²⁾ _____⁽¹³⁾ _____⁽¹⁴⁾
(edition) [Addenda (if applicable) (date)] (class) (Code Case no.)

6. Shop hydrostatic test _____⁽³⁴⁾ at _____⁽³³⁾ (if performed)

7. Description of piping _____⁽⁵¹⁾ _____⁽⁵⁵⁾ _____⁽¹⁵⁾

8. Certificate Holder's Data Reports properly identified and signed by commissioned inspectors have been furnished for the following items of this report _____⁽⁸⁰⁾

9. Remarks _____⁽⁴⁶⁾

⁽⁶⁹⁾ **CERTIFICATE OF SHOP COMPLIANCE**

We certify that the statements made in this report are correct and that the fabrication of the described piping subassembly conforms to the rules for construction of the ASME Code, Section III, Division 1.

NPT Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ Signed _____
(NPT Certificate Holder) (authorized representative)

⁽⁶⁹⁾ **CERTIFICATE OF SHOP INSPECTION**

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have inspected the piping subassembly described in this Data Report on _____, and state that to the best of my knowledge and belief, the Certificate Holder has fabricated this piping subassembly in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the piping subassembly described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ Commission _____
(Authorized Nuclear Inspector) [National Board Number and Endorsement]⁽⁶⁷⁾

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is 8½ × 11; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

FORM NPP-1 (Back — Pg. 2 of 56)

Certificate Holder's Serial No. 6

10. Description of field fabrication 51 55

11. Pneu., hydro., or comb. test pressure 34 at temp. 33 (if performed)

70

CERTIFICATE OF FIELD FABRICATION COMPLIANCE

We certify that the statements made in this report are correct and that the field fabrication of the described piping subassembly conforms with the rules for construction of the ASME Code, Section III, Division 1.

NPT Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ Signed _____
(Certificate Holder) (authorized representation)

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CERTIFICATE OF FIELD FABRICATION INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____

of _____ have compared the statements in this Data Report with the described piping subassembly and state that parts referred to as data items _____, not included in the Certificate of Shop Inspection, have been inspected by me on _____ and that to the best of my knowledge and belief the Certificate Holder has fabricated this piping subassembly in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the piping subassembly described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ Commission _____
(Authorized Nuclear Inspector) [National Board Number and Endorsement] 67

FORM NPV-1 CERTIFICATE HOLDER'S DATA REPORT FOR NUCLEAR PUMPS OR VALVES*
As Required by the Provisions of the ASME Code, Section III, Division 1

[illegible]

(07/10)

2013 SECTION III APPENDICES

FORM NPV-1 (Back — Pg. 2 of 56)

Certificate Holder's Serial No. 6

8. Design conditions 102 (pressure) 106 (temperature) or valve pressure class 81

9. Cold working pressure 103

10. Hydrostatic test 105 . Disk differential test pressure 98

11. Remarks

CERTIFICATION OF DESIGN

Design Specification certified by 49 P.E. State Reg. no.
Design Report certified by 50 P.E. State Reg. no.

69

CERTIFICATE OF COMPLIANCE

We certify that the statements made in this report are correct and that this pump or valve conforms to the rules for construction of the ASME Code, Section III, Division 1.

N Certificate of Authorization No. Expires

Date Name (N Certificate Holder) Signed (authorized representative)

66

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by
of have inspected the pump, or valve, described in this Data Report on
, and state that to the best of my knowledge and belief, the Certificate Holder has constructed this pump, or valve,
in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the component described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date Signed (Authorized Nuclear Inspector) Commission 67 (National Board Number and Endorsement)

**FORM NV-1 CERTIFICATE HOLDER'S DATA REPORT FOR PRESSURE OR VACUUM
RELIEF VALVES***

As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of 56

1. Manufactured and certified by _____ 1
(name and address of NV Certificate Holder)

2. Manufactured for _____ 2
(name and address of Purchaser)

3. Location of installation _____ 3
(name and address)

4. Valve _____ 104 Orifice size _____ 82 Nom. inlet size _____ 83 Outlet size _____ 83
(model no., series no.)

5. ASME Code, Section III, Division 1 _____ 11 _____ 12 _____ 13 _____ 14
(edition) [Addenda (if applicable) (date)] (class) (Code Case no.)

6. Type _____ 84 _____ 85 _____ 86 _____ 87 _____ 34 at _____ 33
(spring, pilot, or power operated) (set pressure) (blowdown) (rated temp.) (hydro. test., inlet)

7. Identification _____ 6 _____ 7 _____ 8 _____ 9 _____ 10
(Cert. Holder's serial no.) (CRN) (drawing no.) (National Bd. no.) (year built)

8. Control ring settings _____ 109

9. Pressure retaining items

	Serial No. or Identification	Material Spec., Including Type or Grade	Tensile Strength
Body	6	15	16
Bonnet or Yoke			
Support Rods			
Nozzle			
Disk			
Spring Washers			
Adjusting Screws			
Spindle			
Spring			
Bolting			
Other Items			

10. Relieving capacity _____ 86 @ _____ 85 overpressure as certified by the National Board _____ 86
(steam or fluid) (date)

11. Remarks _____ 46

CERTIFICATION OF DESIGN

Design Specification certified by _____ 49 P.E. State _____ Reg. no. _____
Design Report certified by _____ 50 P.E. State _____ Reg. no. _____

69

CERTIFICATE OF COMPLIANCE

We certify that the statements made in this report are correct and that this valve conforms to the rules for construction of the ASME Code, Section III, Division 1.

NV Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ Signed _____
(NV Certificate Holder) (authorization representative)

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is 8 1/2 x 11; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

FORM NV-1 (Back — Pg. 2 of 58)

Certificate Holder's Serial No. 6

68

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by

of have inspected the valve described in this Data Report on and state that to the best of my knowledge and belief, the Certificate Holder has constructed this valve in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the component described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date Signed (Authorized Nuclear Inspector) Commission 67 [National Board Number and Endorsement]

FORM NCS-1 CERTIFICATE HOLDER'S DATA REPORT FOR CORE SUPPORT STRUCTURES*
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of (56)

1. Manufactured and certified by _____ (1)
(name and address of N Certificate Holder)

2. Manufactured for _____ (2)
(name and address of Purchaser)

3. Location of installation _____ (3)
(name and address)

4. Type _____ (39) _____ (6) _____ (7) _____ (8) _____ (9) _____ (10)
(structure) (C.H.'s serial no.) (CRN) (drawing no.) (National Bd. no.) (year built)

5. ASME Code, Section III, Division 1 _____ (11) _____ (12) _____ (13) _____ (14)
(edition) [Addenda (if applicable) (date)] (class) (Code Case no.)

6. Manufactured in accordance with Specification _____ (78) Rev. _____ Date _____ (78)

7. List of Drawings (with last revision and date) _____
(Design Report or Load Capacity Data Sheet)

8. Remarks _____ (46)

CERTIFICATION OF DESIGN

Design specification certified by _____ (49) P.E. State _____ Reg. no. _____
 Design report certified by _____ (50) P.E. State _____ Reg. no. _____

(69)

CERTIFICATE OF INTERNAL STRUCTURES

The undersigned, having a valid Certification of Authorization, certify that the construction of the internal structures will not adversely affect the integrity of the core support structures.

N Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ Signed _____
(N Certificate Holder) (authorized representative)

(69)

CERTIFICATE OF COMPLIANCE

We certify that the statements made in this report are correct and that this set of core support structures conforms to the rules of construction of the ASME Code, Section III, Division 1.

N Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ Signed _____
(N Certificate Holder) (authorized representative)

(68)

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____

have inspected the core support structure described in this Data Report on _____, and state that to the best of my knowledge and belief, the Certificate Holder has constructed this item in accordance with the ASME Code, Section III, Division 1. By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the item described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ Commission _____ (67)
(Authorized Nuclear Inspector) [National Board Number and Endorsement]

* Supplemental sheets in the form of lists, sketches, or drawings may be used provided: (1) size is 8 1/2 × 11; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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*Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is 8 1/2 x 11; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

FORM NF-1 (Back — Pg. 2 of 56)

Support I.D. Nos. 64 through

CERTIFICATION OF DESIGN

Design Specification certified by 49 P.E. State Reg. No.

Design Report certified by 50 P.E. State Reg. No.

69

CERTIFICATION OF COMPLIANCE

We certify that the statements made in this report are correct and that these supports conform to the rules for construction of the ASME Code, Section III, Division 1.

NPT Certificate of Authorization No. Expires

Date Name Signed (NPT Certificate Holder) (authorized representative)

68

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by of have inspected the supports described in this Data Report on , and state that to the best of my knowledge and belief, the Certificate Holder has constructed these supports in accordance with the ASME Code, Section III, Division 1.

By signing this certificate, neither the inspector nor his employer makes any warranty, expressed or implied, concerning the component supports described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date Signed (Authorized Nuclear Inspector) Commission [National Board Number and Endorsement]

(07/11)

**FORM NM-1 CERTIFICATE HOLDER'S DATA REPORT FOR
TUBULAR PRODUCTS AND FITTINGS WELDED WITH FILLER METAL***
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of 56

1. Manufactured and certified by _____ (1)
(name and address of NPT Certificate Holder)

2. Manufactured for _____ (2)
(name and address of Purchaser)

3. Location of installation _____ (3)
(name and address)

4. Identification _____ (79) _____ (8) _____ (9) _____ (10)
(lot, etc.) (drawing no.) (National Bd. no.) (year built)

5. ASME Code, Section III, Division 1 _____ (11) _____ (12) _____ (13) _____ (14)
(edition) [Addenda (if applicable) (date)] (class) (Code Case no.)

6. Mat'l. Spec. _____ (15) _____ (16) _____ (17) _____ (19) _____ (20)
(SA or SB and no.) (tensile strength) (nominal thickness) (diameter ID) (pipe length and fitting type)

7. Shop hydrostatic test pressure _____ (34) at _____ (33) (if performed)

8. Remarks _____ (46)

(69)

CERTIFICATE OF COMPLIANCE

We certify that the statements made in this report are correct and that the products defined in this report conform to the rules for construction of the ASME Code, Section III, Division 1. The radiographic film and a radiographic report showing film location are attached to the Certified Material Test Reports provided for the material covered by this report.

NPT Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ Signed _____
(NPT Certificate Holder) (authorized representative)

(69)

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____

of _____ have inspected the products described in this Data Report on _____, and state that to the best of my knowledge and belief, the Certificate Holder has constructed this product in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the products described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ Commission _____ (67)
(Authorized Nuclear Inspector) [National Board Number and Endorsement]

*Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is 8 1/2 × 11; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and number of sheets is recorded at the top of this form.

FORM NS-1 CERTIFICATE HOLDER'S CERTIFICATE OF CONFORMANCE FOR WELDED SUPPORTS*
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of 56

1.	Manufactured by _____ <div style="text-align: center; font-size: small;">(name and address of NS Certificate Holder)</div>																																												
2.	Manufactured for _____ <div style="text-align: center; font-size: small;">(name and address of Purchaser)</div>																																												
3.	Location of installation _____ <div style="text-align: center; font-size: small;">(name and address)</div>																																												
4. Type	_____																																												
	(describe) (Design Report or Load Capacity Data Sheet) (Year built)																																												
5. ASME Code, Section III, Division 1	_____ <div style="display: flex; justify-content: space-between; font-size: x-small;"> (edition) [Addenda (if applicable) (date)] (class) (Code Case no.) </div>																																												
6. Identification	<table border="0" style="width: 100%; font-size: small;"> <thead> <tr> <th style="text-align: left; width: 25%;">(a) Support I.D. No.</th> <th style="text-align: left; width: 25%;">(b) Material Specification No.</th> <th style="text-align: left; width: 25%;">(c) Canadian Registration No.</th> <th style="text-align: left; width: 25%;">(d) Applicable Drawings With Last Rev. & Date</th> </tr> </thead> <tbody> <tr><td>(1) _____</td><td>_____</td><td>_____</td><td>_____</td></tr> <tr><td>(2) _____</td><td></td><td></td><td></td></tr> <tr><td>(3) _____</td><td></td><td></td><td></td></tr> <tr><td>(4) _____</td><td></td><td></td><td></td></tr> <tr><td>(5) _____</td><td></td><td></td><td></td></tr> <tr><td>(6) _____</td><td></td><td></td><td></td></tr> <tr><td>(7) _____</td><td></td><td></td><td></td></tr> <tr><td>(8) _____</td><td></td><td></td><td></td></tr> <tr><td>(9) _____</td><td></td><td></td><td></td></tr> <tr><td>(10) _____</td><td></td><td></td><td></td></tr> </tbody> </table>	(a) Support I.D. No.	(b) Material Specification No.	(c) Canadian Registration No.	(d) Applicable Drawings With Last Rev. & Date	(1) _____	_____	_____	_____	(2) _____				(3) _____				(4) _____				(5) _____				(6) _____				(7) _____				(8) _____				(9) _____				(10) _____			
(a) Support I.D. No.	(b) Material Specification No.	(c) Canadian Registration No.	(d) Applicable Drawings With Last Rev. & Date																																										
(1) _____	_____	_____	_____																																										
(2) _____																																													
(3) _____																																													
(4) _____																																													
(5) _____																																													
(6) _____																																													
(7) _____																																													
(8) _____																																													
(9) _____																																													
(10) _____																																													
7. Remarks	<div style="text-align: center; margin-bottom: 5px;">_____</div> <div style="border: 1px solid black; height: 150px; min-height: 100px;"></div>																																												

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is 8½ × 11; (2) information in items 1 through 4 on this Certificate of Conformance is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

(07/10)

2013 SECTION III APPENDICES

FORM NS-1 (Back — Pg. 2 of 56)

Support I.D. Nos. 64 through

CERTIFICATION OF DESIGN

Design Specification certified by 49 P.E. State Reg. No.
Design Report certified by 50 P.E. State Reg. No.

69

CERTIFICATE OF CONFORMANCE

We certify that the statements made in this report are correct and that these supports conform to the rules for construction of the ASME Code, Section III, Division 1.

NS Certificate of Authorization No. Expires

Date Name (NS Certificate Holder) Signed (authorized representative)

(07/10)

**FORM C-1 CERTIFICATE HOLDER'S DATA REPORT
FOR CONCRETE REACTOR VESSELS AND CONTAINMENTS***
As Required by the Provisions of the ASME Code, Section III, Division 2

Pg. 1 of 56

1. Constructed and certified by _____				(1)	(name and address of N Certificate Holder)																			
2. Constructed for _____				(2)	(name and address of Owner)																			
3. Location _____				(3)	(name and address)																			
4. _____		(6)	_____		(7)	_____		(9)	_____		(10)													
(Certificate Holder's serial no.)			(CRN)			(National Bd. no.)			(year built)															
Type _____				(5)	_____				(88)	_____		(8)												
(reactor vessel or containment)					(construction reinforced or prestressed concrete)					(drawing no.)														
5. ASME Code, Section III, Division 2 _____				(11)	_____				(12)	_____		(13)	_____		(14)									
(edition)					[Addenda (if applicable) (date)]					(class)			(Code Case no.)											
6. Design conditions																								
(a) Drawing and revision _____				(31)	Design pressure _____				(32)	Design temp. _____														
(b) Design specification no. _____				(79)	Revision _____				(91)	Date _____														
(c) Foundation type _____				(91)	(soil, rock bearing, piles, etc.)				(90)	Dome type _____				(90)										
(spherical, elliptical, flat, etc.)																								
7. Nominal dimensions _____				(19)	_____				(17)	_____				(20)	_____				(89)	_____				(17)
(inside diameter)					(wall thickness)					(foundation top to springline height)					(dome height)					(dome thickness)				
8. Construction specifications (list all construction specifications) (77)																								
Title				No.				Revision				Date												
_____				_____				_____				_____												
_____				_____				_____				_____												
_____				_____				_____				_____												
9. Type of post-tensioning system																								
(a) Tendon material _____				(15)	Min. tensile _____				(16)	Diameter or size _____				(38)	Corrosion protection _____				(grout, grease, etc.)					
(b) Fabricated by _____				(1)	Installed by _____				(1)															
10. Liner and sleeves (if within constructor's responsibility)																								
(a) Liner material _____				(15)	Min. yield _____				(107)	Bottom thickness _____				(17)										
Wall thickness _____				(17)	Dome thickness _____				(17)															
(b) Sleeve material _____				(15)	Min. yield _____				(107)	Number and sizes _____				(38)										
11. Parts (fabricated, installed, or constructed by others)																								
List each item and attach copy of Certificate Holder's Data Report																								
Part	Drawing & Rev.	Name of CH	Manufacturer's Serial No.	CRN	National Bd. No.	Year Built																		
(63)	(8)	(1)	(6)	(7)	(9)	(10)																		
_____	_____	_____	_____	_____	_____	_____																		
_____	_____	_____	_____	_____	_____	_____																		
_____	_____	_____	_____	_____	_____	_____																		
_____	_____	_____	_____	_____	_____	_____																		
12. Additional material excluding welding material																								
Name of Supplier				Material Specification				Dimensions																
(59)				(15)				(20)																
_____				_____				_____																
_____				_____				_____																

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is 8 1/2 x 11; (2) information in items 1 through 6 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

2013 SECTION III APPENDICES

FORM C-1 (Back — Pg. 2 of 56)

Certificate Holder's Serial No. 6

13. Construction Report No. 93 Date

14. List of Penetrations 96

Attach a complete list of penetrations (i.e., personnel locks, equipment hatch, electrical, etc.) to this report. State the type, size, manufacturer, and serial number.

15. Test Pressure 34 Date tested

16. Remarks: 46

CERTIFICATION OF DESIGN

Design Specification on file at 109
Design Specification certified by 49 P.E. State Reg. no.
Design Report on file at 48
Design Report certified by 50 P.E. State Reg. no.

69

DESIGNER'S REPORT OF CERTIFICATION

I, the undersigned, representing the Designer and employed by have examined and evaluated the Construction Report for the component described in this Data Report. Following evaluation, the Construction Report has been certified and to the best of my knowledge and belief the Constructor has constructed this component in accordance with the rules of the ASME Code, Section III, Division 2, and the construction specification listed herein, and these construction specifications meet the requirements of the Design Specification.

Date Signed (authorized representative)

P.E. State Reg. no.

70

CERTIFICATE OF CONSTRUCTION COMPLIANCE

We certify that the statements made in this report are correct and that all details of materials, construction, and workmanship of this component conform to the rules for construction of the ASME Code, Section III, Division 2, and the Construction Specifications listed herein.

Certificate of Authorization No. Expires

Date Constructor (N Certificate Holder) Signed (authorized representative)

71

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by of , have inspected the concrete reactor vessel or containment described in this Constructor's Data Report and state that to the best of my knowledge and belief this component has been constructed in accordance with the ASME Code, Section III, Division 2.

By signing this certificate neither the Authorized Inspector nor his employer makes any warranty, expressed or implied, concerning the component described in this report. Furthermore, neither the Authorized Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date Signed (Authorized Nuclear Inspector) Commission 67 [National Board Number and Endorsement]

(07/11)

Table V-1000
Guide for Preparation of Data Report Forms

Applies to Form														Ref. to Circled Nos. in Forms	Instructions for Completion
N-1	N-1A	N-2	N-3	N-5	N-6	NPP-1	NPV-1	NV-1	NCS-1	NS-1	NM-1	C-1			
X	X	X	X	X	X	X	X	X	X	X	X	X	X	(1)	Name and address as listed on ASME Certificate of Authorization.
X	X	X	X	X	X	X	X	X	X	X	X	X	X	(2)	Name and address of purchaser.
X	X	X	X	X	X	X	X	X	X	X	X	X	X	(3)	Name, address, and unit number of power plant where item is to be installed.
X	X	X	(4)	Type of installation intended (horizontal or vertical).
X	X	(5)	Description or application of vessel (reactor vessel tank, jacketed, heat exch., containment, etc.).
X	X	X	X	X	X	X	X	X	X	X	X	(6)	Item serial no.
X	X	X	X	X	X	X	X	X	X	X	...	X	X	(7)	Canadian Registration No. for item.
X	X	X	...	X	X	X	X	X	X	X	X	X	X	(8)	Indicate drawing numbers, including applicable revision number, that cover general assembly and list of materials. For Canadian registered vessels, the number of the drawing approved by provincial authorities.
X	X	X	X	X	X	X	X	X	X	...	X	X	X	(9)	National Board Number from Certificate Holder's series of Numbers to be stamped sequentially without skips or gaps.
X	X	X	X	X	X	X	...	X	X	...	X	X	X	(10)	Shall be the year certified by the Inspector on the Certificate Holder's Data Report.
X	X	X	...	X	X	X	X	X	X	X	X	X	X	(11)	ASME Code, Section III, Edition used for construction (e.g., 1986, etc.).
X	X	X	...	X	X	X	X	X	X	X	X	X	X	(12)	ASME Code, Section III, Addenda used for construction (e.g., A86, A87, etc.).
X	X	X	...	X	X	X	X	X	X	X	X	X	X	(13)	ASME Code Section III, Class 1, 2, 3, MC, CB, CC, or CS.
X	X	X	...	X	X	X	X	X	X	X	X	X	X	(14)	All Code Case Numbers and revisions used for construction, including design, fabrication, and materials used, must be listed. Where more space is needed use the "Remarks" section or list on a supplemental page. Code Cases used by Material Manufacturers and Material Suppliers shall be listed on the Data Report.
X	X	X	...	X	X	X	X	X	...	X	X	X	X	(15)	Material Specification Number. Show complete specification number and grade of actual material used. Material is to be as designated in ASME Code, Section III or as permitted in Code Cases.
X	X	X	X	X	X	X	X	(16)	For the "N" forms, Section II, Part D, "Min. Tensile Strength." For Form C-1, the ASTM specifications per Section III, Div. 2, CC-2400.
X	X	X	X	X	X	X	(17)	Nominal thickness.
X	X	X	X	(18)	Minimum thickness as specified by design.
X	X	X	X	X	X	X	(19)	Inside diameter.
X	X	X	...	X	X	X	X	X	(20)	Length or height overall, including heads.
X	X	X	(21)	Type of longitudinal joint (single butt welded or double butt welded joint).
X	X	X	(22)	Indicate postweld heat treatment (yes or no).
X	X	X	(23)	Indicate degree of radiographic examination (full, partial, spot, or none).
X	X	X	(24)	Weld joint efficiency as determined by design (%).
X	X	X	(25)	Type of girth joint (single butt welded or double butt welded).
X	X	X	(26)	Number of sections (courses) joined by girth welds.

Table V-1000
Guide for Preparation of Data Report Forms (Cont'd)

Applies to Form														Ref. to Circled Nos. in Forms	Instructions for Completion
N-1	N-1A	N-2	N-3	N-5	N-6	NPP-1	NPV-1	NV-1	NCS-1	NS-1	NM-1	C-1			
X	X	X	(27)		Location of heads (top, bottom, ends, floating, or channel) and description of head geometry in applicable space.
X	X	X	(28)		Diameter and number of bolts.
X	X	(29)		Other fastenings such as quick opening; describe fully or attach sketch.
X	(30)		Describe type of jacket closure geometry, including dimensions or attach sketch (e.g., ogee and weld, bar, etc.).
X	X	X	...	X	X	X	(31)		Design Pressure specified in Design Specification.
X	X	X	...	X	X	X	(32)		Design Temperature specified in Design Specification.
X	X	X	...	X	X	X	...	X	X	...	(33)		Minimum pressure-test temperature as specified in Design Specification.
X	X	X	...	X	X	X	...	X	X	X	(34)		Circle type of test used and specify test pressure (pneumatic, hydrostatic, or combination test, as applicable).
X	(35)		Nominal diameter subject to pressure (refer to design documents).
X	(36)		Nominal thickness of tubesheet.
X	(37)		Method of tubesheet attachments (describe whether bolted, welded or other; attach sketch as necessary).
X	X	(38)		Specify nominal outside diameter.
...	X	(39)		Describe type of core support structure component (e.g., bottom grid, fuel support, top guide, etc.).
X	(40)		Nominal outside diameter of tubes.
X	(41)		Nominal wall thickness of tubes or gage size.
X	X	(42)		Number of tubes.
X	(43)		Actual tube configuration (straight or U-tube, etc.).
X	X	X	(44)		Nozzles, inspection, and safety valve openings; list all openings, regardless of size, penetrating pressure boundary.
X	X	X	(45)		Describe: (a) type of support (skirt, lugs, legs, etc.); (b) location of support (top, bottom, side, etc.); (c) method of attachment (bolted, welded, etc.).
X	X	X	X	X	X	X	X	X	X	X	X	X	(46)		Describe any additional Code requirements, restrictions, or additional information, including marking in lieu of stamping, not otherwise covered in Data Report. Include any required tests or examination not performed when Data Report is completed.
...	X	(47)		Specify the name and address of the organization where design information is on file.
...	X	X	(48)		Specify the name and address of the organization where Design Report or Design Specification is on file.
X	X	X	...	X	X	...	X	X	X	X	...	X	(49)		Enter the name of engineer who certified the Design Specification. Show state of registration and number. List name of individual only, signature not required. (Applies to Form N-2 only when Design Specification is required.)

Table V-1000
Guide for Preparation of Data Report Forms (Cont'd)

Applies to Form													Ref. to Circled Nos. in Forms	Instructions for Completion
N-1	N-1A	N-2	N-3	N-5	N-6	NPP-1	NPV-1	NV-1	NCS-1	NS-1	NM-1	C-1		
X	X	X	...	X	X	X	X	X	...	X	(50)	Enter name of engineer who certified the Design Report. Show state of registration and number. List name of individual only, signature not required in space provided. (Applies to Form N-2 only when Design Report is required.)
...	X	X	...	X	(51)	System name is identified in the Design Specification (main steam, feedwater, safety injection, etc.).
...	X	(52)	Name of power plant designated by utility.
...	X	(53)	Number designation of unit.
...	X	(54)	Specify whether pump, valve, or safety relief valve.
...	X	(55)	Describe the piping, flanges, and fittings assembled and covered by Data Report including material specification [see Note (15)] or as an alternative, reference applicable sketch or drawing and attach to the Data Report.
X	X	X	X	X	X	X	X	X	X	X	X	X	(56)	When supplemental sheets are attached to the Data Report, each page is sequentially numbered and the total number of sheets is identified on the top right hand corner of the Data Report in the space provided.
...	(57)	Actual operating pressure for the system. (This may differ from the Design Pressure in the design certification block on back of Form N-5 .)
...	(58)	Actual operating temperature for the system. (This may differ from the Design Pressure in the design certification block on back of Form N-5 .)
...	X	X	(59)	Name of Material Manufacturer.
...	X	(60)	Name of person approving drawing or procedure.
...	X	(61)	List report or data sheet number as applicable.
...	X	(62)	Indicate valve, vessel, pump, or appurtenance.
...	X	X	(63)	Indicate piping subassembly or part.
...	X	X	(64)	List support identification number, model, or catalog item.
...	X	(65)	Indicate fluid used for National Board capacity test; the certified relieving capacity; and the percent overpressure used during the capacity test or pressure differential for vacuum relief valves.
...	X	(66)	Date of National Board capacity test certification.
X	X	X	X	X	X	X	X	X	X	...	X	X	(67)	The Inspector's National Board Commission No. and Endorsement must be shown.
X	X	X	X	X	X	X	X	X	X	...	X	...	(68)	This certificate is to be completed by the Certificate Holder and signed by the Authorized Nuclear Inspector who performs the inspection.
X	X	X	X	X	X	X	X	X	X	X	X	X	(69)	Certificate of Compliance block is to show the name of the responsible Certificate Holder as shown on his ASME Code Certificate of Authorization. This should be signed in accordance with the organizational authority defined in the Quality Assurance Program.

Table V-1000
Guide for Preparation of Data Report Forms (Cont'd)

Applies to Form													Ref. to Circled Nos. in Forms	Instructions for Completion
N-1	N-1A	N-2	N-3	N-5	N-6	NPP-1	NPV-1	NV-1	NCS-1	NS-1	NM-1	C-1		
X	X	X	X	X	(70)	Certificate of Compliance block for field installation work or assembly is to be signed by the Certificate Holder's representative in charge of field fabrication. This should be signed in accordance with organizational authority defined in the Quality Assurance Program.
X	X	X	X	X	(71)	This certificate block is for the Authorized Inspector to sign for any field construction or assembly work. See Note (67) for National Board Commission Number requirements.
...	X	(72)	Specific gravity (density of fluid in relation to water, water being 1.0).
...	X	(73)	Maximum height of fluid in tank.
...	X	(74)	Temperature of test.
...	X	(75)	Impact test lateral expansion.
...	X	(76)	Minimum temperature [only when below -20°F (-29°C)] for design.
...	...	X	X	(77)	Identification number and title of Division 2 Construction Specification, including applicable revision number and the date of the revision (applies to Division 2 items only).
...	X	X	(78)	Design Specification identifying number, including applicable revision number and the date of the revision.
...	X	...	(79)	Heat or lot identification number of material used for fabrication.
...	X	(80)	List items included in the piping subassembly for which Certificate Holder's Data Reports have been completed, including serial number or National Board number and brief identifying description.
...	X	(81)	Valve pressure class designation per ASME B16.34.
...	X	(82)	Diameter of orifice opening.
...	X	X	X	(83)	Enter the nominal pipe size of inlet or outlet opening.
...	X	(84)	Indicate method of valve operation (spring, pilot operated, or power operated).
...	X	(85)	Set opening pressure of valve.
...	X	(86)	Difference between opening and reseating pressure.
...	X	(87)	Temperature rating of valve at the rated relieving capacity.
...	X	(88)	Concrete construction type (reinforced or prestressed).
...	X	(89)	Height of dome above springline.
...	X	(90)	Geometry of dome (spherical, ellipsoidal, conical, flat, etc.).
...	X	...	X	(91)	Foundation type and underlying substrata.
...	(92)	Date Design Report certified.
...	X	(93)	Construction report identification number and certification date.
...	X	(94)	Describe corrosion protection and coatings for post-tensioning tendons.
...	(95)	List all penetrations (openings), regardless of size, passing through the pressure boundary. State type (name), size, shape (circular, rectangular, etc.), and serial number.
...	X	...	X	(96)	Type of support (plate and shell, linear, standard support).

Table V-1000
Guide for Preparation of Data Report Forms (Cont'd)

Applies to Form													Ref. to Circled Nos. in Forms	Instructions for Completion
N-1	N-1A	N-2	N-3	N-5	N-6	NPP-1	NPV-1	NV-1	NCS-1	NS-1	NM-1	C-1		
...	X	(97)	Design Report or load capacity data sheet (indicate which).
...	X	(98)	Disk differential pressure. A pressure equal to 110% of the valve pressure rating at 100°F (38°C).
...	X	(99)	Description of part (support) (snubber, sway brace, clevis, U-bolt, threaded rod with fastener, etc.).
...	X	(100)	Pump or valve (indicate which).
...	(101)	Brief description of service (feedwater, reactor cooling, safety injection, component cooling, etc.).
...	X	(102)	Mark number (the unique identification assigned by the Material Manufacturer to provide traceability to the CMTR).
...	X	(103)	Cold working pressure: the pressure at 100°F (38°C) as established by the pressure-temperature tables in ASME B16.34 (valves only).
...	X	(104)	Model number, series number, type (either a number traceable to the type or a description of the type for example gate, globe, butterfly, etc.).
...	X	(105)	Pressure equal to or greater than the Design Pressure specified in the Design Specification.
...	X	(106)	Coincident temperature per ASME B16.34 to pressure listed in Note (105). This temperature shall be equal to or greater than the Design Temperature specified in the Design Specification.
...	X	(107)	Minimum allowable yield strength specified in the appropriate Material Specification.
...	X	(108)	Indicate final ring position(s) with respect to an indicated reference point.
...	X	(109)	Name of certifying Professional Engineer. Show state of registration and registration number.

GENERAL NOTES:

- All blanks on the Data Report must contain an entry. If an entry is not applicable, enter "N/A" into the blank. Any quantity to which units apply shall be entered on the Manufacturer's Data Report with the chosen units.
- If space on Data Report is not sufficient for required information, either the remarks section is used or a supplementary sheet shall be attached and information listed by line number.
- These instructions constitute a nonmandatory guide for completion of Data Reports for items constructed to Section III Editions and Addenda prior to the Winter 1984 Addenda.
- The NS-1 Certificate is a Certificate of Conformance and is used in lieu of a Code Data Report form for welded supports.

MANDATORY APPENDIX VI

ARTICLE VI-1000 ROUNDED INDICATIONS

VI-1100 ACCEPTANCE STANDARDS FOR RADIOGRAPHICALLY DETERMINED ROUNDED INDICATIONS IN WELDS

VI-1110 APPLICABILITY OF THESE STANDARDS

These standards are applicable to ferritic, austenitic, and nonferrous material.

VI-1120 TERMINOLOGY

VI-1121 Rounded Indications

Indications with a maximum length of three times the width or less on the radiograph are defined as rounded indications. These indications may be circular, elliptical, conical, or irregular in shape and may have tails. When evaluating the size of an indication, the tail shall be included. The indication may be from any source in the weld, such as porosity, slag, or tungsten.

VI-1122 Aligned Indications

A sequence of four or more rounded indications shall be considered to be aligned when they touch a line parallel to the length of the weld drawn through the center of the two outer rounded indications.

VI-1123 Thickness t

t is the thickness of the weld, of the pressure retaining material, or of the thinner of the sections being joined, whichever is least. If a full penetration weld includes a fillet weld, the thickness of the fillet weld throat shall be included in t .

VI-1130 ACCEPTANCE CRITERIA

VI-1131 Image Density

Density within the image of the indication may vary and is not a criterion for acceptance or rejection.

VI-1132 Relevant Indications (see Table VI-1132-1 for Examples)

Only those rounded indications which exceed the following dimensions shall be considered relevant:

(a) $\frac{1}{10}t$ for t less than $\frac{1}{8}$ in. (3 mm);

(b) $\frac{1}{64}$ in. (0.4 mm) for t equal to $\frac{1}{8}$ in. to $\frac{1}{4}$ in. (3 mm to 6 mm), inclusive;

(c) $\frac{1}{32}$ in. (0.8 mm) for t greater than $\frac{1}{4}$ in. to 2 in. (6 mm to 50 mm), inclusive;

(d) $\frac{1}{16}$ in. (1.5 mm) for t greater than 2 in. (50 mm).

VI-1133 Maximum Size of Rounded Indication (See Table VI-1132-1 for Examples)

The maximum permissible size of any indication shall be $\frac{1}{4}t$ or $\frac{5}{32}$ in. (4 mm), whichever is less, except that an isolated indication separated from an adjacent indication by 1 in. (25 mm) or more may be $\frac{1}{3}t$ or $\frac{1}{4}$ in. (6 mm), whichever is less. For t greater than 2 in. (50 mm), the maximum permissible size of an isolated indication shall be increased to $\frac{3}{8}$ in. (10 mm).

VI-1134 Aligned Rounded Indications

Aligned rounded indications are acceptable when the summation of the diameters of the indications is less than t in a length of $12t$ (see Figure VI-1134-1). The length of groups of aligned rounded indications and the spacing between the groups shall meet the requirements of Figure VI-1134-2.

VI-1135 Spacing

The distance between adjacent rounded indications is not a factor in determining acceptance or rejection, except as required for isolated indications or groups of aligned indications.

VI-1136 Rounded Indication Charts

(a) The rounded indications as determined from the radiographic film shall not exceed that shown in the charts.

(b) The charts in Figures VI-1136-1 through VI-1136-6 illustrate various types of assorted, randomly dispersed, and clustered rounded indications for different weld thicknesses greater than $\frac{1}{8}$ in. (3 mm). These charts represent the maximum acceptable concentration limits for rounded indications.

(c) The chart for each thickness range represents full-scale 6 in. (150 mm) radiographs and shall not be enlarged or reduced. The distributions shown are not necessarily

Table VI-1132-1
Maximum Size of Nonrelevant Indications and Acceptable Rounded Indications — Examples Only

Thickness t , in. (mm)	Maximum Size of Acceptable Rounded Indication, in. (mm)		Maximum Size of Nonrelevant Indication, in. (mm)
	Random	Isolated	
$< \frac{1}{8}$ (<3)	$\frac{1}{4}t$	$\frac{1}{3}t$	$\frac{1}{10}t$
$\frac{1}{8}$ (3)	0.031 (0.8)	0.042 (1.1)	0.015 (0.4)
$\frac{3}{16}$ (5)	0.047 (1.2)	0.063 (1.6)	0.015 (0.4)
$\frac{1}{4}$ (6)	0.063 (1.6)	0.083 (2.1)	0.015 (0.4)
$\frac{5}{16}$ (8)	0.078 (2.0)	0.104 (2.6)	0.031 (0.8)
$\frac{3}{8}$ (10)	0.091 (2.3)	0.125 (3.2)	0.031 (0.8)
$\frac{7}{16}$ (11)	0.109 (2.8)	0.146 (3.7)	0.031 (0.8)
$\frac{1}{2}$ (13)	0.125 (3.2)	0.168 (4.3)	0.031 (0.8)
$\frac{9}{16}$ (14)	0.142 (3.6)	0.188 (4.8)	0.031 (0.8)
$\frac{5}{8}$ (16)	0.156 (4.0)	0.210 (5.3)	0.031 (0.8)
$\frac{11}{16}$ (17)	0.156 (4.0)	0.230 (5.8)	0.031 (0.8)
$\frac{3}{4}$ -2, incl. (19-50, incl.)	0.156 (4.0)	0.250 (6.4)	0.031 (0.8)
>2 (>50)	0.156 (4.0)	0.375 (9.5)	0.063 (1.6)

the patterns that may appear on the radiograph, but are typical of the concentration and size of indications permitted.

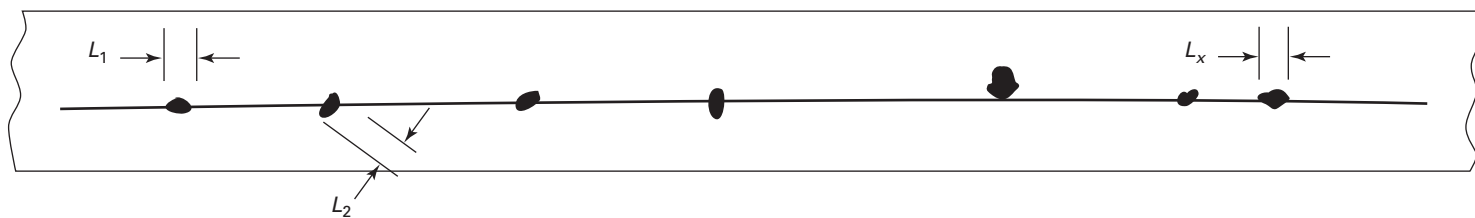
VI-1137 Weld Thickness t Less Than $\frac{1}{8}$ in. (3 mm)

For t less than $\frac{1}{8}$ in. (3 mm), the maximum number of rounded indications shall not exceed 12 in a 6 in. (150 mm) length of weld. A proportionally fewer number of indications shall be permitted in welds less than 6 in. (150 mm) in length.

VI-1138 Clustered Indications

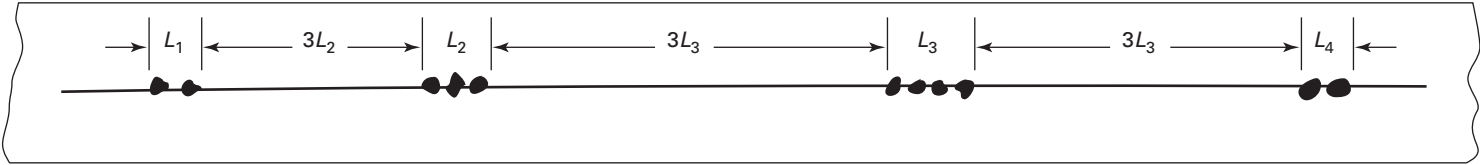
The illustrations for clustered indications show up to four times as many indications in a local area, as that shown in the illustrations for random indications. The length of an acceptable cluster shall not exceed the lesser of 1 in. (25 mm) or $2t$. Where more than one cluster is present, the sum of the lengths of the clusters shall not exceed 1 in. (25 mm) in a 6 in. (150 mm) length of weld.

Figure VI-1134-1
Aligned Rounded Indications



GENERAL NOTE: Sum of L_1 to L_x shall be less than t in a length of $12 t$.

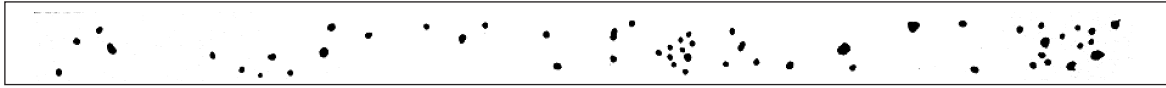
Figure VI-1134-2
Groups of Aligned Rounded Indications



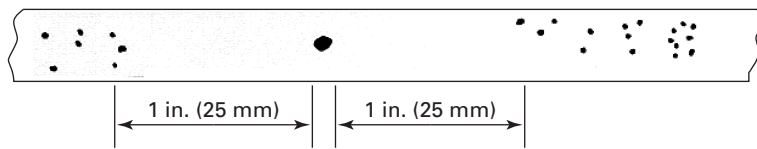
Maximum Group Length	Minimum Group Spacing
$L = \frac{1}{4}$ in. (6 mm) for t less than $\frac{3}{4}$ in. (19 mm)	$3L$ where L is the length of the longest adjacent group being evaluated.
$L = \frac{1}{3} t$ for t equal to $\frac{3}{4}$ in. to $2\frac{1}{4}$ in. (19 mm to 57 mm)	
$L = \frac{3}{4}$ in. (19 mm) for t greater than $2\frac{1}{4}$ in. (57 mm)	

GENERAL NOTE: The sum of the group lengths shall be less than t in a length of $12 t$.

Figure VI-1136-1
Charts for t Equal to $\frac{1}{8}$ in. to $\frac{1}{4}$ in. (3 mm to 6 mm), Inclusive



(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]

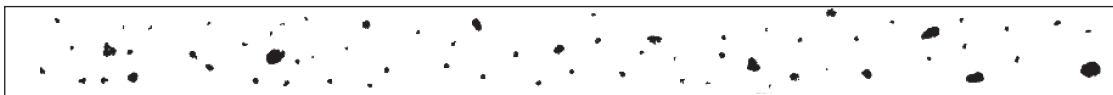


(b) Isolated Indication (Maximum size per Table VI-1132-1.)

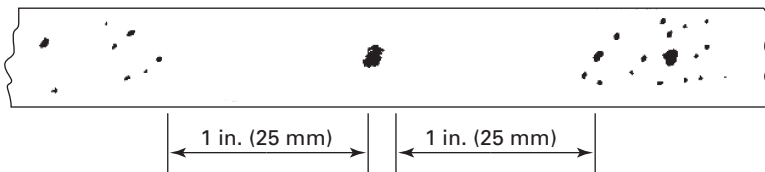


(c) Cluster

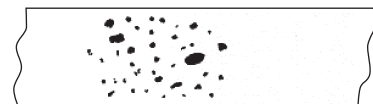
Figure VI-1136-2
Charts for t Over $\frac{1}{4}$ in. to $\frac{3}{8}$ in. (6 mm to 10 mm), Inclusive



(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]



(b) Isolated Indication (Maximum size per Table VI-1132-1.)

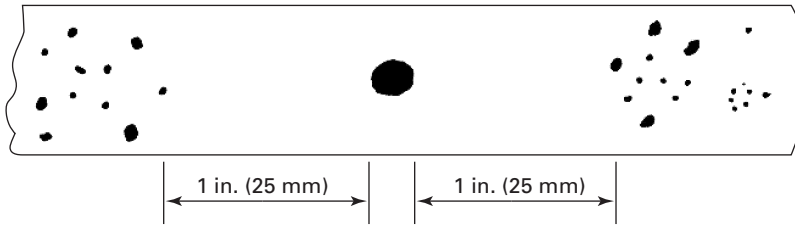


(c) Cluster

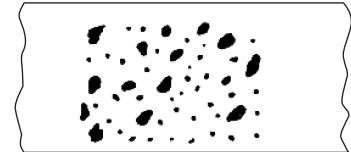
Figure VI-1136-3
Charts for t Over $\frac{3}{8}$ in. to $\frac{3}{4}$ in. (10 mm to 19 mm), Inclusive



(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]



(b) Isolated Indication (Maximum size per Table VI-1132-1.)

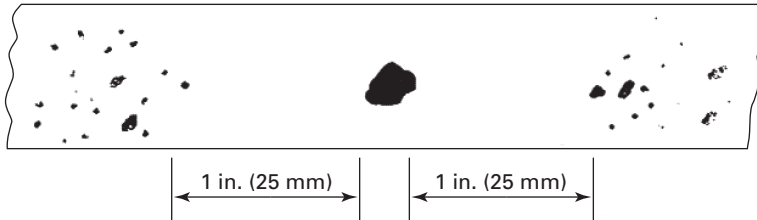


(c) Cluster

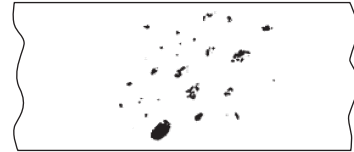
Figure VI-1136-4
Charts for t Over $\frac{3}{4}$ in. to 2 in. (19 mm to 50 mm), Inclusive



(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]

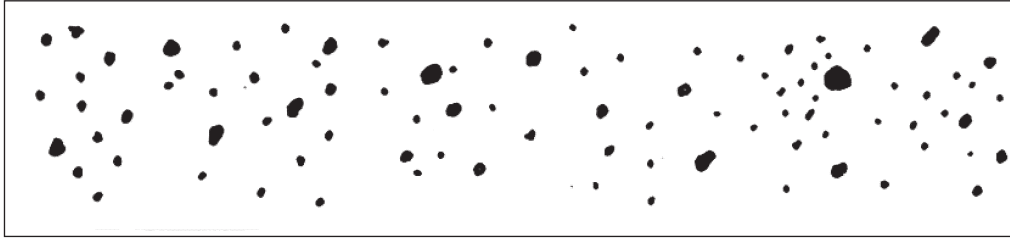


(b) Isolated Indication (Maximum size per Table VI-1132-1.)

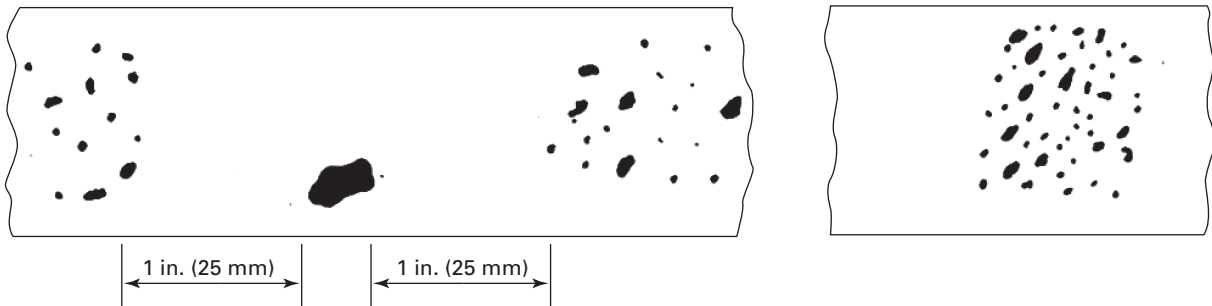


(c) Cluster

Figure VI-1136-5
Charts for t Over 2 in. to 4 in. (50 mm to 100 mm), Inclusive



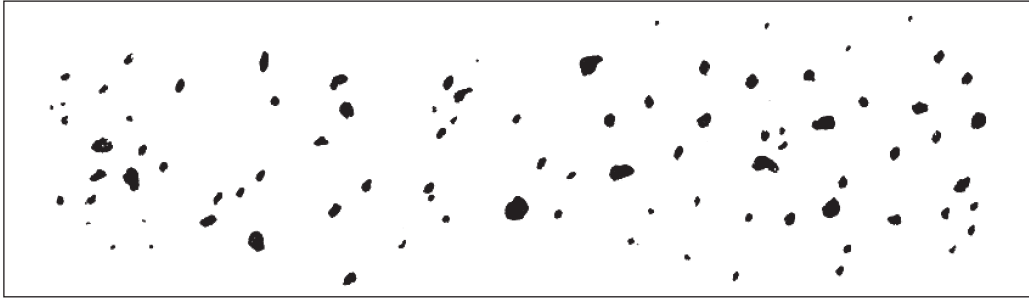
(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]



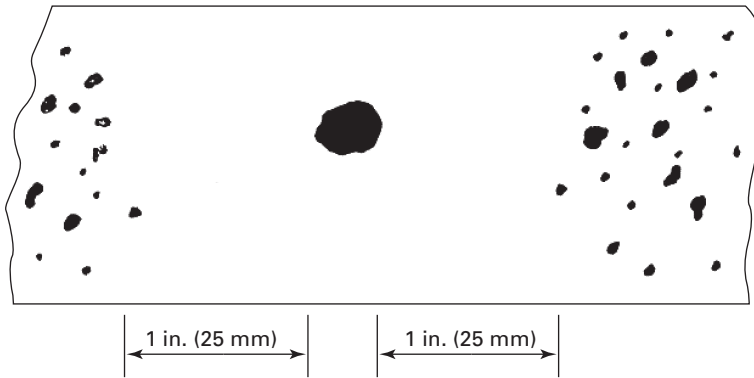
(b) Isolated Indication (Maximum size per Table VI-1132-1.)

(c) Cluster

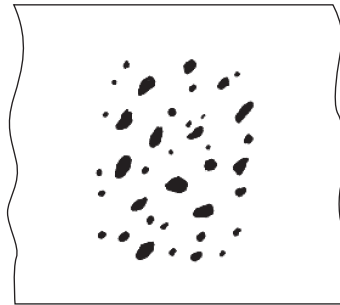
Figure VI-1136-6
Charts for t Over 4 in. (100 mm)



(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]



(b) Isolated Indication (Maximum size per Table VI-1132-1.)



(c) Cluster

MANDATORY APPENDIX XI

RULES FOR BOLTED FLANGE CONNECTIONS FOR CLASS 2 AND 3 COMPONENTS AND CLASS MC VESSELS

ARTICLE XI-1000 INTRODUCTION

XI-1100 GENERAL REQUIREMENTS

XI-1110 SCOPE

(a) The rules in [Mandatory Appendix XI](#) apply specifically to the design of bolted flange connections for Class 2 and 3 components and Class MC vessels, and are to be used in conjunction with the applicable requirements in Subsections NC, ND, and NE. These rules provide only for hydrostatic end loads and gasket seating loads. For a discussion of design considerations for bolted flange connections, see [Mandatory Appendix XII](#).

(b) Only circular flanges designated Class RF are covered by the rules of [Mandatory Appendix XI](#). The design procedures for Class RF flanges, as defined in [XI-3211](#), are given in [XI-3200](#). For Class FF flanges, see [Nonmandatory Appendix L](#).

(c) The flange design methods stipulated in this Appendix are primarily applicable to circular flanges under internal pressure. However, Class RF flanges may be used under external pressure when designed in accordance with [XI-3260](#).

XI-1120 ELEMENTS INVOLVED IN FLANGE DESIGN

XI-1121 General Considerations

This method of designing a flange involves the selection of the gasket material, type and dimensions, flange facing, bolting, hub proportions, flange width, and flange thickness.² Flange dimensions shall be such that the stresses in the flange, calculated in accordance with [XI-3240](#), do not exceed the allowable flange stresses specified in [XI-3250](#) for Class RF flanges. All calculations shall use dimensions in the corroded condition.

XI-1122 Conditions for Which Design Calculations Shall Be Made

In the design of a bolted flange connection, complete calculations shall be made for two separate and independent sets of conditions which are defined in the following subparagraphs.

XI-1122.1 Design Conditions. The Design Conditions are those required to resist the hydrostatic end force of the Design Pressure tending to part the joint and to maintain on the gasket or joint contact surface sufficient compression to assure a tight joint, all at the Design Temperature. The minimum load is a function of the Design Pressure, the gasket material, and the effective gasket or contact area to be kept tight under pressure [as calculated by [eq. XI-3221.1\(1\)](#) and determines one of the two requirements for the amount of bolting A_{m1} . This load is also used for the design of the flange in [eq. XI-3223\(3\)](#).

XI-1122.2 Gasket Seating Conditions. The gasket seating conditions are those existing when the gasket or joint contact surface is seated by applying initial load with the bolts when assembling the joint, at atmospheric temperature and pressure. The minimum initial load considered to be adequate for proper seating is a function of the gasket material and the effective gasket or contact area to be seated [as calculated by [eq. XI-3221.2\(2\)](#) and determines the other of the two requirements for the amount of bolting A_{m2} . For the design of the flange, this load is modified in [eq. XI-3223\(4\)](#) to take account of the Design Conditions when they govern the amount of bolting required A_m , as well as the amount of bolting actually provided A_b .

XI-1123 Bolted Flange Connections to External Piping

It is recommended that bolted flange connections conforming to the standards³ listed in NC-3362, ND-3362, and NE-3362, as applicable, be used for connections to external piping. These standards may be used for other bolted flange connections within the limits of size in the standards and the pressure–temperature ratings permitted in Subsection NC, ND, or NE.

ARTICLE XI-2000

MATERIALS FOR BOLTED FLANGE CONNECTIONS

XI-2100 MATERIAL REQUIREMENTS

XI-2110 GENERAL REQUIREMENTS

Materials used in the construction of bolted flange connections shall comply with the material requirements given in Subsection NC, ND, or NE, as applicable.

XI-2120 HEAT TREATMENT OF FLANGES

Flanges made from ferritic steel and designed in accordance with this Appendix shall be given a normalizing or full annealing heat treatment when the thickness of the flange section exceeds 3 in. (75 mm).

XI-2130 WELDABILITY OF FLANGES AND POSTWELD HEAT TREATMENT

Material on which welding is to be performed shall be proved of good weldable quality. Satisfactory qualification of the welding procedure under Section IX is considered as proof. Welding shall not be performed on steel that has a carbon content greater than 0.35%. All welding on flange connections shall comply with the requirements for postweld heat treatment given in Subsection NC, ND, or NE, as applicable.

XI-2140 FABRICATED HUBBED FLANGES

Fabricated hubbed flanges shall be in accordance with (a) through (c).

(a) Hubbed flanges may be fabricated from a hot-rolled or hot-forged billet. The axis of the finished flange shall be parallel to the long axis of the original billet. (This is not intended to imply that the axis of the finished flange and the original billet must be concentric.)

(b) Hubbed flanges, except as permitted in (a), shall not be machined from plate or bar stock material unless the material has been formed into a ring and, further, provided that

(1) in a ring formed from plate, the original plate surfaces are parallel to the axis of the finished flange (this is not intended to imply that the original plate surface be present in the finished flange).

(2) the joints in the ring are welded butt joints that conform to the requirements of the applicable Subsection. The thickness to be used to determine postweld heat treatment and radiography requirements shall be the lesser of t or $(A - B)/2$, when these symbols are as defined in XI-3130.

(c) The back of the flange and the outer surface of the hub shall be examined by the magnetic particle method (NC-2545) or the liquid penetrant method (NC-2546) to ensure that these surfaces are free from defects.

XI-2150 BOLTING MATERIALS

Bolts, studs, nuts, and washers shall comply with the requirements of the applicable Subsection. It is recommended that bolts and studs not be smaller than $\frac{1}{2}$ in. (13 mm). If bolts or studs smaller than $\frac{1}{2}$ in. (13 mm) are used, ferrous bolting material shall be of alloy steel. Precautions shall be taken to avoid overstressing small diameter bolts.

ARTICLE XI-3000 DESIGN REQUIREMENTS

XI-3100 GENERAL REQUIREMENTS

XI-3110 SCOPE

(a) The rules of XI-3200 apply to Class RF flanges as defined in XI-3212.

(b) The flange design methods given in XI-3210 through XI-3250 apply to Class RF flanges under internal pressure. The flange design methods for Class RF flanges under external pressure or under both internal and external pressure are given in XI-3260.

XI-3120 TYPES OF FLANGES

For purposes of computation, there are three types as described in (a), (b), and (c).

(a) *Loose Type Flanges*. This type covers those designs in which the flange has no direct connection to the nozzle neck, vessel, or pipe wall and designs where the method of attachment is not considered to give the mechanical strength equivalent to integral attachment. Figure XI-3120-1 sketches (1), (1a), (2), (3), and (4) show typical loose type flanges and the location of the loads and moments; welds and other details of construction shall satisfy the dimensional requirements given in the referenced sketches.

(b) *Integral Type Flanges*. This type covers designs where the flange is cast or forged integrally with the nozzle neck, vessel, or pipe wall, butt welded thereto, or attached by other forms of arc or gas welding of such a nature that the flange and nozzle neck, vessel, or pipe wall is considered to be the equivalent of an integral structure. In welded construction, the nozzle neck, vessel, or pipe wall is considered to act as a hub. Figure XI-3120-1 sketches (5), (6), (6a), (6b), and (7) show typical integral type flanges and the location of the loads and moments; welds and other details of construction shall satisfy the dimensional requirements given in the referenced sketches.

(c) *Optional Type Flanges*. This type covers designs where the attachment of the flange to the nozzle neck, vessel, or pipe wall is such that the assembly is considered to act as a unit, which shall be calculated as an integral flange, except that for simplicity the designer may calculate the construction as a loose type flange provided none of the following values are exceeded:

$$g_0 = \frac{5}{8} \text{ in. (16 mm)}$$

$$B/g_0 = 300$$

$$P = 300 \text{ psi (2 MPa)}$$

$$\text{Design Temperature} = 700^\circ\text{F (370}^\circ\text{C)}$$

Figure XI-3120-1 sketches (8), (8a), (8b), and (9) show typical optional type flanges. Welds and other details of construction shall satisfy the dimensional requirements given in those sketches.

XI-3130 NOMENCLATURE

The nomenclature defined below and shown in Figure XI-3120-1 is used in the equations for the design of flanges.

A = outside diameter of flange or, when slotted holes extend to the outside of the flange, the diameter to the bottom of the slots

A_b = actual total cross-sectional area of bolts at root of thread or section of least diameter under stress

A_m = total required cross-sectional area of bolts taken as the greater of A_{m1} and A_{m2}

A_{m1} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the Design Conditions

$$= W_{m1}/S_b$$

A_{m2} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating

$$= W_{m2}/S_a$$

B = inside diameter of flange (when B is less than $20g_1$ it will be optional for the designer to substitute B_1 for B in the equation for longitudinal stress S_H)

B_1 = $B + g_1$, for loose type hub flanges and also for integral type flanges when $f < 1$

$$= B + g_0, \text{ for integral type flanges when } f \geq 1$$

b = effective gasket or joint contact surface seating width²

b_0 = basic gasket seating width (Table XI-3221.1-2)

C = bolt circle diameter

C_b = effective width factor

$$= 0.5 \text{ for U.S. Customary calculations}$$

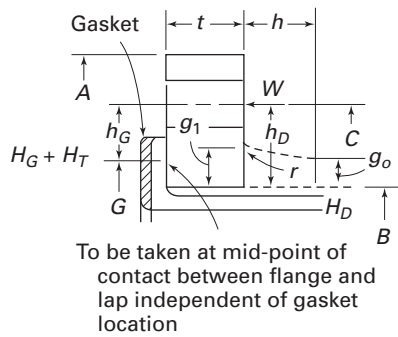
$$= 2.52 \text{ for SI calculations}$$

c = basic dimension used for the minimum sizing of welds; equal to t_n or t_D , whichever is less

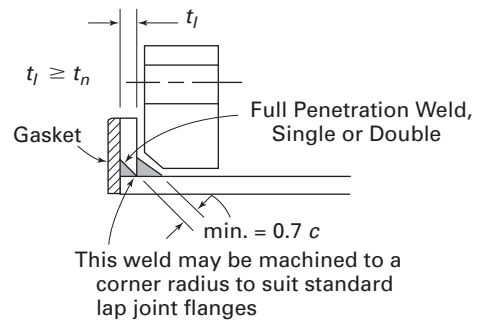
d = factor, as follows:

$$= \frac{U}{V} h_0 g_0^2 \text{ for integral type flanges}$$

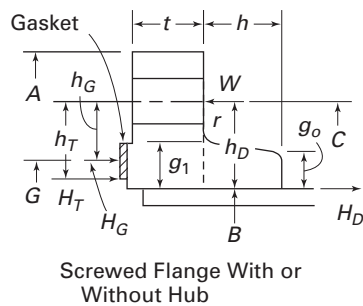
Figure XI-3120-1
Types of Flanges



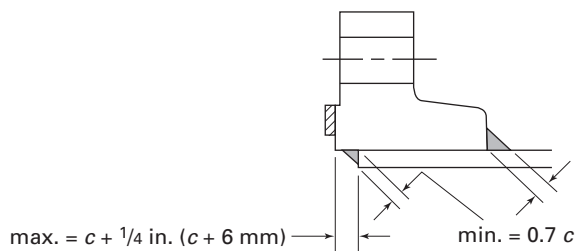
(1)



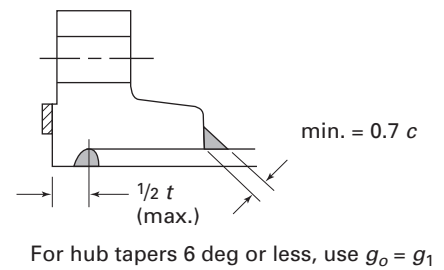
(1a)



(2)



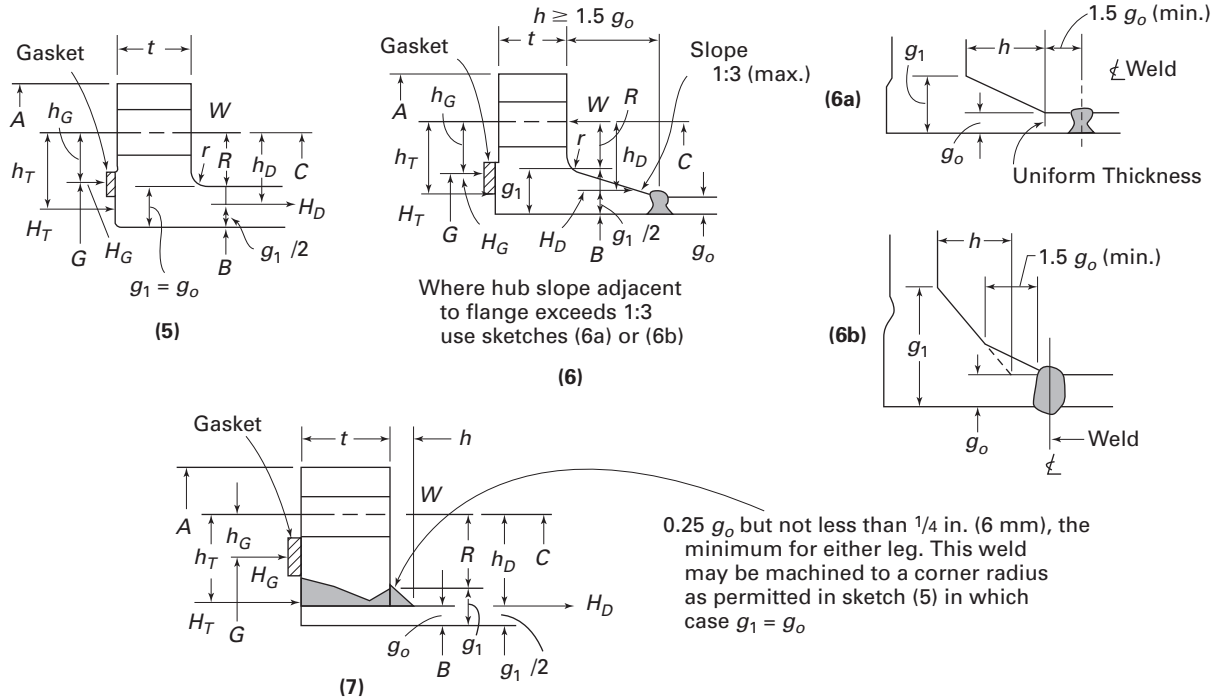
(3) [Note (1)]



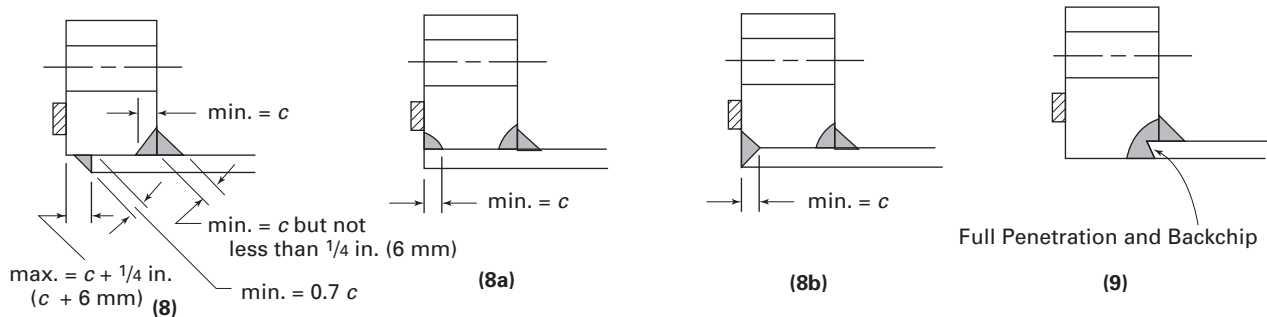
(4) [Note (1)]

Loose-Type Flanges

Figure XI-3120-1
Types of Flanges (Cont'd)



Integral-Type Flanges [Note (2)]



Optional-Type Flanges [Notes (3) and (4)]

NOTES:

- (1) Loadings and dimensions not shown are the same as for sketch (2).
- (2) Fillet radius r to be at least $0.25 g_o$ but not less than $3/16$ in. (5 mm). Added thickness greater than $1/16$ in. (1.5 mm) for raised face, tongue and groove, "O" rings, and ring joint facings shall be in excess of the required minimum flange thickness t ; those less than or equal to $1/16$ in. (1.5 mm) may be included in the required minimum flange thickness.
- (3) These may be calculated as either loose or integral type [(c)].
- (4) Loading and dimensions not shown are the same as for sketch (2) for loose type flanges or (7) for integral type.

$$= \frac{U}{V_L} h_0 g_0^2 \text{ for loose type flanges}$$

e = factor, as follows:

$$= F/h_0 \text{ for integral type flanges}$$

$$= F_L/h_0 \text{ for loose type flanges}$$

F = factor for integral type flanges (Figure XI-3240-2)

F_L = factor for loose type flanges (Figure XI-3240-4)

f = hub stress correction factor for integral flanges from Figure XI-3240-6 (when greater than 1, this is the ratio of the stress in the small end of hub to the stress in the large end; for values below limit of Figure use $f = 1$)

G = diameter at location of gasket load reaction; except as noted in sketch (1) of Figure XI-3120-1, G is defined as follows for Class RF flanges (see Table XI-3221.1-2):

(a) when $b_0 \leq \frac{1}{4}$ in. (6 mm), G is the mean diameter of gasket contact face, in. (mm);

(b) when $b_0 > \frac{1}{4}$ in. (6 mm), G is the outside diameter of gasket contact face less $2b$

g_0 = thickness of hub at small end

g_1 = thickness of hub at back of flange

H = total hydrostatic end force
 $= 0.785G^2P$

H_D = hydrostatic end force on area inside of flange
 $= 0.785B^2P$

H_G = gasket load (difference between flange design bolt load and total hydrostatic end force)
 $= W - H$

H_p = total joint contact surface compression load
 $= 2b \times 3.14GmP$

H_T = difference between total hydrostatic end force and the hydrostatic end force on area inside of flange
 $= H - H_D$

h = hub length

h_D = radial distance from the bolt circle to the circle on which H_D acts, as prescribed in Table XI-3230-1

h_G = radial distance from gasket load reaction to the bolt circle
 $= (C - G)/2$

h_0 = factor equal to $\sqrt{Bg_0}$

h_T = radial distance from the bolt circle to the circle on which H_T acts, as prescribed in Table XI-3230-1

K = ratio of outside diameter of flange to inside diameter of flange
 $= A/B$

L = factor

$$= \frac{te + 1}{T} + \frac{t^3}{d}$$

M_D = component of moment due to H_D
 $= H_D h_D$

M_G = component of moment due to H_G
 $= H_G h_G$

M_O = total moment acting upon the flange, for the Design Conditions or gasket seating, as may apply (XI-3230)

M_T = component of moment due to H_T
 $= H_T h_T$

m = gasket factor obtained from Table XI-3221.1-1²

N = width used to determine the basic gasket seating width b_0 , based upon the possible contact width of the gasket (Table XI-3221.1-2)

P = Design Pressure (for flanges subject to external pressure see XI-3260 for Class RF flanges)

R = radial distance from bolt circle to point of intersection of hub and back of flange (integral and hub flanges)
 $= (C - B)/2 - g_1$

S_a = allowable bolt stress at atmospheric temperature (given in Section II, Part D, Subpart 1, Table 3)

S_b = allowable bolt stress at Design Temperature (given in Section II, Part D, Subpart 1, Table 3)

S_f = allowable design stress for material of flange at Design Temperature (Design Condition) or atmospheric temperature (gasket seating), as applicable (given in Section II, Part D, Subpart 1, Tables 1A and 1B, as applicable)

S_n = allowable design stress for material of nozzle neck, vessel, or pipe wall at Design Temperature (Design Condition) or atmospheric temperature (gasket seating) as applicable (given in Section II, Part D, Subpart 1, Tables 1A and 1B, as applicable)

S_H = calculated longitudinal stress in hub

S_R = calculated radial stress in flange

S_T = calculated tangential stress in flange

T = factor involving K (Figure XI-3240-1)

t = flange thickness

t_D = two times the thickness g_0 , when the design is calculated as an integral flange, or two times the thickness of shell or nozzle wall required for internal pressure when the design is calculated as loose flange, but not less than $\frac{1}{4}$ in. (6 mm)

t_n = nominal thickness of shell or nozzle wall to which flange or lap is attached, less corrosion allowance

U = factor involving K (Figure XI-3240-1)

V = factor for integral type flanges (Figure XI-3240-3)

V_L = factor for loose type flanges (Figure XI-3240-5)

W = flange design bolt load for the Design Conditions or gasket seating as applicable (XI-3223)

W_{m1} = minimum required bolt load for the Design Conditions (XI-3220)

W_{m2} = minimum required bolt load for gasket seating (XI-3220)

w = width used to determine the basic gasket seating width b_0 , based upon the contact width between the flange facing and the gasket (Table XI-3221.1-2)

Y = factor involving K (Figure XI-3240-1)

y = gasket or joint contact surface unit seating load²

Z = factor involving K (Figure XI-3240-1)

XI-3200 CLASS RF FLANGE DESIGN

XI-3210 GENERAL REQUIREMENTS

XI-3211 Definition of Class RF Flanges

Class RF flanges are circular flanges having gaskets which are entirely within the circle enclosed by the bolt holes and which have no contact outside this circle.

XI-3212 Acceptability

The requirements for acceptability of Class RF flange design are given in (a) and (b).

(a) The design shall be such that the general design requirements of NC-3100, ND-3100, or NE-3100, as appropriate, and the specific design requirements of this Subarticle are met.

(b) The designs shall be limited to the types of flanges defined in XI-3120.

XI-3220 BOLT LOADS AND BOLT AREAS

XI-3221 Determination of Bolt Loads

In the design of a bolted flange connection, calculations shall be made for each of the two conditions, namely, Design Loadings and gasket seating loads, and the more severe condition shall control. In the design of flange pairs used to contain a tubesheet of a heat exchanger, or any similar design where the flanges and/or gaskets may not be the same, loads shall be determined for the most severe condition of Design Loadings and/or gasket seating loads applied to each side at the same time. This most severe condition may be gasket seating on one flange with Design Loadings on the other, gasket seating on each flange at the same time, or Design Loadings on each flange at the same time.

XI-3221.1 Bolt Load for Design Conditions. The required bolt load for the Design Conditions W_{m1} shall be sufficient to resist the hydrostatic end force H exerted by the Design Pressure on the area bounded by the diameter of gasket reaction and, in addition, to maintain on the gasket or joint contact surface a compression load H_p which experience has shown to be sufficient to ensure a tight joint. This compression load is expressed as a multiple m of the internal pressure. Its value is a function of the gasket material and construction.² The required bolt load for the Design Conditions W_{m1} is determined in accordance with eq. (1):

$$\begin{aligned} W_{m1} &= H + H_p \\ &= 0.785G^2P + (2b \times 3.14GmP) \end{aligned} \quad (1)$$

XI-3221.2 Bolt Load for Gasket Seating Condition. Before a tight joint can be obtained, it is necessary to seat the gasket or joint contact surface properly by applying a minimum initial load under atmospheric temperature conditions without the presence of internal pressure, which is

a function of the gasket material and the effective gasket area to be seated. The minimum initial bolt load W_{m2} required for this purpose shall be determined in accordance with eq. (2):

$$W_{m2} = 3.14bGy \quad (2)$$

The need for providing sufficient bolt load to seat the gasket or joint contact surfaces in accordance with eq. (2) will prevail on many low pressure designs and with facings and materials that require a high seating load and where the bolt load computed by eq. XI-3221.1(1) for the Design Conditions is insufficient to seat the joint. Accordingly, it is necessary to furnish bolting and to pretighten the bolts to provide a bolt load sufficient to satisfy both of these requirements, each one being individually investigated. When eq. (2) governs, flange proportions will be a function of the bolting instead of internal pressure.

XI-3221.3 Bolt Load When Self-Energizing Gaskets Are Used. Bolt loads for flanges using gaskets of the self-energizing type differ from those shown in XI-3221.2 as stipulated in (a) and (b).

(a) The required bolt load for the Design Conditions W_{m1} shall be sufficient to resist the hydrostatic end force H exerted by the Design Pressure on the area bounded by the outside diameter of the gasket. H_p is to be considered as zero for all self-energizing gaskets except certain seal configurations which generate axial loads which shall be considered.

(b) $W_{m2} = 0$. Self-energizing gaskets may be considered to require an inconsequential amount of bolting force to produce a seal. Bolting, however, shall be pretightened to provide a bolt load sufficient to withstand the hydrostatic end force H .

XI-3222 Total Required and Actual Bolt Areas A_m and A_b

The total cross-sectional area of bolts A_m required for both the Design Conditions and gasket seating is the greater of the values for A_{m1} and A_{m2} , where $A_{m1} = W_{m1}/S_b$ and $A_{m2} = W_{m2}/S_a$. A selection of bolts to be used shall be made such that the actual total cross-sectional areas of bolts A_b will not be less than A_m .

XI-3223 Flange Design Bolt Load W

The bolt loads used in the design of the flange shall be the values obtained from eqs. (3) and (4). For Design Conditions,

$$W = W_{m1} \quad (3)$$

For gasket seating,

$$W = \frac{(A_m + A_b)S_a}{2} \quad (4)$$

Table XI-3221.1-1
Gasket Materials and Contact Facings
Gasket Factors m for Operating Conditions and Minimum Design Seating Stress y



















Gasket Material	Gasket Factor m	Min. Design Seating Stress y , <i>psi (MPa)</i>	Sketches	Facing Sketch and Column in Table XI-3221.1-2
Self-energizing types (O rings, metallic, elastomer, other gasket types considered as self-sealing)	0	0 (0)
Elastomers without fabric or high percent of asbestos fiber:				
Below 75A Shore Durometer	0.50	0 (0)		(1a),(1b),(1c),(1d), (4), (5); Column II
Below 75A Shore Durometer	1.00	200 (1.4)		
Asbestos with suitable binder for operating conditions:				
1/8 in. (3 mm) thick	2.00	1,600 (11)		(1a),(1b),(1c),(1d), (4), (5); Column II
1/16 in. (1.5 mm) thick	2.75	3,700 (26)		
1/32 in. (0.8 mm) thick	3.50	6,500 (45)		
Elastomers with cotton fabric insertion	1.25	400 (2.8)		(1a),(1b),(1c),(1d), (4), (5); Column II
Elastomers with asbestos fabric insertion (with or without wire reinforcement):				
3-ply	2.25	2,200 (15)		...
2-ply	2.50	2,900 (20)		(1a),(1b),(1c),(1d), (4), (5); Column II
1-ply	2.75	3,700 (26)		...
Vegetable fiber	1.75	1,100 (8)		(1a),(1b),(1c),(1d), (4), (5); Column II
Spiral-wound metal, asbestos filled:				
Carbon	2.50	10,000 (69)		(1a),(1b); Column II
Stainless or Monel	3.00	10,000 (69)		
Corrugated metal, asbestos inserted; or corrugated metal, jacketed asbestos filled:				
Soft aluminum	2.50	2,900 (20)		(1a),(1b); Column II
Soft copper or brass	2.75	3,700 (26)		
Iron or soft steel	3.00	4,500 (31)		
Monel or 4% to 6% chrome	3.25	5,500 (38)		
Stainless steels	3.50	6,500 (45)		
Corrugated metal:				
Soft aluminum	2.75	3,700 (26)		(1a),(1b),(1c),(1d), Column II
Soft copper or brass	3.00	4,500 (31)		
Iron or soft steel	3.25	5,500 (38)		
Monel or 4% to 6% chrome	3.50	6,500 (45)		
Stainless steels	3.75	7,600 (52)		
Flat metal, jacketed asbestos filled:				
Soft aluminum	3.25	5,500 (38)		(1a),(1b),(1c) [Note (1)], (1d) [Note (1)],
Soft copper or brass	3.50	6,500 (45)		(2)[Note (1)];
Iron or soft steel	3.75	7,600 (52)		Column II
Monel	3.50	8,000 (55)		
4-6% chrome	3.75	9,000 (62)		
Stainless steels	3.75	9,000 (62)		
Grooved metal:				
Soft aluminum	3.25	5,500 (38)		(1a),(1b),(1c),(1d), (2), (3); Column II
Soft copper or brass	3.50	6,500 (45)		
Iron or soft steel	3.75	7,600 (52)		
Monel or 4-6% chrome	3.75	9,000 (62)		
Stainless steels	4.25	10,100 (70)		

Table XI-3221.1-1
Gasket Materials and Contact Facings
Gasket Factors m for Operating Conditions and Minimum Design Seating Stress y (Cont'd)

Gasket Material	Gasket Factor m	Min. Design Seating Stress y , <i>psi (MPa)</i>	Sketches	Facing Sketch and Column in Table XI-3221.1-2
Solid flat metal:				
Soft aluminum	4.00	8,800 (61)		(1a),(1b),(1c),(1d), (2), (3),(4),(5); Column I
Soft copper or brass	4.75	13,000 (90)		
Iron or soft steel	5.50	18,000 (124)		
Monel or 4% to 6% chrome	6.00	21,800 (150)		
Stainless steels	6.50	26,000 (180)		
Ring joint:				
Iron or soft steel	5.50	18,000 (124)		(6), Column I
Monel or 4% to 6% chrome	6.00	21,800 (150)		
Stainless steels	6.50	26,000 (180)		

GENERAL NOTE: This Table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using effective gasket seating width b given in Table XI-3221.1-2. The design values and other details given in this Table are suggested only and are not mandatory.

NOTE:

(1) The surface of a gasket having a lap should not be against the nubbin.

In addition to the minimum requirements for safety, eq. (4) provides a margin against abuse of the flange from overbolting. Since margin against such abuse is needed primarily for the initial bolting up operation, which is done at atmospheric temperature and before application of internal pressure, the flange design is required to satisfy this loading only under such conditions.⁴

XI-3230 FLANGE MOMENTS

(a) In the calculation of flange stresses, the moment of a loading acting on the flange is the product of the load and its moment arm. The moment arm is determined by the relative position of the bolt circle with respect to that of the load producing the moment (Figure XI-3120-1). No consideration shall be given to any possible reduction in moment arm due to cupping of the flanges or due to inward shifting of the line of action of the bolts as a result thereof.

(b) For the Design Conditions, the total flange moment M_0 is the sum of the three individual moments M_D , M_T , and M_G , as defined in XI-3130, and based on the flange design bolt load of eq. XI-3223(3) with moment arms as given in Table XI-3230-1.

(c) For gasket seating, the total flange moment M_0 is based on the flange design bolt load of eq. XI-3223(4), which is opposed only by the gasket load, in which case:

$$M_0 = W \frac{(C - G)}{2} \quad (5)$$

XI-3240 CALCULATION OF FLANGE STRESSES

The stresses in the flange shall be determined for both the Design Conditions and gasket seating, whichever controls, in accordance with the equations in (a) or (b).

(a) For integral type flanges and all hub type flanges
Longitudinal hub stress

$$S_H = f M_0 / L g_1^2 B \quad (6)$$

Radial flange stress

$$S_R = [(1.33te + 1)M_0] / Lt^2 B \quad (7)$$

Tangential flange stress

$$S_T = (Y M_0 / t^2 B) - Z S_R \quad (8)$$

(b) For loose type ring flanges including optional type calculated as loose type having a rectangular cross section

$$\begin{aligned} S_T &= Y M_0 / t^2 B \\ S_R &= 0 \\ S_H &= 0 \end{aligned} \quad (9)$$

XI-3250 ALLOWABLE FLANGE DESIGN STRESSES

The flange stresses calculated by the equations in XI-3240 shall not exceed the values given in (a) through (f).

(a) The longitudinal hub stress S_H shall not be greater than the smaller of $1.5S_f$ or $1.5S_n$ for optional type flanges designed as integral [Figure XI-3120-1 sketches (8), (8a),

Table XI-3221.1-2
Effective Gasket Width

...		Column I	Column II
Facing Stretch (Exaggerated)		Basic Gasket Seating Width, b_o	
(1a)		$\frac{N}{2}$	$\frac{N}{2}$
(1b)			
(1c)		$\frac{w + T}{2}; \left(\frac{w + N}{4}\right)_{\max.}$	$\frac{w + T}{2}; \left(\frac{w + N}{4}\right)_{\max.}$
(1d)			
(2)		$\frac{w + N}{4}$	$\frac{w + 3N}{8}$
(3)		$\frac{N}{4}$	$\frac{3N}{8}$
(4)		$\frac{3N}{8}$	$\frac{7N}{16}$
(5)		$\frac{N}{4}$	$\frac{3N}{8}$
(6)		$\frac{W}{8}$...
<p align="center">Location of Gasket Load Reaction</p> <div style="display: flex; justify-content: space-around;"> <div style="text-align: center;"> <p>For $b_o > 1/4$ in. (6 mm)</p> </div> <div style="text-align: center;"> <p>For $b_o \leq 1/4$ in. (6 mm)</p> </div> </div>			

Table XI-3221.1-2
Effective Gasket Width (Cont'd)

GENERAL NOTES:

(a) Effective Gasket Seating Width:

$$b = b_0 \text{ when } b_0 \leq \frac{1}{4} \text{ in. (6 mm)}$$

$$b = C_b \sqrt{b_0} \text{ when } b_0 > \frac{1}{4} \text{ in. (6 mm)}$$

(b) The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of bolt holes.

NOTE:

(1) Where serrations do not exceed $\frac{1}{64}$ in. (0.4mm) depth and $\frac{1}{32}$ in. (0.8 mm) width spacing, sketches (1b) and (1d) shall be used.

(8b), and (9)], and also for integral type flanges [Figure XI-3120-1 sketch (7)] where the neck material constitutes the hub of the flange.

(b) The longitudinal hub stress S_H shall not be greater than the smaller of $1.5S_f$ or $1.5S_n$ for integral type flanges with hub welded to the neck, pipe, or vessel wall [Figure XI-3120-1 sketches (6), (6a), and (6b)].

(c) The radial flange stress S_R shall not be greater than S_f .

(d) The tangential flange stress S_T shall not be greater than S_f .

(e) Also $(S_H + S_R)/2$ shall not be greater than S_f and $(S_H + S_T)/2$ shall not be greater than S_f .

(f) In the case of loose type flanges with laps, as shown in Figure XI-3120-1 sketches (1) and (1a), where the gasket is so located that the lap is subjected to shear, the shearing stress shall not exceed $0.8S_n$ for the material of the lap, as defined in XI-3130. In the case of welded flanges, shown in Figure XI-3120-1 sketches (3), (4), (7), (8), (8a), and (8b), where the nozzle neck, vessel, or pipe wall extends near to the flange face and may form the gasket contact face, the shearing stress carried by the welds shall not exceed $0.8S_n$. The shearing stress shall be calculated on the basis of W_{m1} or W_{m2} (as defined in XI-3130),

whichever is greater. Similar cases where flange parts are subjected to shearing stress shall be governed by the same requirements.

XI-3260 FLANGES SUBJECT TO EXTERNAL PRESSURE

XI-3261 Flanges for External Pressure Only

The design of flanges for external pressure⁵ only shall be based on the equations given in XI-3240 for internal pressure except that for Design Conditions,

$$M_0 = H_D(h_D - h_G) + H_T(h_T - h_G) \quad (10)$$

for gasket seating,

$$M_0 = Wh_G \quad (11)$$

In eqs. (10) and (11):

$$W = \frac{A_{m2} + A_b S_a}{2}$$

$$H_D = 0.785 B^2 P_e$$

$$H_T = H - H_D$$

$$H = 0.785 G^2 P_e$$

$$P_e = \text{external Design Pressure, psi (kPa)}$$

See XI-3130 for definitions of other symbols.

XI-3262 Flanges for Both External and Internal Pressure

When flanges are subject at different times during service to external or internal pressure, the design shall satisfy the external pressure design requirements⁶ given in XI-3261 and the internal pressure design requirements given elsewhere in this Appendix.

Table XI-3230-1
Moment Arms for Flange Loads

Flange Type	h_D	h_T	h_G
Integral type flanges [see Figure XI-3120-1 sketches (5), (6), (6a), (6b), (7), (8), (8a), (8b), and (9)]	$R + 0.5g_1$	$\frac{R + g_1 + h_G}{2}$	$\frac{C - G}{2}$
Loose type, except lap joint flanges [Figure XI-3120-1 sketches (2), (3), and (4)]; and optional type flanges [Figure XI-3120-1 sketches (8), (8a), (8b), and (9)]	$\frac{C - B}{2}$	$\frac{h_D + h_G}{2}$	$\frac{C - G}{2}$
Lap joint flanges [Figure XI-3120-1 sketches (1) and (1a)]	$\frac{C - B}{2}$	$\frac{C - G}{2}$	$\frac{C - G}{2}$

Table XI-3240-1
Flange Factors in Formula Form

Integral Flange [Note (1)]	Loose Hub Flange [Note (2)]
For F (Figure XI-3240-2) use:	For F_L (Figure XI-3240-4) use:
$F = - \frac{E_6}{\left(\frac{C}{2.73}\right)^{1/4} \frac{(1+A)^3}{C}}$	$F_L = - \frac{C_{18}\left(\frac{1}{2} + \frac{A}{6}\right) + C_{21}\left(\frac{1}{4} + \frac{11A}{84}\right) + C_{24}\left(\frac{1}{70} + \frac{A}{105}\right) - \left(\frac{1}{40} + \frac{A}{72}\right)}{\left(\frac{C}{2.73}\right)^{1/4} \frac{(1+A)^3}{C}}$
For V (Figure XI-3240-3) use:	For V_L (Figure XI-3240-5) use:
$V = \frac{E_4}{\left(\frac{2.73}{C}\right)^{1/4} (1+A)^3}$	$V_L = \frac{\frac{1}{4} - \frac{C_{24}}{5} - \frac{3C_{21}}{2} - C_{18}}{\left(\frac{2.73}{C}\right)^{1/4} (1+A)^3}$
For f (Figure XI-3240-6) use:	For f (Figure XI-3240-6) use:
$f = C_{36} / (1 + A)$	$f = 1$

Equations

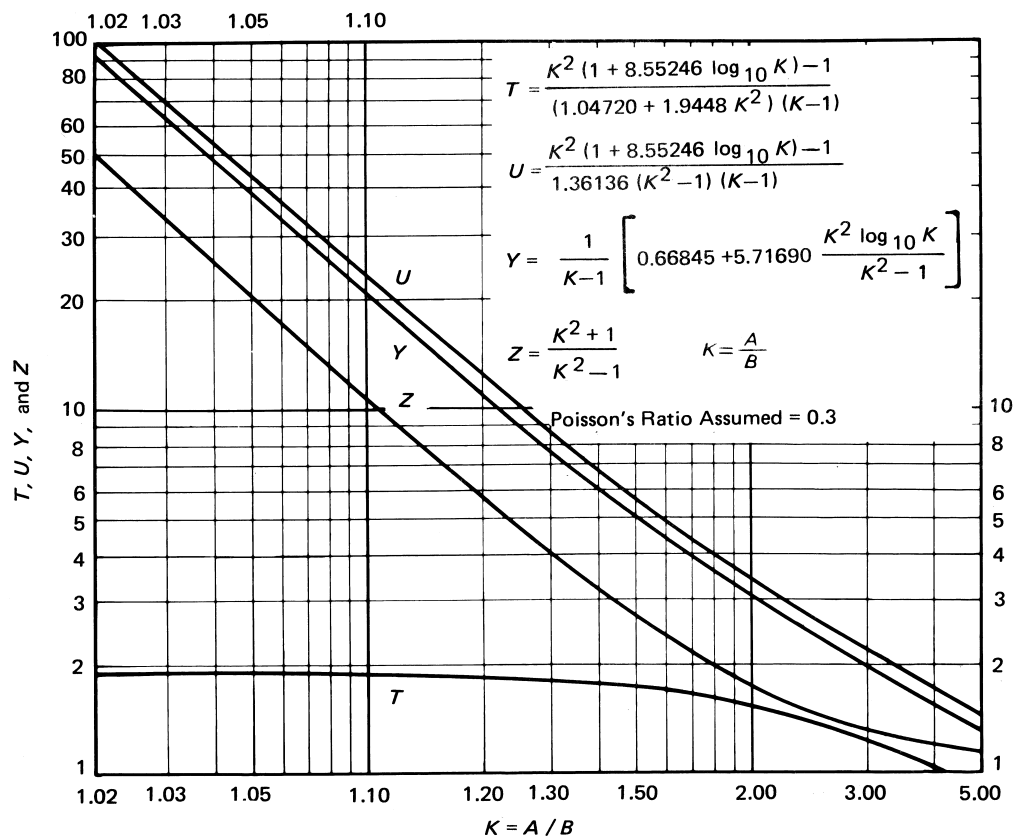
- | | |
|---|--|
| <p>(1) $A = (g_1/g_0) - 1$</p> <p>(3) $C_1 = 1/3 + A/12$</p> <p>(5) $C_3 = 1/210 + A/360$</p> <p>(7) $C_5 = 1/90 + 5A/1008 - (1+A)^3/C$</p> <p>(9) $C_7 = 215/2772 + 51A/1232 + (60/7 + 225A/14 + 75A^2/7 + 5A^3/2)/C$</p> <p>(11) $C_9 = 533/30,240 + 653A/73,920 + (1/2 + 33A/14 + 39A^2/28 + 25A^3/84)/C$</p> <p>(13) $C_{11} = 31/6048 + 1763A/665,280 + (1/2 + 6A/7 + 15A^2/28 + 5A^3/42)/C$</p> <p>(15) $C_{13} = 761/831,600 + 937A/1,663,200 + (1/35 + 6A/35 + 11A^2/70 + 3A^3/70)/C$</p> <p>(17) $C_{15} = 233/831,600 + 97A/554,400 + (1/35 + 3A/35 + A^2/14 + 2A^3/105)/C$</p> <p>(19) $C_{17} = [C_4C_7C_{12} + C_2C_8C_{13} + C_3C_8C_9 - (C_{13}C_7C_3 + C_8^2C_4 + C_{12}C_2C_9)]/C_{16}$</p> <p>(21) $C_{19} = [C_6C_7C_{12} + C_2C_8C_{15} + C_3C_8C_{11} - (C_{15}C_7C_3 + C_8^2C_6 + C_{12}C_2C_{11})]/C_{16}$</p> <p>(23) $C_{21} = [C_1C_{10}C_{12} + C_5C_8C_3 + C_3C_{14}C_2 - (C_3^2C_{10} + C_{14}C_8C_1 + C_{12}C_5C_2)]/C_{16}$</p> <p>(25) $C_{23} = [C_1C_7C_{13} + C_2C_9C_3 + C_4C_8C_2 - (C_3C_7C_4 + C_8C_9C_1 + C_2^2C_{13})]/C_{16}$</p> <p>(27) $C_{25} = [C_1C_7C_{15} + C_2C_{11}C_3 + C_6C_8C_2 - (C_3C_7C_6 + C_8C_{11}C_1 + C_2^2C_{15})]/C_{16}$</p> <p>(29) $C_{27} = C_{20} - C_{17} - 5/12 - [C_{17}(C/4)^{1/4}]$</p> <p>(31) $C_{29} = -(C/4)^{1/2}$</p> <p>(33) $C_{31} = 3A/2 + C_{17}(C/4)^{3/4}$</p> <p>(35) $C_{33} = 0.5C_{26}C_{32} + C_{28}C_{31}C_{29} - (0.5C_{30}C_{28} + C_{32}C_{27}C_{29})$</p> <p>(37) $C_{35} = -C_{18}(C/4)^{3/4}$</p> <p>(39) $C_{37} = [0.5C_{26}C_{35} + C_{34}C_{31}C_{29} - (0.5C_{30}C_{34} + C_{35}C_{27}C_{29})]/C_{33}$</p> <p>(41) $E_2 = C_{20}C_{36} + C_{21} + C_{22}C_{37}$</p> | <p>(2) $C = 43.68(h/h_0)^4$</p> <p>(4) $C_2 = 5/42 + 17A/336$</p> <p>(6) $C_4 = 11/360 + 59A/5040 + (1 + 3A)/C$</p> <p>(8) $C_6 = 1/120 + 17A/5040 + 1/C$</p> <p>(10) $C_8 = 31/6930 + 128A/45,045 + (6/7 + 15A/7 + 12A^2/7 + 5A^3/11)/C$</p> <p>(12) $C_{10} = 29/3780 + 3A/704 - (1/2 + 33A/14 + 81A^2/28 + 13A^3/12)/C$</p> <p>(14) $C_{12} = 1/2925 + 71A/300,300 + (8/35 + 18A/35 + 156A^2/385 + 6A^3/55)/C$</p> <p>(16) $C_{14} = 197/415,800 + 103A/332,640 - (1/35 + 6A/35 + 17A^2/70 + A^3/10)/C$</p> <p>(18) $C_{16} = C_1C_7C_{12} + C_2C_8C_3 + C_3C_8C_2 - (C_3^2C_7 + C_8^2C_1 + C_2^2C_{12})$</p> <p>(20) $C_{18} = [C_5C_7C_{12} + C_2C_8C_{14} + C_3C_8C_{10} - (C_{14}C_7C_3 + C_8^2C_5 + C_{12}C_2C_{10})]/C_{16}$</p> <p>(22) $C_{20} = [C_1C_9C_{12} + C_4C_8C_3 + C_3C_{13}C_2 - (C_3^2C_9 + C_{13}C_8C_1 + C_{12}C_4C_2)]/C_{16}$</p> <p>(24) $C_{22} = [C_1C_{11}C_{12} + C_6C_8C_3 + C_3C_{15}C_2 - (C_3^2C_{11} + C_{15}C_8C_1 + C_{12}C_6C_2)]/C_{16}$</p> <p>(26) $C_{24} = [C_1C_7C_{14} + C_2C_{10}C_3 + C_5C_8C_2 - (C_3C_7C_5 + C_8C_{10}C_1 + C_2^2C_{14})]/C_{16}$</p> <p>(28) $C_{26} = -(C/4)^{1/4}$</p> <p>(30) $C_{28} = C_{22} - C_{19} - 1/12 - [C_{19}(C/4)^{1/4}]$</p> <p>(32) $C_{30} = -(C/4)^{3/4}$</p> <p>(34) $C_{32} = 1/2 + C_{19}(C/4)^{3/4}$</p> <p>(36) $C_{34} = 1/12 + C_{18} - C_{21} + C_{18}(C/4)^{1/4}$</p> <p>(38) $C_{36} = (C_{28}C_{35}C_{29} - C_{32}C_{34}C_{29})/C_{33}$</p> <p>(40) $E_1 = C_{17}C_{36} + C_{18} + C_{19}C_{37}$</p> <p>(42) $E_3 = C_{23}C_{36} + C_{24} + C_{25}C_{37}$</p> |
|---|--|

Table XI-3240-1
Flange Factors in Formula Form (Cont'd)

Integral Flange [Note (1)]	Loose Hub Flange [Note (2)]
(43) $E_4 = \frac{1}{4} + C_{37}/12 + C_{36}/4 - E_3/5 - 3E_2/2 - E_1$	(44) $E_5 = E_1 (\frac{1}{2} + A/6) + E_2 (\frac{1}{4} + 11A/84) + E_3 (\frac{1}{70} + A/105)$
(45) $E_6 = E_5 - C_{36} (\frac{7}{120} + A/36 + 3 A/C) - \frac{1}{40} - A/72 - C_{37} (\frac{1}{60} + A/120 + 1/C)$	

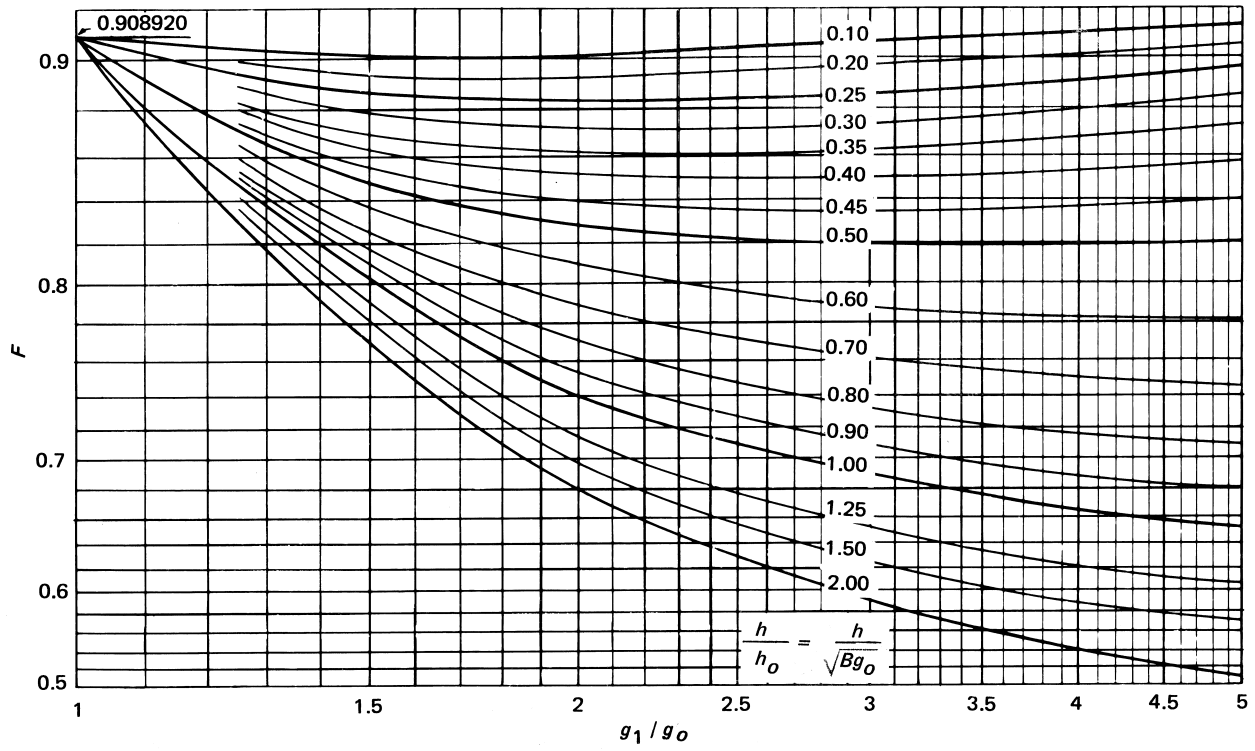
- NOTES:
- (1) Except for the case when $g_1 = g_0$, the values used in the Integral Flange equations are determined by using Eqs. (1) through (45), which are based on the values of g_1, g_0, h , and h_0 (see XI-3130 for definitions). When $g_1 = g_0$, Eqs. (1) through (45) are not required and should not be used. For this case ($g_1 = g_0$), $F = 0.908920$, $V = 0.550103$, and $f = 1$.
- (2) The values used in the Loose Hub Flange equations are determined by using Eqs. (1) through (5), (7), (9), (10), (12), (14), (16), (18), (20), (23), and (26), which are based on the values of g_1, g_0, h , and h_0 (see XI-3130 for definitions).

Figure XI-3240-1
Values of T , U , Y , and Z (Terms Involving K)



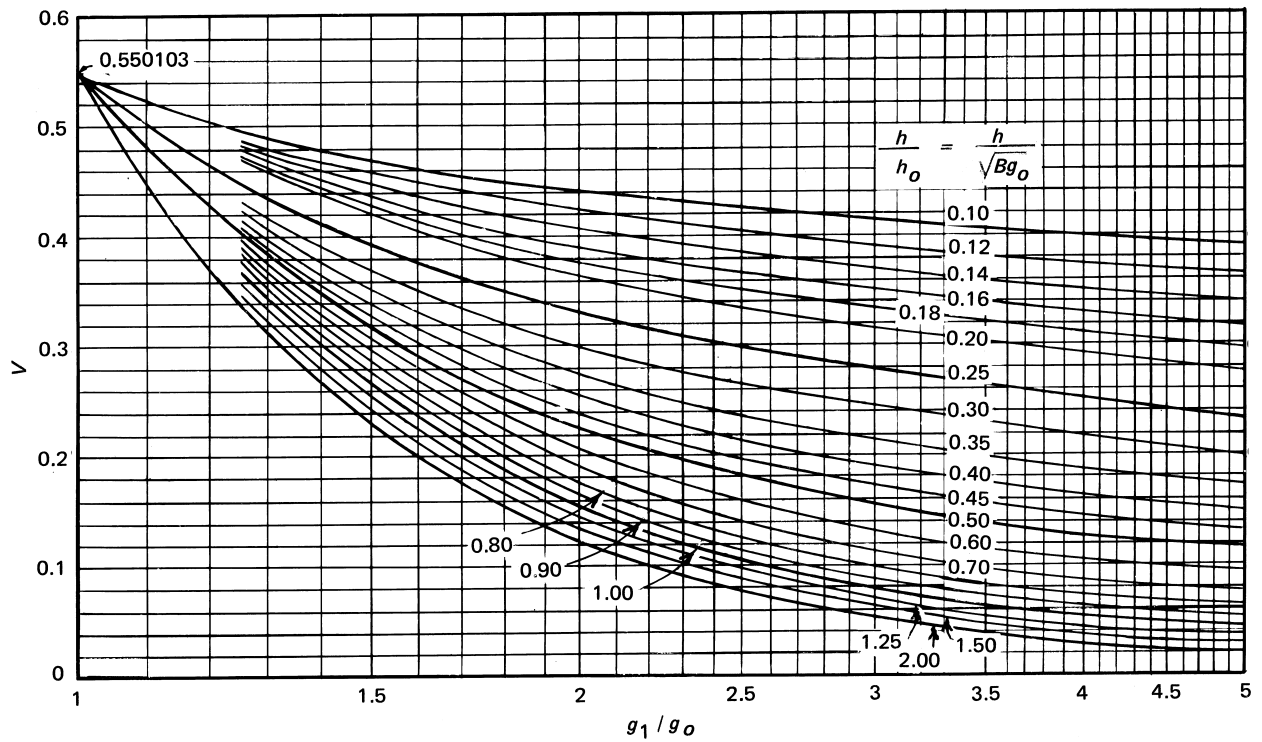
GENERAL NOTE: The calculation of values of T , U , Y , and Z for values of K outside the boundaries of the graph is acceptable.

Figure XI-3240-2
Values of F (Integral Flange Factors)



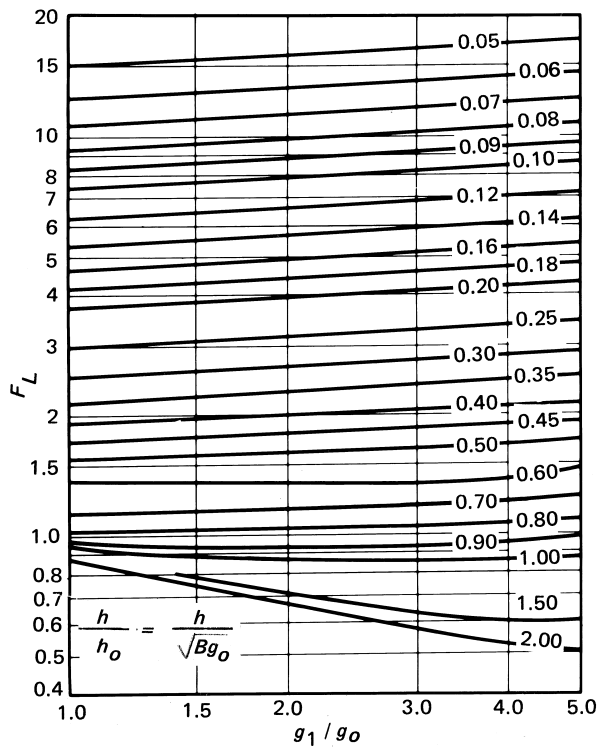
GENERAL NOTE: See [Table XI-3240-1](#) for equations.

Figure XI-3240-3
Values of V (Integral Flange Factors)



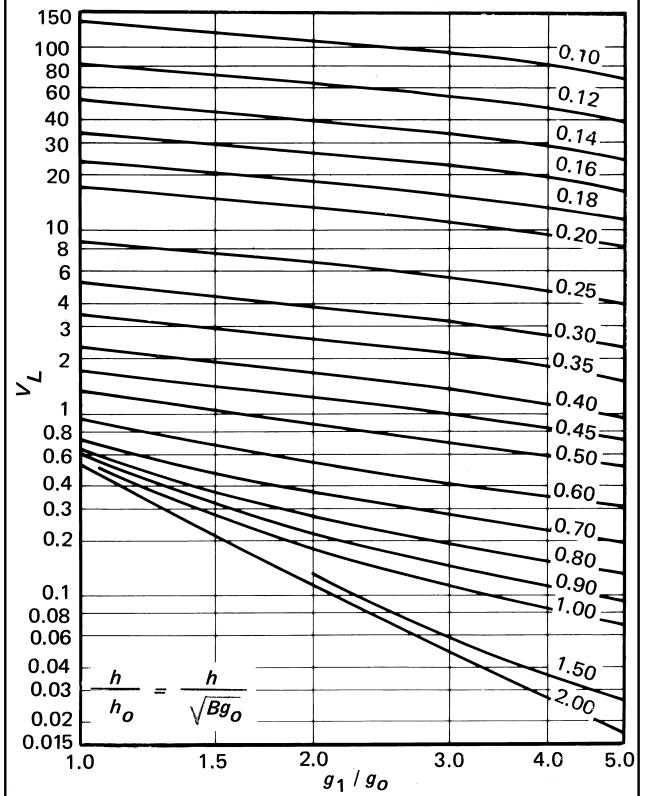
GENERAL NOTE: See [Table XI-3240-1](#) for equations.

Figure XI-3240-4
Values of F_L (Loose Hub Flange Factors)



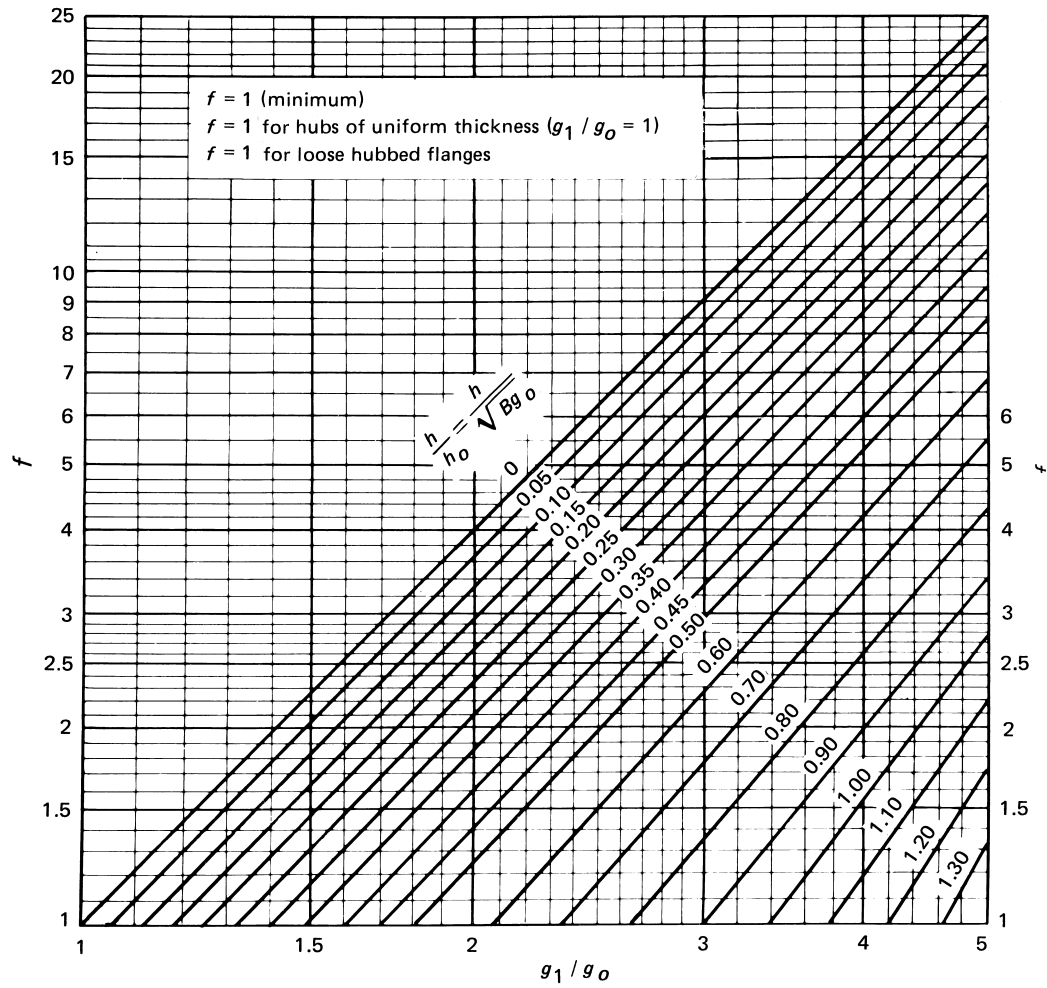
GENERAL NOTE: See Table XI-3240-1 for equations.

Figure XI-3240-5
Values of V_L (Loose Hub Flange Factors)



GENERAL NOTE: See Table XI-3240-1 for equations.

Figure XI-3240-6
Values of f (Hub Stress Correction Factor)



MANDATORY APPENDIX XII

ARTICLE XII-1000

DESIGN CONSIDERATIONS FOR BOLTED FLANGE CONNECTIONS

XII-1100 CONSIDERATIONS

(a) The primary purpose of the rules for bolted flange connections in [Mandatory Appendix XI](#) is to ensure safety, but there are certain other practical matters to be taken into consideration in order to obtain a serviceable design. One of the most important of these is the proportioning of the bolting, i.e., determining the number and size of the bolts.

(b) In the great majority of designs the practice that has been used in the past should be adequate: to follow the design rules in [Mandatory Appendix XI](#) and tighten the bolts sufficiently to withstand the test pressure without leakage. The considerations presented in the discussion in (c) through (m) will be important only when some unusual feature exists, such as a very large diameter, a high Design Pressure, a high temperature, severe temperature gradients, an unusual gasket arrangement, and so on.

(c) The maximum allowable stress values for bolting given in Section II, Part D, Subpart 1, Table 3 are design values to be used in determining the minimum amount of bolting required under the rules. However, a distinction must be kept carefully in mind between the design value and the bolt stress that might actually exist or that might be needed for conditions other than the Design Pressure. The initial tightening of the bolts is a prestressing operation, and the amount of bolt stress developed must be within proper limits to ensure, on the one hand, that it is adequate to provide against all conditions that tend to produce a leaking joint and, on the other hand, that it is not so excessive that yielding of the bolts or flanges can produce relaxation that also can result in leakage.

(d) The first important consideration is the need for the joint to be tight in the hydrostatic test. An initial bolt stress of some magnitude greater than the design value therefore must be provided. If it is not, further bolt strain develops during the test, which tends to part the joint and thereby to decompress the gasket enough to allow leakage. The test pressure is usually $1\frac{1}{2}$ times the Design Pressure, and on this basis it may be thought that 50% extra bolt stress above the design value will be sufficient. However, this is an oversimplification because, on the one hand, the safety factor against leakage under test conditions in general need not be as great as under Design Conditions.

On the other hand, if a stress-strain analysis of the joint is made, it may indicate that an initial bolt stress still higher than $1\frac{1}{2}$ times the design value is needed. Such an analysis is one that considers the changes in bolt elongation, flange deflection, and gasket load that take place with the application of internal pressure, starting from the prestressed condition. In any event, it is evident that an initial bolt stress higher than the design value may and, in some cases, must be developed in the tightening operation, and it is the intent of Subsections NC, ND, and NE that such a practice is permissible, provided it includes necessary and appropriate provision to ensure against excessive flange distortion and gross crushing of the gasket.

(e) It is possible for the bolt stress to decrease after initial tightening, because of slow creep or relaxation of the gasket, particularly in the case of the softer gasket materials. This may be the cause of leakage in the hydrostatic test, in which case it may suffice merely to retighten the bolts. A decrease in bolt stress can also occur in service at elevated temperatures, as a result of creep in the bolt or flange or gasket material, with consequent relaxation. When this results in leakage under service conditions, it is common practice to retighten the bolts, and sometimes a single such operation or perhaps several repeated at long intervals is sufficient to correct the condition. To avoid chronic difficulties of this nature, however, it is advisable when designing a joint for high temperature service to give attention to the relaxation properties of the materials involved, especially for temperatures where creep is the controlling factor in design.

(f) In the other direction, excessive initial bolt stress can present a problem in the form of yielding in the bolting itself and may occur in the tightening operation to the extent of damage or even breakage. This is especially likely with bolts of small diameter and with bolt materials having a relatively low yield strength. The yield strength of mild carbon steel, annealed austenitic stainless steel, and certain of the nonferrous bolting materials can easily be exceeded with ordinary wrench effort in the smaller bolt sizes. Even if no damage is evident, any additional load generated when internal pressure is applied can produce

further yielding with possible leakage. Such yielding can also occur when there is very little margin between initial bolt stress and yield strength.

(g) An increase in bolt stress, above any that may be due to internal pressure, might occur in service during startup or other transient conditions, or perhaps even under normal service. This can happen when there is an appreciable differential in temperature between the flanges and the bolts or when the bolt material has a different coefficient of thermal expansion than the flange material. Any increase in bolt load due to this thermal effect, superposed on the load already existing, can cause yielding of the bolt material, whereas any pronounced decrease due to such effects can result in such a loss of bolt load as to be a direct cause of leakage. In either case, retightening of the bolts may be necessary, but it must not be forgotten that the effects of repeated retightening can be cumulative and may ultimately make the joint unserviceable.

(h) In addition to the difficulties created by yielding of the bolts as described above, the possibility of similar difficulties arising from yielding of the flange or gasket material, under like circumstances or from other causes, should also be considered.

(i) Excessive bolt stress, whatever the reason, may cause the flange to yield even though the bolts may not yield. Any resulting excessive deflection of the flange, accompanied by permanent set, can produce a leaking joint when other effects are superposed. It can also damage the flange by making it more difficult to effect a tight joint thereafter. For example, irregular permanent distortion of the flange due to uneven bolt load around the circumference of the joint can warp the flange face and its gasket contact surface out of a true plane.

(j) The gasket, too, can be overloaded, even without excessive bolt stress. The full initial bolt load is imposed entirely on the gasket, unless the gasket has a stop ring or the flange face detail is arranged to provide the equivalent. Without such means of controlling the compression of the gasket, consideration must be given to the selection of gasket type, size, and material that will prevent gross crushing of the gasket.

(k) From the foregoing, it is apparent that the bolt stress can vary over a considerable range above the design stress value. The design stress values for bolting have been set at a conservative value to provide a factor against yielding. At elevated temperatures, the design stress values are governed by the creep rate and stress rupture strength. Any higher bolt stress existing before creep occurs in operation will have already served its purpose of seating the gasket and holding the hydrostatic test pressure, all at atmospheric temperature, and is not needed at the Design Pressure and Temperature.

(l) Theoretically, the margin against flange yielding is not as great. The design values for flange materials may be as high as five-eighths or two-thirds of the yield strength. However, the highest stress in a flange is usually the bending stress in the hub or shell and is more or less localized. It is too conservative to assume that local yielding is followed immediately by overall yielding of the entire flange. Even if a plastic hinge should develop, the ring portion of the flange takes up the portion of the load that the hub and shell refuse to carry. Yielding is far more significant if it occurs first in the ring but the limitation in the rules on the combined hub and ring stresses provides a safeguard. In this connection, reference should be made to Notes G10 and G8 of Section II, Part D, Subpart 1, Tables 1A and 1B, respectively, which provides guidance in the case of high alloy materials to which a strain limiting factor may have to be applied.

(m) Another very important item in bolting design is the question of whether the necessary bolt stress is actually realized and what special means of tightening, if any, must be employed. Most joints are tightened manually by ordinary wrenching and it is advantageous to have designs that require no more than this. Some pitfalls must be avoided, however. The probable bolt stress developed manually, when using standard wrenches, is

(U.S. Customary Units)

$$S = 45,000 / \sqrt{d}$$

(SI Units)

$$S = 1600 / \sqrt{d}$$

where S is the bolt stress (psi, MPa) and d is the nominal diameter of the bolt (in., mm). It can be seen that smaller bolts will have excessive stress unless judgment is exercised in pulling up on them. On the other hand, it will be impossible to develop the desired stress in very large bolts by ordinary hand wrenching. Impact wrenches may prove serviceable, but, if not, resort may be had to such methods as preheating the bolt or using hydraulically powered bolt tensioners. With some of these methods, control of the bolt stress is possible by means inherent in the procedure, especially if effective thread lubricants are employed, but in all cases the bolt stress can be regulated within reasonable tolerances by measuring the bolt elongation with suitable extensometer equipment. Ordinarily, simple wrenching without verification of the actual bolt stress meets all practical needs, and measured control of the stress is employed only when there is some special or important reason for doing so.

MANDATORY APPENDIX XIII

ARTICLE XIII-1000 DESIGN BASED ON STRESS ANALYSIS

XIII-1100 GENERAL REQUIREMENTS

XIII-1110 SCOPE

This Appendix is applicable only to the design of Class 2 vessels meeting the requirements of NC-3200 and containments meeting the requirements of WC-3200.

(a) When a fatigue evaluation is not required (NC-3219 or WC-3219), a vessel or a vessel part may be designed using the stress analysis methods given in this Appendix. When a fatigue evaluation is required, this Appendix shall be used together with [Mandatory Appendix XIV](#), Design Based on Fatigue Analysis for vessels designed in accordance with NC-3200 or WC-3200.

(b) Use of the analytical methods contained in the Articles of this Appendix following this Article is an acceptable way of obtaining solutions to problems for which these procedures are applicable. Other techniques, analytical or experimental ([Mandatory Appendix II](#)), may be used provided they are either more accurate or more conservative than those contained herein.

XIII-1120 DESIGN ACCEPTABILITY

XIII-1121 Requirements for Design Acceptability

The requirements for the acceptability of a design are given in (a) and (b).

(a) The design shall be such that the stresses shall not exceed the limits described in [XIII-1140](#).

(b) For configurations where compressive stresses occur, in addition to the requirement in (a), the critical buckling stress shall be taken into account. For the case of external pressure, see NC-3240 or WC-3133.

XIII-1122 Basis for Determining Stresses

The equivalent stress at any point in a vessel is the value of stress deduced from the stress condition at the point by means of a theory of failure for comparison with the mechanical properties of the material obtained in tests under uniaxial load. The theory of failure, used in the rules of this Appendix for combining stresses, is the maximum shear stress theory. The maximum shear stress at a point is equal to one-half the difference between the algebraically largest and the algebraically smallest of the three principal stresses at the point.

XIII-1123 Terms Relating to Stress Analysis

Terms used in this Appendix relating to stress analysis are defined in (a) through (af).

(a) *Stress Intensity*.⁷ Stress intensity is defined as twice the maximum shear stress, which is the difference between the algebraically largest principal stress and the algebraically smallest principal stress at a given point. Tensile stresses are considered positive, and compressive stresses are considered negative.

(b) *Gross Structural Discontinuity*. Gross structural discontinuity is a geometric or material discontinuity that affects the stress or strain distribution throughout the entire wall thickness of the pressure retaining member. Gross discontinuity-type stresses are those portions of the actual stress distributions that produce net bending and membrane force resultants when integrated throughout the wall thickness. Examples of a gross structural discontinuity are head-to-shell junctions, flange-to-shell junctions, nozzles, and junctions between shells of different diameters or thicknesses.

(c) *Local Structural Discontinuity*. Local structural discontinuity is a geometric or material discontinuity that affects the stress or strain distribution throughout a fractional part of the wall thickness. The stress distribution associated with a local discontinuity causes only very localized deformation or strain and has no significant effect on the shell-type discontinuity deformations. Examples are small fillet radii, small attachments, and partial penetration welds.

(d) *Normal Stress*. Normal stress is the component of stress normal to the plane of reference. This is also referred to as direct stress. Usually the distribution of normal stress is not uniform through the thickness of a part, so this stress is considered to have two components, one uniformly distributed and equal to the average stress across the thickness under consideration and the other varying from this average value across the thickness.

(e) *Shear Stress*. Shear stress is the component of stress tangent to the plane of reference.

(f) *Membrane Stress*. Membrane stress is the component of normal stress that is uniformly distributed and equal to the average stress across the thickness of the section under consideration.

(g) *Bending Stress.* Bending stress is the component of normal stress that varies across the thickness. The variation may or may not be linear.

(h) *Primary Stress.* Primary stress is any normal stress or shear stress developed by an imposed loading that is necessary to satisfy the laws of equilibrium of external and internal forces and moments. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses that considerably exceed the yield strength will result in failure or, at least, in gross distortion. Primary membrane stress is divided into general and local categories. A general primary membrane stress is one that is so distributed in the structure that no redistribution of load occurs as a result of yielding. Examples of primary stress are

(1) general membrane stress in a circular cylindrical shell or a spherical shell due to internal pressure or to distributed loads

(2) bending stress in the central portion of a flat head due to pressure

Refer to [Table XIII-1130-1](#) for examples of primary stress.

(i) *Secondary Stress.* Secondary stress is a normal stress or a shear stress developed by the constraint of adjacent material or by self-constraint of the structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions that cause the stress to occur and failure from one application of the stress is not to be expected. Examples of secondary stress are

(1) general thermal stress [see (m)]

(2) bending stress at a gross structural discontinuity

Refer to [Table XIII-1130-1](#) for examples of secondary stress.

(j) *Local Primary Membrane Stress.* Cases arise in which a membrane stress produced by pressure or other mechanical loading and associated with a discontinuity would, if not limited, produce excessive distortion in the transfer of load to other portions of the structure. Conservatism requires that such a stress be classified as a local primary membrane stress even though it has some characteristics of a secondary stress.

A stressed region may be considered local if the distance over which the membrane stress intensity exceeds $1.1S_m$ does not extend in the meridional direction more than $1.0\sqrt{Rt}$, where R is the minimum midsurface radius of curvature and t is the minimum thickness in the region considered. Regions of local primary stress intensity involving axisymmetric membrane stress distributions which exceed $1.1S_m$ shall not be closer in the meridional direction than $2.5\sqrt{R_L t_L}$, where R_L is defined as $(R_1 + R_2)/2$ and t_L is defined as $(t_1 + t_2)/2$ where t_1 and t_2 are the minimum thicknesses at each of the regions considered, and R_1 and R_2 are the minimum midsurface radii of curvature at these regions where the membrane stress intensity exceeds $1.1S_m$. Discrete regions of local primary

membrane stress intensity, such as those resulting from concentrated loads acting on brackets, where the membrane stress intensity exceeds $1.1S_m$ shall be spaced so that there is no overlapping of the areas in which the membrane stress intensity exceeds $1.1S_m$.

(k) *Peak Stress.* Peak stress is that increment of stress that is additive to the primary plus secondary stresses by reason of local discontinuities or local thermal stress [see (m)(2)] including the effects, if any, of stress concentrations. The basic characteristic of a peak stress is that it does not cause any noticeable distortion and is objectionable only as a possible source of a fatigue crack or a brittle fracture. A stress that is not highly localized falls into this category if it is of a type that cannot cause noticeable distortion. Examples of peak stress are

(1) the thermal stress in the austenitic steel cladding of a carbon steel part

(2) certain thermal stresses that may cause fatigue but not distortion

(3) the stress at a local structural discontinuity

(4) surface stresses produced by thermal shock

(l) *Load Controlled Stress.* Load controlled stress is the stress resulting from the application of a loading, such as internal pressure, inertial loads, or gravity, whose magnitude is not reduced as a result of displacement.

(m) *Thermal Stress.* Thermal stress is a self-balancing stress produced by a nonuniform distribution of temperature or by differing thermal coefficients of expansion. Thermal stress is developed in a solid body whenever a volume of material is prevented from assuming the size and shape that it normally would under a change in temperature. For the purpose of establishing allowable stresses, two types of thermal stress are recognized, depending on the volume or area in which distortion takes place, as described in (1) and (2) below.

(1) General thermal stress is associated with distortion of the structure in which it occurs. If a stress of this type, neglecting stress concentrations, exceeds twice the yield strength of the material, the elastic analysis may be invalid and successive thermal cycles may produce incremental distortion. Therefore, this type is classified as secondary stress in [Table XIII-1130-1](#) and [Figure XIII-1141-1](#). Examples of general thermal stress are

(-a) stress produced by an axial temperature distribution in a cylindrical shell

(-b) stress produced by the temperature difference between a nozzle and the shell to which it is attached

(-c) the equivalent linear stress⁸ produced by the radial temperature distribution in a cylindrical shell

(2) Local thermal stress is associated with almost complete suppression of the differential expansion and thus produces no significant distortion. Such stresses shall be considered only from the fatigue standpoint and are therefore classified as peak stresses in [Table XIII-1130-1](#) and [Figure XIII-1141-1](#). In evaluating local thermal stresses the procedures of [XIII-1151.1\(b\)](#) shall be used. Examples of local thermal stress are

(-a) the stress in a small hot spot in a vessel wall
 (-b) the difference between the actual stress and the equivalent linear stress resulting from a radial temperature distribution in a cylindrical shell

(-c) the thermal stress in a cladding material that has a coefficient of expansion different from that of the base metal

(n) *Total Stress*. Total stress is the sum of the primary, secondary, and peak stress contributions. Recognition of each of the individual contributions is essential to establishment of appropriate limitations.

(o) *Service Cycle*. Service cycle is defined as the initiation and establishment of new conditions followed by a return to the conditions that prevailed at the beginning of the cycle. The types of service conditions that may occur are further defined in the Design Specification as required by NCA-2142 and WA-2123.

(p) *Stress Cycle*. Stress cycle is a condition in which the alternating stress difference (XIV-1212) goes from an initial value through an algebraic maximum value and an algebraic minimum value and then returns to the initial value. A single service cycle may result in one or more stress cycles. Dynamic effects shall also be considered as stress cycles.

(q) *Fatigue Strength Reduction Factor*. Fatigue strength reduction factor is a stress intensification factor which accounts for the effect of a local structural discontinuity (stress concentration) on the fatigue strength. Values for some specific cases, based on experiment, are given elsewhere and in Subsection NC or Subsection WC. In the absence of experimental data, the theoretical stress concentration factor may be used.

(r) *Shakedown*. Shakedown of a structure occurs if, after a few cycles of load application, ratcheting ceases. The subsequent structural response is elastic, or elastic-plastic, and progressive incremental inelastic deformation is absent. Elastic shakedown is the case in which the subsequent response is elastic.

(s) *Deformation*. Deformation of a component part is an alteration of its shape or size.

(t) *Inelasticity*. Inelasticity is a general characteristic of material behavior in which the material does not return to its original shape and size after removal of all applied loads. Plasticity and creep are special cases of inelasticity.

(u) *Creep*. Creep is the special case of inelasticity that relates to the stress induced time-dependent deformation under load. Small time-dependent deformations may occur after the removal of all applied loads.

(v) *Plasticity*. Plasticity is the special case of inelasticity in which the material undergoes time-independent nonrecoverable deformation.

(w) *Plastic Analysis*. Plastic analysis is that method which computes the structural behavior under given loads considering the plasticity characteristics of the materials including strain hardening and the stress redistribution occurring in the structure.

(x) *Plastic Analysis — Collapse Load*. A plastic analysis may be used to determine the collapse load for a given combination of loads on a given structure. The following criterion for determination of the collapse load shall be used. A load-deflection or load-strain curve is plotted with load as the ordinate and deflection or strain as the abscissa. The angle that the linear part of the load-deflection or load-strain curve makes with the ordinate is called θ . A second straight line, hereafter called the collapse limit line, is drawn through the origin so that it makes an angle $\phi = \tan^{-1} (2 \tan \theta)$ with the ordinate. The collapse load is the load at the intersection of the load-deflection or load-strain curve and the collapse limit line. If this method is used, particular care should be given to ensure that the strains or deflections that are used are indicative of the load carrying capacity of the structure.

(y) *Plastic Instability Load*. The plastic instability load for members under predominantly tensile or compressive loading is defined as that load at which unbounded plastic deformation can occur without an increase in load. At the plastic tensile instability load the true stress in the material increases faster than strain hardening can accommodate.

(z) *Limit Analysis*. Limit analysis is a special case of plastic analysis in which the material is assumed to be ideally plastic (nonstrain-hardening). In limit analysis the equilibrium and flow characteristics at the limit state are used to calculate the collapse load. The two bounding methods which are used in limit analysis are the lower bound approach, which is associated with a statically admissible stress field, and the upper bound approach, which is associated with a kinematically admissible velocity field. For beams and frames the term *mechanism* is commonly used in lieu of *kinematically admissible velocity field*.

(aa) *Collapse Load — Lower Bound*. If, for a given load, any system of stresses can be found which everywhere satisfies equilibrium, and nowhere exceeds the material yield strength, the load is at or below the collapse load. This is the lower bound theorem of limit analysis which permits calculations of a lower bound to the collapse load.

(ab) *Limit Analysis — Collapse Load*. The methods of limit analysis are used to compute the maximum load that a structure assumed to be made of ideally plastic material can carry. At this load, which is termed the collapse load, the deformations of the structure increase without bound.

(ac) *Plastic Hinge*. A plastic hinge is an idealized concept used in Limit Analysis. In a beam or a frame, a plastic hinge is formed at the point where the moment, shear, and axial force lie on the yield interaction surface. In plates and shells a plastic hinge is formed where the generalized stresses lie on the yield surface.

(ad) *Strain Limiting Load*. When a limit is placed upon a strain, the load associated with the strain limit is called the strain limiting load.

(ae) *Test Collapse Load*. Test collapse load is the collapse load determined by tests according to the criteria given in II-1430 of Section III.

(af) *Ratcheting*. Ratcheting is a progressive incremental inelastic deformation or strain which can occur in a component that is subjected to variations of mechanical stress, thermal stress, or both.

XIII-1130 DERIVATION OF STRESS INTENSITIES

One requirement for the acceptability of a design [XIII-1121(a)] is that the calculated stress intensities shall not exceed specified allowable limits. These limits differ depending on the stress category (primary, secondary, etc.) from which the stress intensity is derived. This paragraph describes the procedure for the calculation of the stress intensities which are subject to specified limits. The steps in the procedure are given in (a) through (f).

(a) At the point on the vessel which is being investigated, choose an orthogonal set of coordinates, such as tangential, longitudinal, and radial, and designate them by the subscripts t , l , and r . The stress components in these directions are then designated σ_t , σ_r , and σ_l for direct stresses and τ_{tl} , τ_{lr} , and τ_{rt} for shear stresses.

(b) Calculate the stress components for each type of loading to which the part will be subjected and assign each set of stress values to one or a group of the following categories:⁹

(1) general primary membrane stress P_m [XIII-1123(f) and XIII-1123(h)]

(2) local primary membrane stress P_L [XIII-1123(j)]

(3) primary bending stress P_b [XIII-1123(g) and XIII-1123(h)]

(4) secondary stress¹⁰ Q [XIII-1123(i)]

(5) peak stress F [XIII-1123(k)]

(c) Group the stress components in accordance with (b). Figure XIII-1141-1 is to provide assistance in assigning the stress values to the appropriate category. At any rectangular box calculate the algebraic sum of the values of σ_t which result from the different types of loadings and which have entered the box, and similarly for the other five stress components. The result is a set of six stress components in each box.

(d) Translate the stress components in the t , l , and r directions into principal stresses σ_1 , σ_2 , and σ_3 . (In many pressure vessel calculations, the t , l , and r directions may be so chosen that the shearing stress components are zero and so σ_1 , σ_2 , and σ_3 are identical to σ_t , σ_l , and σ_r .)

(e) Calculate the stress differences S_{12} , S_{23} , and S_{31} from the relations

$$S_{12} = \sigma_1 - \sigma_2$$

$$S_{23} = \sigma_2 - \sigma_3$$

$$S_{31} = \sigma_3 - \sigma_1$$

The stress intensity S is the largest absolute value of S_{12} , S_{23} , and S_{31} .

(f) The stress intensity calculated as in the preceding (e) from the stress components in any rectangle in Figure XIII-1141-1 shall not exceed the allowable values of XIII-1140 which are shown in the circle adjacent to the rectangle in Figure XIII-1141-1.

NOTE: Membrane stress intensity is derived from the stress components averaged across the thickness of the section. The averaging shall be performed at the component level, in step (b) or (c) above.

XIII-1140 BASIC STRESS INTENSITY LIMITS

XIII-1141 Five Basic Limits

The five basic stress intensity limits which are to be satisfied are given in XIII-1142 through XIII-1146 (Figure XIII-1141-1).

XIII-1142 General Primary Membrane Stress Intensity

This stress intensity is derived from the average value across the thickness of a section of the general primary stresses [XIII-1123(h)] produced by

(a) Design Pressure and specified Design Mechanical Loads for Design Loadings

(b) coincident pressure and mechanical loads associated with each Service Loading as specified in the Design Specification, but excluding all secondary and peak stresses.

Averaging is to be applied to the stress components prior to the determination of the stress intensity values. The allowable value of this stress intensity is kS_m , with S_m as given in Section II, Part D, Subpart 1, Tables 2A and 2B. Values of k , reflecting the specific Design or Service Limit under consideration, are given in Table NC-3217-1 or Table WC-3217-1 as appropriate.

XIII-1143 Local Membrane Stress Intensity

This stress intensity is derived from the average value across the thickness of a section of the local primary stresses [XIII-1123(j)] produced by

(a) Design Pressure and specified Design Mechanical Loads for Design Loadings

(b) coincident pressure and mechanical loads associated with each Service Loading as specified in the Design Specification, but excluding all thermal and peak stresses.

Averaging is to be applied to the stress components prior to the determination of the stress intensity values. The allowable value of this stress intensity is $1.5kS_m$, with S_m as given in Section II, Part D, Subpart 1, Tables 2A and 2B. Values of k , reflecting the specific Design or Service Limit under consideration, are given in Table NC-3217-1 or Table WC-3217-1 as appropriate.

Table XIII-1130-1
Classification of Stress Intensity in Vessels for Some Typical Cases

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
Cylindrical or spherical shell	Shell plate remote from discontinuities	Internal pressure	General membrane	P_m
			Gradient through plate thickness	Q
		Axial thermal gradient	Membrane	Q
			Bending	
	Junction with head or flange	Internal pressure	Membrane	P_L
			Bending	Q [Note (1)]
Any shell or head	Any section across entire vessel	External load or moment, or internal pressure	General membrane averaged across full section	P_m
		External load or moment	Bending across full section	
	Near nozzle or other opening	External load or moment, or internal pressure	Local membrane	P_L
			Bending	Q
			Peak (fillet or corner)	F
	Any location	Temperature difference between shell and head	Membrane	
			Bending	Q
Dished head or conical head	Crown	Internal pressure	Membrane	P_m
			Bending	P_b
	Knuckle or junction to shell	Internal pressure	Membrane	P_L [Note (2)]
			Bending	Q
Flat head	Center region	Internal pressure	Membrane	P_m
			Bending	P_b
	Junction to shell	Internal pressure	Membrane	P_L
			Bending	P_b or Q [Note (1)]
Perforated head or shell	Typical ligament in a uniform pattern	Pressure	Membrane (averaged through cross section)	P_m
			Bending (averaged through width of ligament, but gradient through plate)	P_b
			Peak	F
	Isolated or atypical ligament	Pressure	Membrane	Q
			Bending	F
			Peak	F

Table XIII-1130-1
Classification of Stress Intensity in Vessels for Some Typical Cases (Cont'd)

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
Nozzle (XIII-1170)	Within the limits of reinforcement defined by NC-3334 or WC-3230	Pressure and external loads and moments, including those attributable to restrained free end displacements of the attached piping	General membrane	P_m
			Bending (other than gross structural discontinuity stresses) averaged through nozzle thickness	
	Outside the limits of reinforcement defined by NC-3334 or WC-3230	Pressure and external axial, shear, and torsional loads other than those attributable to restrained free end displacements of the attached piping	General membrane stresses	P_m
		Pressure and external loads and moments other than those attributable to restrained free end displacements of the attached piping	Membrane	P_L
			Bending	P_b
		Pressure and all external loads and moments	Membrane	P_L
			Bending	Q
			Peak	F
	Nozzle wall	Gross structural discontinuities	Local membrane	P_L
			Bending	Q
			Peak	F
		Differential expansion	Membrane	Q
			Bending	
			Peak	F
Cladding	Any	Differential expansion	Membrane	F
			Bending	
Any	Any	Radial temperature distribution [Note (4)]	Equivalent linear stress [Note (3)]	Q
			Nonlinear portion of stress distribution	F
Any	Any	Any	Stress concentration (notch effect)	F

NOTES:

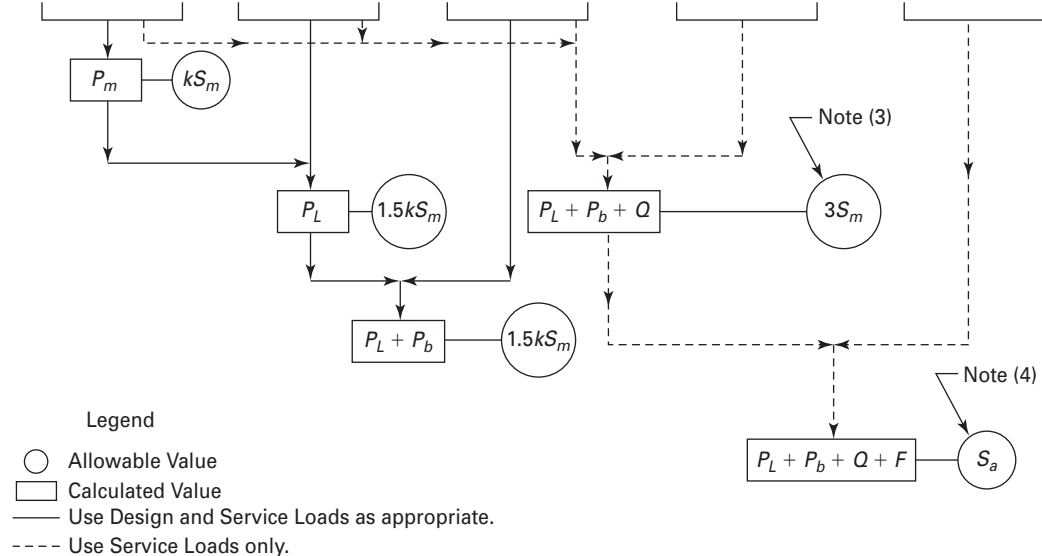
- (1) If the bending moment at the edge is required to maintain the bending stress in the middle to acceptable limits, the edge bending is classified as P_b . Otherwise, it is classified as Q .
- (2) Consideration shall also be given to the possibility of wrinkling and excessive deformation in vessels with a large diameter-thickness ratio.
- (3) Consider possibility of thermal stress ratchet.
- (4) Equivalent linear stress is defined as the linear stress distribution that has the same net bending moment as the actual stress distribution.

(13)

Figure XIII-1141-1
Stress Categories and Limits of Stress Intensity

Stress Category	Primary			Secondary Membrane Plus Bending	Peak
	General Membrane	Local Membrane	Bending		
Description (For examples, see Table XIII-1130-1)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced by pressure and mechanical loads.	Average stress across any solid sections. Considers discontinuities but not concentrations. Produced by pressure and mechanical loads.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced by pressure and mechanical loads.	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by pressure, mechanical loads, or by differential thermal expansion. Excludes local stress concentrations.	(1) Increment added to primary or secondary stress by a concentration (notch). (2) Certain thermal stresses which may cause fatigue but not distortion of vessel shape.
Symbol [Note (1)]	P_m	P_L	P_b	Q [Note (2)]	F

Combination of stress components and allowable limits of stress intensities



NOTES:

- (1) The symbols P_m , P_L , P_b , Q , and F do not represent single quantities but rather sets of six quantities representing the six stress components σ_t , σ_l , σ_r , τ_{lt} , τ_{lr} , and τ_{rt} .
- (2) The stresses in category Q are those parts of the total stress which are produced by thermal gradients, structural discontinuities, etc., and do not include primary stresses which may also exist at the same point. It should be noted, however, that a detailed stress analysis frequently gives the combination of primary and secondary stresses directly and, when appropriate, this calculated value represents the total of P_m (or P_L) + P_b + Q and not Q alone. Similarly, if the stress in category F is produced by a stress concentration, the quantity F is the additional stress, produced by the notch, over and above the nominal stress. For example, if a plate has a nominal stress intensity S and has a notch with a stress concentration factor K , then $P_m = S$, $P_b = 0$, $Q = 0$, $F = P_m (K - 1)$, and the total stress intensity equals $P_m + P_m (K - 1) = KP_m$.
- (3) This limitation applies to the range of stress intensity. When the secondary stress is due to a temperature excursion at the point at which the stresses are being analyzed, the value of S_m shall be taken as the average of the S_m values tabulated in Section II, Part D, Subpart 1, Tables 2A and 2B for the highest and the lowest temperature of the metal during the transient. When part or all of the secondary stress is due to mechanical load, the value of S_m shall be taken as the S_m value for the highest temperature of the metal during the transient.
- (4) S_a is obtained from the fatigue curves (Section III Appendices, Mandatory Appendix I, Figures I-9.1 through I-9.8). The allowable stress intensity for the full range of fluctuation is $2 S_a$.

XIII-1144 Primary Membrane Plus Primary Bending Stress Intensity

This stress intensity is derived from the highest value across the thickness of a section of the general or local primary membrane stresses plus primary bending stresses produced by

(a) Design Pressure and specified Design Mechanical Loads for Design Loadings

(b) coincident pressure and mechanical loads associated with each Service Loading as specified in the Design Specification, but excluding all secondary and peak stresses.

For solid rectangular sections, the allowable value of this stress intensity is $1.5kS_m$, with S_m as given in Section II, Part D, Subpart 1, Tables 2A and 2B. Values of k , reflecting the appropriate Design or Service Limit under consideration, are given in Table NC-3217-1 or Table WC-3217-1 as appropriate. For other than solid rectangular sections, a value of α times the limit established in NC-3217(b) or WC-3217(b) may be used, where the factor α is defined as the ratio of the load set producing a fully plastic section to the load set producing initial yielding in the extreme fibers of the section. In the evaluation of the initial yield and fully plastic section capacities, the ratios of each individual load in the respective load set to each other load in that load set shall be the same as the respective ratios of the individual loads in the specified design load set. The value of α shall not exceed the value calculated for bending only ($P_m = 0$). In no case shall the value of α exceed 1.5. The propensity for buckling of the part of the section that is in compression shall be investigated. The α factor is not permitted for Level D Service Limits when inelastic component analysis is used as permitted in [Nonmandatory Appendix F](#).

XIII-1145 Primary Plus Secondary Stress Intensity

This is the stress intensity, derived from the highest value at any point across the thickness of a section, of the general or local primary membrane stresses plus primary bending stresses plus secondary stresses produced by specified service pressure and other specified mechanical loads and by general thermal effects. The effects of gross structural discontinuities but not of local structural discontinuities (stress concentrations) shall be included. The allowable value of this stress intensity is $3S_m$ [Note (3) of [Figure XIII-1141-1](#)].

XIII-1146 Peak Stress Intensity

This is the stress intensity, derived from the highest value at any point across the thickness of a section, of the combination of all primary, secondary, and peak stresses produced by specified service pressures and other mechanical loads and by general and local thermal effects and including the effects of gross and local structural discontinuities. The allowable value of this stress intensity is

dependent on the range of the stress difference from which it is derived and on the number of times it is to be applied. The stress intensity is to be compared with the allowable value obtained by the methods of analysis for cyclic operation described in [XIV-1200](#) through the use of the fatigue curves.

XIII-1150 PLASTIC ANALYSIS, LIMIT ANALYSIS, AND SHAKEDOWN ANALYSIS

XIII-1151 Plastic Analysis

(a) Certain of the allowable stresses permitted in these design criteria are such that the maximum stress calculated on an elastic basis may exceed the yield strength of the material. The limit on primary plus secondary stress intensity of $3S_m$ ([XIII-1145](#)) has been placed at a level which ensures shakedown to elastic action after a few repetitions of the stress cycle except in regions containing significant local structural discontinuities or local thermal stresses. These last two factors are considered only in the performance of a fatigue evaluation.

(b) The limits on local membrane stress intensity ([XIII-1143](#)) and primary membrane plus primary bending stress intensity of $1.5kS_m$ ([XIII-1144](#)) have been placed at a level which conservatively ensures the prevention of collapse as determined by the principles of limit analysis. The following subparagraphs provide guidance in the application of plastic analysis and some relaxation of the basic stress limits which are allowed if plastic analysis is used.

XIII-1151.1 Value of Poisson's Ratio.

(a) In evaluating stresses for comparison with any stress limits other than fatigue allowables, stresses shall be calculated on an elastic basis using the elastic value of Poisson's ratio.

(b) In evaluating stresses for comparison with fatigue allowables, the elastic equations shall be used, except that the numerical value substituted for Poisson's ratio shall be determined from the expression:

$$\nu = 0.5 - 0.2(S_y/S_a), \text{ but not less than } 0.3$$

where

S_a = alternating stress intensity determined in [XIV-1221.3\(a\)](#) or [XIV-1221.3\(b\)](#) prior to the elastic modulus adjustment in [XIV-1221.3\(d\)](#).

S_y = the yield strength of the material at the mean value of the temperature of the cycle

XIII-1151.2 Plastic Analysis Procedures. The limits on local membrane stress intensity ([XIII-1143](#)), primary plus secondary stress intensity ([XIII-1145](#)), thermal stress ratchet in shell ([XIV-1410](#)), and thermal stress in nonintegral connections ([XIV-1420](#)) need not be satisfied at a specific location if at that location the procedures of the following (a), (b), and (c) are used.

(a) In evaluating stresses for comparison with the stress limits of [XIII-1142](#), [XIII-1144](#) (general primary membrane plus primary bending stress intensity only), [XIII-1180](#), [NC-3216.3\(a\)](#), [XIII-1160](#), and [XIII-1121\(b\)](#), the stresses are calculated on an elastic basis.

(b) In lieu of satisfying the specific requirements of [XIII-1143](#), [XIII-1145](#), [NC-3216.3\(b\)](#), and [XIV-1420](#) at a specific location, the structural action is calculated on a plastic basis and the design shall be considered to be acceptable if shakedown occurs, as opposed to continuing deformation, and if the deformations which occur prior to shakedown do not exceed specified limits.

(c) In evaluating stresses for comparison with fatigue allowables, the numerically maximum principal total strain range which occurs after shakedown shall be multiplied by one-half of the modulus of elasticity of the material at the mean value of the temperature of the cycle.

XIII-1152 Limit Analysis

The limits on local membrane stress intensity ([XIII-1143](#)) and primary membrane plus primary bending stress intensity need not be satisfied at a specific location if it can be shown by means of limit analysis or by tests that the specified mechanical and thermal loadings do not exceed two-thirds the lower bound collapse load.

XIII-1153 Shakedown Analysis

The $3S_m$ stress intensity limit on the range of primary plus secondary stress intensity ([XIII-1145](#)) may be exceeded provided that the range of primary plus secondary membrane plus bending stress intensity, excluding thermal bending stresses, shall be $\leq 3S_m$ and that the following requirements of (a) through (e) are met.

(a) The value of S_a used for entering the design fatigue curve is multiplied by the factor K_e , where

$$K_e = 1.0, \text{ for } S_n \leq 3S_m$$

$$= 1.0 + [(1 - n)/n(m - 1)][(S_n)/(3S_m) - 1], \text{ for } 3S_m < S_n < 3mS_m$$

$$= 1/n, \text{ for } S_n \geq 3mS_m$$

S_n = range of primary plus secondary stress intensity

The values of the material parameters m and n are given for the various classes of permitted materials in [Table XIII-1153\(a\)-1](#).

(b) The rest of the fatigue evaluation stays the same as required in [Mandatory Appendix XIV](#), except that the procedure of [XIII-1151.1](#) need not be used.

(c) The component meets the thermal ratcheting requirement of [Appendix XIV-1400](#).

(d) The temperature does not exceed those listed in [Table XIII-1153\(a\)-1](#) for the various classes of permitted materials.

(e) The material shall have a specified minimum yield strength to specified minimum tensile strength ratio of less than 0.80.

Table XIII-1153(a)-1
Values of m , n , and T_{max} for Various Classes of Permitted Materials

Materials	m	n	T_{max} , °F (°C)
Low alloy steel	2.0	0.2	700 (370)
Martensitic stainless steel	2.0	0.2	700 (370)
Carbon steel	3.0	0.2	700 (370)
Austenitic stainless steel	1.7	0.3	800 (425)
Nickel-chromium-iron	1.7	0.3	800 (425)
Nickel-copper	1.7	0.3	800 (425)

XIII-1160 TRIAXIAL STRESSES

The algebraic sum of the three primary principal stresses ($\sigma_1 + \sigma_2 + \sigma_3$) shall not exceed four times the tabulated value of S_m .

XIII-1170 NOZZLE PIPING TRANSITION

The P_m classification of stresses in nozzles resulting from external load or moment is applicable for that length of nozzle which lies within the limits of reinforcement whether or not nozzle reinforcement is required. Beyond the limits of reinforcement, a P_L classification may be applied to the stresses resulting from external load or moment attributable to thermal expansion of the attached piping. However, the pressure and mechanical load induced membrane stresses retain a P_m classification.

XIII-1180 BOLTING

XIII-1181 Bolt and Gasket Requirements

(a) The number and cross-sectional area of bolts required to resist internal pressure shall be determined in accordance with the procedures of [Mandatory Appendix XI](#). The allowable bolt design stresses, as used in the equations of [Mandatory Appendix XI](#), shall be the values given in Section II, Part D, Subpart 1, Table 4 for bolting materials. When sealing is effected by a seal weld instead of a gasket, the gasket factor m and the minimum design seating stress y may be taken as zero.

(b) When gaskets are used for preservice testing only, the design is satisfactory if the above requirements are satisfied for $m = y = 0$ and the requirements of [XIII-1182](#) are satisfied when the appropriate m and y are used for the test gasket.

XIII-1182 Allowable Maximum Service Stresses in Bolts

It is recognized that actual service stresses in bolts, such as those produced by the combination of preload, pressure, and differential thermal expansion, may be higher than the values given in Section II, Part D, Subpart 1, Table 4. The maximum of such service stress, averaged across the bolt cross section and neglecting stress concentrations, shall not exceed two times the stress values of

Section II, Part D, Subpart 1, Table 4. Except as restricted by XIV-1322(b), the maximum value of such service stress at the periphery of the bolt cross section resulting from direct tension plus bending and neglecting stress concentrations shall not exceed three times the stress values of

Section II, Part D, Subpart 1, Table 4. Stress intensity, rather than maximum stress, shall be limited to this value when the bolts are tightened by methods other than heaters, stretchers, or other means which minimize residual torsion.

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ARTICLE XIII-2000

PRESSURE STRESSES IN OPENINGS FOR FATIGUE EVALUATION

XIII-2100 METHODS OF EVALUATION

XIII-2110 ACCEPTABLE METHODS

For the purpose of detecting peak stresses around the opening, three acceptable methods are listed in (a), (b), and (c) below.

(a) *Analytical Method.* This method uses suitable analytical techniques, such as finite element computer analyses, which provide detailed stress distributions around openings. In addition to peak stresses due to pressure, the effects of other loadings shall be included. The total peak stress at any given point shall be determined by combining stresses due to pressure, thermal, and external loadings in accordance with the rules of XIII-1000.

(b) *Experimental Stress Analysis.* This is based on data from experiments (XIII-2130).

(c) *Stress Index Method.* This uses various equations based on data obtained from an extensive series of tests covering a range of variations of applicable dimensional ratios and configurations (para. XIII-2120). This method covers only single, isolated openings. Stress indices may also be determined by theoretical or experimental stress analysis. Such analysis may be included in the Design Report.

XIII-2120 STRESS INDEX METHOD

XIII-2121 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 computed membrane hoop 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 wall is increased over that required, the values of r_1 and r_2 in Figure XIII-2124(e)-1 shall be referred to the thickened section.

XIII-2122 Nomenclature

The nomenclature for the stress components are shown in Figure XIII-2122-1 and are defined as follows:

- R = inside radius, in corroded condition, of cylindrical vessel, spherical vessel, or spherical head
- S = the stress intensity at the point under consideration
- t = nominal wall thickness less corrosion allowance, of vessel or head

σ_n = the stress component normal to the plane of the section, such as the circumferential stress around the hole in the shell

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

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

XIII-2123 Stress Indices for Nozzles

When the conditions of XIII-2122 are satisfied, the stress indices given in Tables XIII-2123-1 and XIII-2123-2 may be used for nozzles designed in accordance with the applicable rules of NC-3230 or WC-3230, and NC-3259 or WC-3259. These stress indices deal only with the maximum stresses, at certain general locations, due to internal pressure. 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. In such cases, the total stress at a given point may be determined by superposition. In the case of combined stresses due to internal pressure and nozzle loading, the maximum stresses for a given location shall be considered as acting at the same point and added algebraically unless positive evidence is available to the contrary.

Figure XIII-2122-1
Direction of Stress Components

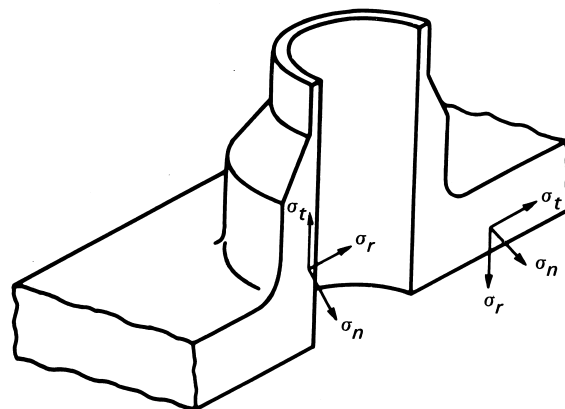


Table XIII-2123-1
Nozzles in Spherical Shells and Formed Heads

Stress	Inside Corner	Outside Corner
σ_n	2.0	2.0
σ_t	-0.2	2.0
σ_r	$-2t/R$	0
S	2.2	2.0

XIII-2124 Limitations of Indices of XIII-2123

The indices of XIII-2123 apply provided the following conditions set forth in (a) through (g) exist.

(a) 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, an estimate of the σ_n index on the inside may be obtained from one of the following equations, provided $d/D \leq 0.15$.

For hillside connections in spheres or cylinders,

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

For lateral connections in cylinders,

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

where

K_1 = the σ_n inside stress index of XIII-2123 for a radial connection

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

(b) The arc distance measured between the center lines 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 two times the sum of their radii for opening along the circumference of a cylindrical shell. See NC-3231.1(a)(3) when two nozzles are not on a longitudinal or circumferential line.

Table XIII-2123-2
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	$-t/R$	0	$-t/R$	0
S	3.3	1.2	1.2	2.6

(c) The following dimensional ratios are met:

Ratio	Cylinder	Sphere
D/T	10 to 100	10 to 100
d/D	0.5 max.	0.5 max.
d/\sqrt{Dt}	...	0.8 max.
$d/\sqrt{Dt_r r_2/t}$	1.5 max.	...

where D is the inside shell diameter, t is the shell thickness, and d is the inside nozzle diameter. 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 Figure NC-3332.2-1 or WC-3232.2-1 unless a tapered transition section is incorporated into the reinforcement and the shell, meeting the requirements of NC-3261 or WC-3261.

(d) In the case of spherical shells and formed heads, at least 40% of the total nozzle reinforcement area shall be located beyond the outside surface of the minimum required vessel wall thickness.

(e) The inside corner radius r_1 [Figure XIII-2124(e)-1] is between 10% and 100% of the shell thickness t .

(f) The outer corner radius r_2 [Figure XIII-2124(e)-1] is large enough to provide a smooth transition between the nozzles and the shell. In addition, for opening diameters in cylindrical shells and 2:1 ellipsoidal heads greater than $1\frac{1}{2}$ shell thicknesses and in spherical shells greater than 3 shell thicknesses, the value of r_2 shall be not less than one-half the thickness of the shell or nozzle wall, whichever is greater.

(g) The radius r_3 [Figure XIII-2124(e)-1] is not less than the greater of (1) or (2) below

(1) $0.002\theta d_o$, where d_o is the outside diameter of the nozzle and is as shown in Figure XIII-2124(e)-1 and the angle θ is expressed in degrees

(2) $2(\sin \theta)^3$ times offset for the configuration shown in Figure XIII-2124(e)-1 sketches (a) and (b)

XIII-2130 EXPERIMENTAL STRESS ANALYSIS METHOD

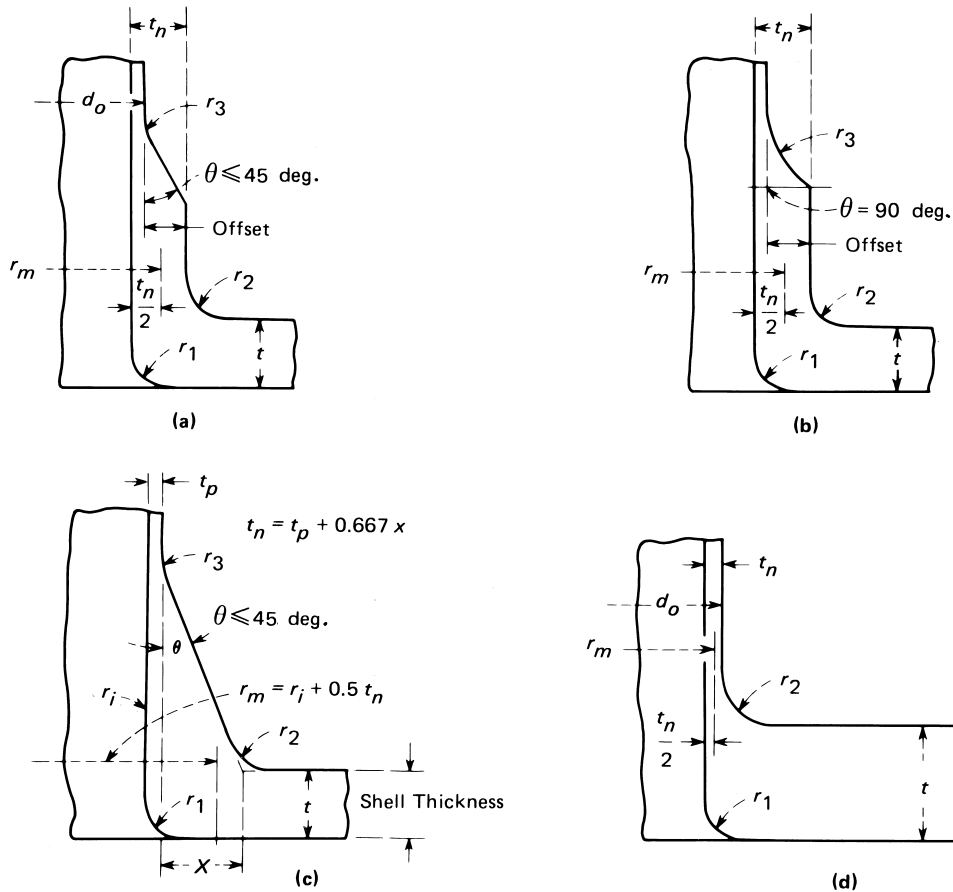
XIII-2131 Determination of Stress Intensities

The stress intensities for opening configurations which do not meet the requirements of NC-3230, WC-3230, or of XIII-2124 shall be determined in accordance with methods of Mandatory Appendix II.

XIII-2132 Evaluation of Data for Similar But Slightly Different Configurations

In accordance with II-1112, reevaluation is not required for configurations for which there are available detailed experimental results that are consistent with the requirements of Mandatory Appendix II. In order that available experimental data may be interpreted as providing information pertinent to the analysis of slightly different

Figure XIII-2124(e)-1
Nozzle Nomenclature and Dimensions



configurations, thereby possibly minimizing the need for additional investigations, the following guidelines of (a) through (d) are presented:

(a) For an unreinforced opening or for an opening where the reinforcement is provided primarily by a uniform increase in vessel wall thickness, the stresses around the opening will increase with increasing d/D ratio of the opening (diameter of nozzle or opening to diameter of shell). Therefore, experimental data for a small d/D ratio cannot be safely applied to a larger d/D ratio but can be applied to a smaller d/D ratio provided the experiments were made at a d/D ratio < 0.5 .

(b) For an unreinforced opening or for an opening where the reinforcement is provided primarily by a uniform increase in vessel wall thickness, the stresses around the opening will increase with increasing d/t ratio for thinner shells. Therefore, experimental data for a relatively small d/t ratio cannot be safely applied to a larger d/t ratio but can be applied to a smaller d/t ratio.

(c) Generally speaking, the stress data available in the literature are applicable only to single openings. Such data should be considered valid only for a connection sufficiently removed from another nozzle, opening, flange, or other major discontinuity so that superposition of stresses will not produce an unacceptable value of stress intensity.

(d) Stresses at the outside juncture of a nozzle and shell are greatly influenced by the fillet or transition at the juncture. Generally speaking, stress data available in the literature are for certain specific fillet radii. Other factors being equal, these stress data may be considered valid for fillet radii equal to or greater than used in the test but should not be considered valid for smaller fillet radii or undefined fillets and transitions, such as for a triangular weld fillet, as commonly used.

MANDATORY APPENDIX XIV

ARTICLE XIV-1000 DESIGN BASED ON FATIGUE ANALYSIS

XIV-1100 INTRODUCTION

XIV-1110 SCOPE

This Appendix is applicable only to the fatigue analysis of Class 2 vessels meeting the requirements of NC-3200 and containments meeting the requirements of WC-3200.

XIV-1200 ANALYSIS FOR CYCLIC SERVICE OF VESSELS

XIV-1210 GENERAL REQUIREMENTS

XIV-1211 Suitability of Vessel for Cyclic Service

The suitability of a vessel component for specified Service Conditions involving cyclic application of loads and thermal conditions shall be determined by the methods described in this Appendix, except that the suitability of high strength bolts shall be determined by the methods of [XIV-1300](#) and the possibility of thermal stress ratchet shall be investigated in accordance with [XIV-1400](#). If the specified service of the vessel meets all of the conditions of NC-3219 or WC-3219, no analysis for cyclic service is required and it may be assumed that the limits on peak stress intensities as governed by fatigue have been satisfied by compliance with the applicable requirements for materials, design, fabrication, examination, and testing of Subsections NC or WC. If the service does not meet all the conditions of NC-3219 or WC-3219, a fatigue analysis shall be made in accordance with [XIV-1220](#) or a fatigue test shall be made in accordance with II-1150.

XIV-1212 Allowable Amplitude of Alternating Stresses

The conditions and procedures of NC-3219 or WC-3219, and [XIV-1220](#) are based on a comparison of peak stresses with strain cycling fatigue data. The strain cycling fatigue data are represented by the design fatigue strength curves of Figures I-9.0.¹¹ These curves show the allowable amplitude S_a of the alternating stress component (one-half of the alternating stress range) plotted against the number of cycles. The stress amplitude is calculated on the assumption of elastic behavior and hence has the dimensions of stress, but it does not represent a real stress

when the elastic range is exceeded. The fatigue curves are obtained from uniaxial strain cycling data in which the imposed strains have been multiplied by the elastic modulus and a design margin has been provided, so as to make the calculated stress intensity and amplitude and the allowable stress amplitude directly comparable. The curves have been adjusted, where necessary, to include the maximum effects of mean stress, which is the condition where the stress fluctuates about a mean value which is different from zero. As a consequence of this procedure, it is essential that the requirements of [XIII-1145](#) and [XIII-1160](#) be satisfied at all times, with transient stresses included and that the calculated value of the alternating stress intensity be proportional to the actual strain amplitude. To evaluate the effect of alternating stresses of varying amplitudes, a linear damage relation is assumed in [XIV-1220](#).

XIV-1213 Loadings to Be Considered

The loadings to be considered shall include those loads that are due to testing of the vessel when such testing is in addition to that required by NC-6100 or WC-6000.

XIV-1220 CYCLIC LOADING

XIV-1221 Design for Cyclic Loading

XIV-1221.1 Determination of Vessel's Ability to Withstand Cyclic Loading. If the specific service of the vessel does not meet the conditions of NC-3219 or WC-3219, the ability of the vessel to withstand the specified cyclic service without fatigue failure shall be determined as provided herein. The determination shall be made on the basis of the stresses at a point of the vessel and the allowable stress cycles shall be adequate for the specified service at every point. Only the stresses due to the specified service cycle need be considered; stresses produced by any load or thermal condition which does not vary during the cycle need not be considered, since they are mean stresses and the maximum possible effect of mean stress is included in the fatigue design curves.

XIV-1221.2 Significance of Compliance With Requirements for Cyclic Loading. Compliance with these requirements means only that the vessel is suitable from the

standpoint of possible fatigue failure; complete suitability for the specified service is also dependent on meeting the general stress limits of XIII-1140 and any applicable special stress limits of XIV-1410.

XIV-1221.3 Cyclic Loading Design Procedure.

(a) *When Principal Stress Direction Does Not Change.* For any case in which the directions of the principal stresses at the point being considered do not change during the cycle, steps (1), (2), and (3) shall be followed to determine the alternating stress intensity.

(1) *Principal Stresses.* Consider the values of the three principal stresses at the point versus time for the complete stress cycle, taking into account both the gross and local structural discontinuities and the thermal effects which vary during the cycle. These are designated as σ_1 , σ_2 , and σ_3 for later identification.

(2) *Stress Differences.* Determine the stress differences $S_{12} = \sigma_1 - \sigma_2$, $S_{23} = \sigma_2 - \sigma_3$, and $S_{31} = \sigma_3 - \sigma_1$ versus time for the complete cycle. In what follows, the symbol S_{ij} is used to represent any one of these three stress differences.

(3) *Alternating Stress Intensity.* Determine the extremes of the range through which each stress difference S_{ij} fluctuates, and find the absolute magnitude of this range for each S_{ij} . Call this magnitude S_{rij} and let $S_{alt\ ij} = 0.5S_{rij}$. The alternating stress intensity S_{alt} is the largest of the $S_{alt\ ij}$ values.

(b) *When Principal Stress Direction Changes.* For any case in which the directions of the principal stresses at the point being considered change during the stress cycle, it is necessary to use steps (1) through (5).

(1) Consider the values of the six stress components σ_t , σ_l , σ_r , τ_{lt} , τ_{lr} , and τ_{rt} versus time for the complete stress cycle, taking into account both the gross and local structural discontinuities and the thermal effects which vary during the cycle.

(2) Choose a point in time when the conditions are one of the extremes for the cycle, either maximum or minimum algebraically, and identify the stress components at this time by the subscript i . In most cases it will be possible to choose at least one time during the cycle when the conditions are known to be extreme. In some cases it may be necessary to try different points in time to find the one which results in the largest value of alternating stress intensity.

(3) Subtract each of the six stress components σ_{ti} , σ_{li} , etc., from the corresponding stress components σ_t , σ_l , etc., at each point in time during the cycle and call the resulting components σ'_{ti} , σ'_{li} , etc.

(4) At each point in time during the cycle, calculate the principal stresses σ'_1 , σ'_2 , and σ'_3 derived from the six stress components σ'_{ti} , σ'_{li} , etc. Note that the directions of the principal stresses may change during the cycle but each principal stress retains its identity as it rotates.

(5) Determine the stress differences $S'_{12} = \sigma'_1 - \sigma'_2$, $S'_{23} = \sigma'_2 - \sigma'_3$, and $S'_{31} = \sigma'_3 - \sigma'_1$ versus time for the complete cycle and find the largest absolute magnitude of any stress difference at any time. The alternating stress intensity S_{alt} is one-half of this magnitude.

(c) *Design Fatigue Curves.* Figures I-9.1 through I-9.6 contain the applicable fatigue design curves for some of the materials permitted by Subsection NC or Subsection WC. When more than one curve is presented for a given material, the applicability of each curve to materials of various strength levels is identified. Linear interpolation may be used for intermediate strength levels of these materials. As used herein, the strength level is the specified minimum room temperature value.

(d) *Use of Design Fatigue Curve.* Multiply S_{alt} , as determined in (a) or (b), by the ratio of the modulus of elasticity given on the design fatigue curve to the value used in the analysis. Enter the applicable design fatigue curve at this value on the ordinate axis and find the corresponding number of cycles on the abscissa. If the operational cycle being considered is the only one which produces significant fluctuating stresses, this is the allowable number of cycles.

(e) *Cumulative Damage.* If there are two or more types of stress cycles which produce significant stresses, their cumulative effect shall be evaluated as given in (1) through (6).

(1) Designate the specified number of times each type of stress cycle of types 1, 2, 3, ..., n will be repeated during the life of the vessel as n_1 , n_2 , n_3 , ..., n_n , respectively. In determining n_1 , n_2 , n_3 , ..., n_n , consideration shall be given to the superposition of cycles of various origins which produce a total stress difference range greater than the stress difference ranges of the individual cycles. For example, if one type of stress cycle produces 1000 cycles of a stress difference variation from 0 to +60.0 ksi and another type of stress cycle produces 10,000 cycles of a stress difference variation from 0 to -50.0 ksi, the two types of cycle to be considered are defined by the following parameters.

For type 1 cycle:

$$n_1 = 1000$$

$$S_{alt\ 1} = (60,000 + 50,000) / 2 = 55,000 \text{ psi}$$

For type 2 cycle:

$$n_2 = 9000$$

$$S_{alt\ 2} = (50,000 + 0) / 2 = 25,000 \text{ psi}$$

(2) For each type of stress cycle determine the alternating stress intensity S_{alt} by the procedures of the preceding (a) or (b). Call these quantities $S_{alt\ 1}$, $S_{alt\ 2}$, $S_{alt\ 3}$, ..., $S_{alt\ n}$.

(3) For each value $S_{alt\ 1}, S_{alt\ 2}, S_{alt\ 3}, \dots, S_{alt\ n}$, use the applicable design fatigue curve to determine the maximum number of repetitions which would be allowable if this type of cycle were the only one acting. Call these values $N_1, N_2, N_3, \dots, N_n$.

(4) For each type of stress cycle, calculate the usage factors $U_1, U_2, U_3, \dots, U_n$ from $U_1 = n_1/N_1, U_2 = n_2/N_2, U_3 = n_3/N_3, \dots, U_n = n_n/N_n$.

(5) Calculate the cumulative usage factor U from $U = U_1 + U_2 + U_3 + \dots + U_n$.

(6) The cumulative usage factor U shall not exceed 1.0.

XIV-1222 Local Structural Discontinuities

These effects shall be evaluated for all conditions using stress concentration factors determined from theoretical, experimental, or photoelastic studies, or numerical stress analysis techniques. Experimentally determined fatigue strength reduction factors may be used when determined in accordance with the procedures of II-1600, except for high strength alloy steel bolting for which the requirements of XIV-1322 shall apply when using the design fatigue curve of Figure I-9.4. Except for the case of crack-like defects, no fatigue strength factor greater than five need be used.

XIV-1223 Fillet Welds

Fillet welds shall not be used in vessels for joints of Category A, B, C, or D (Figure NC-3351-1 or Figure WC-3251-1) except as permitted for joints of Category C for slip on flanges (NC-3252.3 and NC-3262.2) and for the joints of Category D as permitted in NC-3259 or WC-3259. Fillet welds may be used for attachment to pressure vessels using one-half the stress limits of XIII-1142 through XIII-1145 for primary and secondary stresses. Evaluation for cyclic loading shall be made in accordance with Mandatory Appendix XIV using a fatigue strength reduction factor of four and shall include consideration of temperature differences between the vessel and the attachment and expansion or contraction of the vessel produced by internal or external pressure.

XIV-1300 ANALYSIS FOR CYCLIC SERVICE OF BOLTS

XIV-1310 GENERAL REQUIREMENTS

XIV-1311 Suitability of Bolts for Cyclic Service

Unless the vessel on which they are installed meets all the conditions of NC-3219 or WC-3219 and thus requires no fatigue analysis, the suitability of bolts for cyclic service shall be determined in accordance with the procedures of XIV-1320.

XIV-1320 EVALUATION OF BOLTS DEPENDENT ON THEIR MATERIAL PROPERTIES

XIV-1321 Bolts With Tensile Strengths Less Than 100 ksi (689 MPa)

Bolts made of materials which have minimum specified tensile strengths of less than 100.0 ksi (689 MPa) shall be evaluated for cyclic operation by the methods of XIV-1220, using the applicable design fatigue curve, Figure I-9.1 or Figure I-9.2 and an appropriate stress concentration factor (XIV-1324).

XIV-1322 High Strength Bolts

High strength alloy steel bolts and studs may be evaluated for cyclic operation by the methods of XIV-1220, using the design fatigue curve of Figure I-9.4, provided the following requirements of (a) through (e) are met:

(a) The material is one of the following: SA-193 Grade B-7 or B-16; SA-320 Grade L-43; SA-540 Grades B-23 and B-24, heat treated in accordance with SA-540, Section 5.

(b) The maximum value of the service stress at the periphery of the bolt cross section resulting from direct tension plus bending and neglecting stress concentrations shall not exceed $2.7S_m$, if the higher of the two fatigue design curves given in Figure I-9.4 is used. The $2.0S_m$ limit for direct tension is unchanged.

(c) Threads shall be of a Vee type, having a minimum thread root radius no smaller than 0.003 in. (0.076 mm).

(d) Fillet radii at the end of the shank shall be such that the ratio of fillet radius to shank diameter is not less than 0.060.

(e) The fatigue strength reduction factor used in the fatigue evaluation shall not be less than 4.0.

XIV-1323 Acceptability for Cyclic Service

The bolts shall be acceptable for the specific cyclic application of loads and thermal stresses provided the cumulative usage factor U , as determined in XIV-1221.3(e), does not exceed 1.0.

XIV-1324 Fatigue Strength Reduction Factor for Threads

Unless it can be shown by analysis or test that a lower value is appropriate, the fatigue strength reduction factor used in the fatigue evaluation of threaded members shall not be less than 4.0.

XIV-1400 ANALYSIS FOR THERMAL STRESS RATCHET

XIV-1410 THERMAL STRESS RATCHET IN SHELL

Under certain combinations of steady state and cyclic loadings there is a possibility of large distortions developing as the result of ratchet action, that is, the deformation

increases by a nearly equal amount for each cycle. Examples of this phenomenon are treated in [XIV-1410](#) and in [XIV-1420](#).

(a) The limiting value of the maximum cyclic thermal stress permitted in a shell loaded by steady state internal pressure in order to prevent cyclic growth in diameter is as follows. Let

- y' = maximum allowable thermal stress, computed on an elastic basis, divided by the yield strength¹² S_y
 x = maximum general membrane stress due to pressure divided by the yield strength¹² S_y

(1) Case 1: Linear variation of temperature through the shell wall

$$y' = 1/x \text{ for } 0 < x < 0.5$$

$$y' = 4(1 - x) \text{ for } 0.5 < x < 1.0$$

(2) Case 2: Parabolic constantly increasing or constantly decreasing variation of temperature through the shell wall

$$y' = 5.2(1 - x) \text{ for } 0.615 < x < 1.0$$

and approximately, for $x < 0.615$, $y' = 4.65$, 3.55 , and 2.70 for $x = 0.3$, 0.4 , and 0.5 , respectively.

(b) Use of the yield strength S_y in the above relations instead of the proportional limit allows a small amount of growth during each cycle until strain hardening raises

the proportional limit to S_y . If the yield strength of the material is higher than 2 times the S_a value for the maximum number of cycles on the applicable fatigue curve of Figures I-9.0 for the material, the latter value shall be used, if there are to be a large number of cycles, because strain softening may occur.

XIV-1420 PROGRESSIVE DISTORTION OF NONINTEGRAL CONNECTIONS

Screwed on caps, screwed in plugs, shear ring closures, and breech lock closures are examples of nonintegral connections which are subject to failure by bell mouthing or other types of progressive deformation. If any combination of applied loads produces yielding, such joints are subject to ratcheting because the mating members may become loose at the end of each complete operating cycle and start the next cycle in a new relationship with each other, with or without manual manipulation. Additional distortion may occur in each cycle so that interlocking parts, such as threads, can eventually lose engagement. Therefore, primary plus secondary stress intensities ([XIII-1145](#)) which result in slippage between the parts of a nonintegral connection in which disengagement could occur as a result of progressive distortion shall be limited to the allowable stress limits given in [XIII-1142](#) and [XIII-1143](#).

MANDATORY APPENDIX XVIII

ARTICLE XVIII-1000 CAPACITY CONVERSIONS FOR PRESSURE RELIEF VALVES

XVIII-1100 PROCEDURE FOR CONVERSION

For air:

XVIII-1110 EQUATIONS FOR CONVERSION (COMPRESSIBLE FLUIDS)

$$W_a = CKAP\sqrt{M/T} \quad (2)$$

The capacity of a relief valve in terms of a gas or vapor other than the medium for which the valve was officially rated shall be determined by application of the following equations:¹³

For dry saturated steam:

pressures up to 1,500 psig (10.3 MPa)

$$W_s = C_N K A P \quad (1)$$

where

$$C_N = 51.5 \text{ (5.25)}$$

Pressures over 1,500 psig (10.3 MPa) and up to 3,200 psig (22 MPa), the value of W_s , calculated by the above equation, shall be multiplied by the following factor, F_N .

(U.S. Customary Units)

$$F_N = \frac{0.1906P - 1,000}{0.2232P - 1,061}$$

(SI Units)

$$F_N = \frac{27.6P - 1,000}{33.2P - 1,061}$$

For superheated steam:

For superheated steam the value of W_s shall be multiplied by the appropriate superheat correction factor, K_{sh} , of Table XVIII-1110-1 (or Table XVIII-1110-1M for SI calculations).

For wet saturated steam:

For wet saturated steam with a quality (dryness fraction) of 0.90 or greater, the value of W_s shall be corrected by dividing by the quality of the steam used for testing.

where

$$C = 356 \text{ (27.03)}$$

$$M = 28.97 \text{ mol. wt.}$$

$$T = 520 \text{ (288) when } W_a \text{ is the rated capacity}$$

For any gas or vapor (other than steam):

$$W = CKAP\sqrt{M/T} \quad (3)$$

where

A = actual discharge area of the pressure relief valve, in.² (mm²)

C = constant for gas or vapor which is a function of the ratio of specific heats, $k = c_p/c_v$ (Figure XVIII-1110-1)

K = coefficient of discharge

M = molecular weight

P = (set pressure + overpressure) plus atmospheric pressure, psia (MPa_{abs.})

T = absolute temperature at inlet (°F plus 460) (K)

W = flow of any gas or vapor, lb/hr (kg/h)

W_a = rated capacity,¹⁴ converted to lb/hr (kg/h) of air at 60°F (15°C) inlet temperature

W_s = rated capacity, lb/hr (kg/h) of steam

Z = ratio of deviation of the actual gas from a perfect gas, a ratio evaluated at inlet conditions

These equations shall also be used when the required flow of any gas or vapor is known and it is necessary to compute the rated capacity of steam or air.

(a) Molecular weight of some of the common gases and vapors are given in Table XVIII-1110(a)-1.

(b) In the case of hydrocarbons, the compressibility factor Z shall be included in the equation for gases and vapors as follows:

$$W = CKAP\sqrt{M/ZT} \quad (4)$$

2013 SECTION III APPENDICES

**Table XVIII-1110-1
Superheat Correction Factor K_{sh}**

Flowing Pressure (psia)	Superheat Correction Factor K_{sh} , Total Temperature, °F, of Superheated Steam																
	400	450	500	550	600	650	700	750	800	850	900	950	1000	1050	1100	1150	1200
50	0.987	0.957	0.930	0.905	0.882	0.861	0.841	0.823	0.805	0.789	0.774	0.759	0.745	0.732	0.719	0.708	0.696
100	0.998	0.963	0.935	0.909	0.885	0.864	0.843	0.825	0.807	0.790	0.775	0.760	0.746	0.733	0.720	0.708	0.697
150	0.984	0.970	0.940	0.913	0.888	0.866	0.846	0.826	0.808	0.792	0.776	0.761	0.747	0.733	0.721	0.709	0.697
200	0.979	0.977	0.945	0.917	0.892	0.869	0.848	0.828	0.810	0.793	0.777	0.762	0.748	0.734	0.721	0.709	0.698
250	...	0.972	0.951	0.921	0.895	0.871	0.850	0.830	0.812	0.794	0.778	0.763	0.749	0.735	0.722	0.710	0.698
300	...	0.968	0.957	0.926	0.898	0.874	0.852	0.832	0.813	0.796	0.780	0.764	0.750	0.736	0.723	0.710	0.699
350	...	0.968	0.963	0.930	0.902	0.877	0.854	0.834	0.815	0.797	0.781	0.765	0.750	0.736	0.723	0.711	0.699
400	0.963	0.935	0.906	0.880	0.857	0.836	0.816	0.798	0.782	0.766	0.751	0.737	0.724	0.712	0.700
450	0.961	0.940	0.909	0.883	0.859	0.838	0.818	0.800	0.783	0.767	0.752	0.738	0.725	0.712	0.700
500	0.961	0.946	0.914	0.886	0.862	0.840	0.820	0.801	0.784	0.768	0.753	0.739	0.725	0.713	0.701
550	0.962	0.952	0.918	0.889	0.864	0.842	0.822	0.803	0.785	0.769	0.754	0.740	0.726	0.713	0.701
600	0.964	0.958	0.922	0.892	0.867	0.844	0.823	0.804	0.787	0.770	0.755	0.740	0.727	0.714	0.702
650	0.968	0.958	0.927	0.896	0.869	0.846	0.825	0.806	0.788	0.771	0.756	0.741	0.728	0.715	0.702
700	0.958	0.931	0.899	0.872	0.848	0.827	0.807	0.789	0.772	0.757	0.742	0.728	0.715	0.703
750	0.958	0.936	0.903	0.875	0.850	0.828	0.809	0.790	0.774	0.758	0.743	0.729	0.716	0.703
800	0.960	0.942	0.906	0.878	0.852	0.830	0.810	0.792	0.774	0.759	0.744	0.730	0.716	0.704
850	0.962	0.947	0.910	0.880	0.855	0.832	0.812	0.793	0.776	0.760	0.744	0.730	0.717	0.704
900	0.965	0.953	0.914	0.883	0.857	0.834	0.813	0.794	0.777	0.760	0.745	0.731	0.718	0.705
950	0.969	0.958	0.918	0.886	0.860	0.836	0.815	0.796	0.778	0.761	0.746	0.732	0.718	0.705
1,000	0.974	0.959	0.923	0.890	0.862	0.838	0.816	0.797	0.779	0.762	0.747	0.732	0.719	0.706
1,050	0.960	0.927	0.893	0.864	0.840	0.818	0.798	0.780	0.763	0.748	0.733	0.719	0.707
1,100	0.962	0.931	0.896	0.867	0.842	0.820	0.800	0.781	0.764	0.749	0.734	0.720	0.707
1,150	0.964	0.936	0.899	0.870	0.844	0.821	0.801	0.782	0.765	0.749	0.735	0.721	0.708
1,200	0.966	0.941	0.903	0.872	0.846	0.823	0.802	0.784	0.766	0.750	0.735	0.721	0.708
1,250	0.969	0.946	0.906	0.875	0.848	0.825	0.804	0.785	0.767	0.751	0.736	0.722	0.709
1,300	0.973	0.952	0.910	0.878	0.850	0.826	0.805	0.786	0.768	0.752	0.737	0.723	0.709
1,350	0.977	0.958	0.914	0.880	0.852	0.828	0.807	0.787	0.769	0.753	0.737	0.723	0.710
1,400	0.982	0.963	0.918	0.883	0.854	0.830	0.808	0.788	0.770	0.754	0.738	0.724	0.710
1,450	0.987	0.968	0.922	0.886	0.857	0.832	0.809	0.790	0.771	0.754	0.739	0.724	0.711
1,500	0.993	0.970	0.926	0.889	0.859	0.833	0.811	0.791	0.772	0.755	0.740	0.725	0.711
1,550	0.972	0.930	0.892	0.861	0.835	0.812	0.792	0.773	0.756	0.740	0.726	0.712
1,600	0.973	0.934	0.894	0.863	0.836	0.813	0.792	0.774	0.756	0.740	0.726	0.712
1,650	0.973	0.936	0.895	0.863	0.836	0.812	0.791	0.772	0.755	0.739	0.724	0.710
1,700	0.973	0.938	0.895	0.863	0.835	0.811	0.790	0.771	0.754	0.738	0.723	0.709
1,750	0.974	0.940	0.896	0.862	0.835	0.810	0.789	0.770	0.752	0.736	0.721	0.707
1,800	0.975	0.942	0.897	0.862	0.834	0.810	0.788	0.768	0.751	0.735	0.720	0.705
1,850	0.976	0.944	0.897	0.862	0.833	0.809	0.787	0.767	0.749	0.733	0.718	0.704
1,900	0.977	0.946	0.898	0.862	0.832	0.807	0.785	0.766	0.748	0.731	0.716	0.702
1,950	0.979	0.949	0.898	0.861	0.832	0.806	0.784	0.764	0.746	0.729	0.714	0.700
2,000	0.982	0.952	0.899	0.861	0.831	0.805	0.782	0.762	0.744	0.728	0.712	0.698
2,050	0.985	0.954	0.899	0.860	0.830	0.804	0.781	0.761	0.742	0.726	0.710	0.696
2,100	0.988	0.956	0.900	0.860	0.828	0.802	0.779	0.759	0.740	0.724	0.708	0.694
2,150	0.956	0.900	0.859	0.827	0.801	0.778	0.757	0.738	0.722	0.706	0.692
2,200	0.955	0.901	0.859	0.826	0.799	0.776	0.755	0.736	0.720	0.704	0.690
2,250	0.954	0.901	0.858	0.825	0.797	0.774	0.753	0.734	0.717	0.702	0.687
2,300	0.953	0.901	0.857	0.823	0.795	0.772	0.751	0.732	0.715	0.699	0.685
2,350	0.952	0.902	0.856	0.822	0.794	0.769	0.748	0.729	0.712	0.697	0.682
2,400	0.952	0.902	0.855	0.820	0.791	0.767	0.746	0.727	0.710	0.694	0.679
2,450	0.951	0.902	0.854	0.818	0.789	0.765	0.743	0.724	0.707	0.691	0.677
2,500	0.951	0.902	0.852	0.816	0.787	0.762	0.740	0.721	0.704	0.688	0.674
2,550	0.951	0.902	0.851	0.814	0.784	0.759	0.738	0.718	0.701	0.685	0.671
2,600	0.951	0.903	0.849	0.812	0.782	0.756	0.735	0.715	0.698	0.682	0.664
2,650	0.952	0.903	0.848	0.809	0.779	0.754	0.731	0.712	0.695	0.679	0.664
2,700	0.952	0.903	0.846	0.807	0.776	0.750	0.728	0.708	0.691	0.675	0.661
2,750	0.953	0.903	0.844	0.804	0.773	0.747	0.724	0.705	0.687	0.671	0.657
2,800	0.956	0.903	0.842	0.801	0.769	0.743	0.721	0.701	0.684	0.668	0.653

Table XVIII-1110-1
Superheat Correction Factor K_{sh} (Cont'd)

Flowing Pressure (psia)	Superheat Correction Factor K_{sh} , Total Temperature, °F, of Superheated Steam																
	400	450	500	550	600	650	700	750	800	850	900	950	1000	1050	1100	1150	1200
2,850	0.959	0.902	0.839	0.798	0.766	0.739	0.717	0.697	0.679	0.663	0.649
2,900	0.963	0.902	0.836	0.794	0.762	0.735	0.713	0.693	0.675	0.659	0.645
2,950	0.902	0.834	0.790	0.758	0.731	0.708	0.688	0.671	0.655	0.640
3,000	0.901	0.831	0.786	0.753	0.726	0.704	0.684	0.666	0.650	0.635
3,050	0.899	0.827	0.782	0.749	0.722	0.699	0.679	0.661	0.645	0.630
3,100	0.896	0.823	0.777	0.744	0.716	0.693	0.673	0.656	0.640	0.625
3,150	0.894	0.819	0.772	0.738	0.711	0.688	0.668	0.650	0.634	0.620
3,200	0.889	0.815	0.767	0.733	0.705	0.682	0.662	0.644	0.628	0.614

XVIII-1120 EXAMPLES**XVIII-1121 Example 1**

Given: A pressure relief valve bears a rated capacity of 3,020 lb/hr of saturated steam for a pressure setting of 200 psi.

Problem: What is the relieving capacity of that valve in terms of air at 100°F for the same pressure setting?

Solution:

(a) For steam:

$$W_s = 51.5KAP$$

$$3,020 = 51.5KAP$$

$$KAP = 3,020 / 51.5 = 58.6$$

(b) For air:

$$\begin{aligned} W_a &= CKAP \sqrt{\frac{M}{T}} \\ &= 356 KAP \sqrt{\frac{28.97}{460 + 100}} \\ &= (356)(58.6) \sqrt{\frac{28.97}{560}} \\ &= 4,745 \text{ lb / hr} \end{aligned}$$

XVIII-1222 Example 2

Given: It is required to relieve 5,000 lb/hr of propane from a pressure vessel through a pressure relief valve set to relieve at a pressure of P_s , psi, and with an inlet temperature of 125°F.

Problem: What total capacity in lb/hr of steam in pressure relief valves must be furnished?

Solution:

(a) For propane:

$$W = CKAP \sqrt{\frac{M}{T}}$$

Value of C is not definitely known. Use the conservative value, $C = 315$:

$$\begin{aligned} 5,000 &= 315 KAP \sqrt{\frac{44.09}{460 + 125}} \\ KAP &= 57.7 \end{aligned}$$

(b) For steam:

$$\begin{aligned} W_s &= 51.5 KAP \\ &= (51.5)(57.7) \\ &= 2,970 \text{ lb / hr set to relieve at } P_s, \text{ psi} \end{aligned}$$

XVIII-1123 Example 3

Given: It is required to relieve 1,000 lb/hr of ammonia from a pressure vessel at 150°F.

Problem: What is the required total capacity in lb/hr of steam at the same pressure setting?

Solution:

(a) For ammonia:

$$W = CKAP \sqrt{M/T}$$

Manufacturer and Owner agree to use $k = 1.33$. From Figure XVIII-1110-1, $C = 350$:

$$\begin{aligned} 1,000 &= 350 KAP \sqrt{\frac{17.03}{460 + 150}} \\ KAP &= 17.10 \end{aligned}$$

(b) For steam:

$$\begin{aligned} W_s &= 51.5 KAP \\ &= (51.5)(17.10) \\ &= 880 \text{ lb / hr} \end{aligned}$$

2013 SECTION III APPENDICES

Table XVIII-1110-1M
Superheat Correction Factor K_{sh}

Flowing Pressure (MPa)	Superheat Correction Factor K_{sh} Total Temperature, °C, of Superheated Steam																	
	205	225	250	275	300	325	350	375	400	425	450	475	500	525	550	575	600	625
0.50	0.991	0.968	0.942	0.919	0.896	0.876	0.857	0.839	0.823	0.807	0.792	0.778	0.765	0.752	0.74	0.728	0.717	0.706
0.75	0.995	0.972	0.946	0.922	0.899	0.878	0.859	0.841	0.824	0.808	0.793	0.779	0.766	0.753	0.74	0.729	0.717	0.707
1.00	0.985	0.973	0.95	0.925	0.902	0.88	0.861	0.843	0.825	0.809	0.794	0.78	0.766	0.753	0.741	0.729	0.718	0.707
1.25	0.981	0.976	0.954	0.928	0.905	0.883	0.863	0.844	0.827	0.81	0.795	0.781	0.767	0.754	0.741	0.729	0.718	0.707
1.50	0.957	0.932	0.907	0.885	0.865	0.846	0.828	0.812	0.796	0.782	0.768	0.755	0.742	0.73	0.718	0.708
1.75	0.959	0.935	0.91	0.887	0.866	0.847	0.829	0.813	0.797	0.782	0.769	0.756	0.743	0.731	0.719	0.708
2.00	0.96	0.939	0.913	0.889	0.868	0.849	0.831	0.814	0.798	0.784	0.769	0.756	0.744	0.731	0.72	0.708
2.25	0.963	0.943	0.916	0.892	0.87	0.85	0.832	0.815	0.799	0.785	0.77	0.757	0.744	0.732	0.72	0.709
2.50	0.946	0.919	0.894	0.872	0.852	0.834	0.816	0.8	0.785	0.771	0.757	0.744	0.732	0.72	0.71
2.75	0.948	0.922	0.897	0.874	0.854	0.835	0.817	0.801	0.786	0.772	0.758	0.745	0.733	0.721	0.71
3.00	0.949	0.925	0.899	0.876	0.855	0.837	0.819	0.802	0.787	0.772	0.759	0.746	0.733	0.722	0.71
3.25	0.951	0.929	0.902	0.879	0.857	0.838	0.82	0.803	0.788	0.773	0.759	0.746	0.734	0.722	0.711
3.50	0.953	0.933	0.905	0.881	0.859	0.84	0.822	0.804	0.789	0.774	0.76	0.747	0.734	0.722	0.711
3.75	0.956	0.936	0.908	0.883	0.861	0.841	0.823	0.806	0.79	0.775	0.761	0.748	0.735	0.723	0.711
4.00	0.959	0.94	0.91	0.885	0.863	0.842	0.824	0.807	0.791	0.776	0.762	0.748	0.735	0.723	0.712
4.25	0.961	0.943	0.913	0.887	0.864	0.844	0.825	0.808	0.792	0.776	0.762	0.749	0.736	0.724	0.713
4.50	0.944	0.917	0.89	0.866	0.845	0.826	0.809	0.793	0.777	0.763	0.749	0.737	0.725	0.713
4.75	0.946	0.919	0.892	0.868	0.847	0.828	0.81	0.793	0.778	0.764	0.75	0.737	0.725	0.713
5.00	0.947	0.922	0.894	0.87	0.848	0.829	0.811	0.794	0.779	0.765	0.751	0.738	0.725	0.714
5.25	0.949	0.926	0.897	0.872	0.85	0.83	0.812	0.795	0.78	0.765	0.752	0.738	0.726	0.714
5.50	0.952	0.93	0.899	0.874	0.851	0.831	0.813	0.797	0.78	0.766	0.752	0.739	0.727	0.714
5.75	0.954	0.933	0.902	0.876	0.853	0.833	0.815	0.798	0.782	0.767	0.753	0.739	0.727	0.715
6.00	0.957	0.937	0.904	0.878	0.855	0.834	0.816	0.798	0.783	0.768	0.753	0.74	0.727	0.716
6.25	0.96	0.94	0.907	0.88	0.856	0.836	0.817	0.799	0.783	0.768	0.754	0.74	0.728	0.716
6.50	0.964	0.944	0.91	0.882	0.859	0.837	0.818	0.801	0.784	0.769	0.754	0.741	0.729	0.716
6.75	0.966	0.946	0.913	0.885	0.86	0.839	0.819	0.802	0.785	0.769	0.755	0.742	0.729	0.717
7.00	0.947	0.916	0.887	0.862	0.84	0.82	0.802	0.786	0.77	0.756	0.742	0.729	0.717
7.25	0.949	0.919	0.889	0.863	0.842	0.822	0.803	0.787	0.771	0.756	0.743	0.73	0.717
7.50	0.951	0.922	0.891	0.865	0.843	0.823	0.805	0.788	0.772	0.757	0.744	0.73	0.718
7.75	0.953	0.925	0.893	0.867	0.844	0.824	0.806	0.788	0.772	0.758	0.744	0.731	0.719
8.00	0.955	0.928	0.896	0.869	0.846	0.825	0.806	0.789	0.773	0.758	0.744	0.732	0.719
8.25	0.957	0.932	0.898	0.871	0.847	0.827	0.807	0.79	0.774	0.759	0.745	0.732	0.719
8.50	0.96	0.935	0.901	0.873	0.849	0.828	0.809	0.791	0.775	0.76	0.746	0.732	0.72
8.75	0.963	0.939	0.903	0.875	0.85	0.829	0.81	0.792	0.776	0.76	0.746	0.733	0.721
9.00	0.966	0.943	0.906	0.877	0.852	0.83	0.811	0.793	0.776	0.761	0.747	0.734	0.721
9.25	0.97	0.947	0.909	0.879	0.853	0.832	0.812	0.794	0.777	0.762	0.747	0.734	0.721
9.50	0.973	0.95	0.911	0.881	0.855	0.833	0.813	0.795	0.778	0.763	0.748	0.734	0.722
9.75	0.977	0.954	0.914	0.883	0.857	0.834	0.814	0.796	0.779	0.763	0.749	0.735	0.722
10.00	0.981	0.957	0.917	0.885	0.859	0.836	0.815	0.797	0.78	0.764	0.749	0.735	0.722
10.25	0.984	0.959	0.92	0.887	0.86	0.837	0.816	0.798	0.78	0.764	0.75	0.736	0.723
10.50	0.961	0.923	0.889	0.862	0.838	0.817	0.799	0.781	0.765	0.75	0.737	0.723
10.75	0.962	0.925	0.891	0.863	0.839	0.818	0.799	0.782	0.766	0.751	0.737	0.724
11.00	0.963	0.928	0.893	0.865	0.84	0.819	0.8	0.782	0.766	0.751	0.737	0.724
11.25	0.964	0.93	0.893	0.865	0.84	0.819	0.799	0.781	0.765	0.75	0.736	0.723
11.50	0.964	0.931	0.894	0.865	0.84	0.818	0.798	0.78	0.764	0.749	0.735	0.722
11.75	0.965	0.932	0.894	0.865	0.839	0.817	0.797	0.78	0.763	0.748	0.734	0.721
12.00	0.966	0.933	0.894	0.864	0.839	0.817	0.797	0.779	0.762	0.747	0.733	0.719
12.25	0.967	0.935	0.895	0.864	0.839	0.816	0.796	0.778	0.761	0.746	0.732	0.718
12.50	0.967	0.936	0.896	0.864	0.838	0.816	0.796	0.777	0.76	0.745	0.731	0.717
12.75	0.968	0.937	0.896	0.864	0.838	0.815	0.795	0.776	0.759	0.744	0.729	0.716
13.00	0.969	0.939	0.896	0.864	0.837	0.814	0.794	0.775	0.758	0.743	0.728	0.715
13.25	0.971	0.94	0.897	0.864	0.837	0.813	0.792	0.774	0.757	0.741	0.727	0.713
13.50	0.972	0.942	0.897	0.863	0.837	0.813	0.792	0.773	0.756	0.74	0.725	0.712
14.00	0.976	0.946	0.897	0.863	0.835	0.811	0.79	0.771	0.753	0.737	0.723	0.709
14.25	0.978	0.947	0.898	0.862	0.834	0.81	0.789	0.77	0.752	0.736	0.721	0.707
14.50	0.948	0.898	0.862	0.833	0.809	0.787	0.768	0.751	0.734	0.72	0.706

Table XVIII-1110-1M
Superheat Correction Factor K_{sh} (Cont'd)

Flowing Pressure (MPa)	Superheat Correction Factor K_{sh} Total Temperature, °C, of Superheated Steam																	
	205	225	250	275	300	325	350	375	400	425	450	475	500	525	550	575	600	625
14.75	0.948	0.898	0.862	0.832	0.808	0.786	0.767	0.749	0.733	0.719	0.704
15.00	0.948	0.899	0.861	0.832	0.807	0.785	0.766	0.748	0.732	0.717	0.703
15.25	0.947	0.899	0.861	0.831	0.806	0.784	0.764	0.746	0.73	0.716	0.702
15.50	0.947	0.899	0.861	0.83	0.804	0.782	0.763	0.745	0.728	0.714	0.7
15.75	0.946	0.899	0.86	0.829	0.803	0.781	0.761	0.743	0.727	0.712	0.698
16.00	0.945	0.9	0.859	0.828	0.802	0.779	0.759	0.741	0.725	0.71	0.696
16.25	0.945	0.9	0.859	0.827	0.801	0.778	0.757	0.739	0.723	0.708	0.694
16.50	0.945	0.9	0.858	0.826	0.799	0.776	0.756	0.738	0.721	0.706	0.692
16.75	0.944	0.9	0.857	0.825	0.797	0.774	0.754	0.736	0.719	0.704	0.69
17.00	0.944	0.9	0.856	0.823	0.796	0.773	0.752	0.734	0.717	0.702	0.688
17.25	0.944	0.9	0.855	0.822	0.794	0.771	0.75	0.732	0.715	0.7	0.686
17.50	0.944	0.9	0.854	0.82	0.792	0.769	0.748	0.73	0.713	0.698	0.684
17.75	0.944	0.9	0.853	0.819	0.791	0.767	0.746	0.728	0.711	0.696	0.681
18.00	0.944	0.901	0.852	0.817	0.789	0.765	0.744	0.725	0.709	0.694	0.679
18.25	0.945	0.901	0.851	0.815	0.787	0.763	0.742	0.723	0.706	0.691	0.677
18.50	0.945	0.901	0.85	0.814	0.785	0.761	0.739	0.72	0.704	0.689	0.674
18.75	0.945	0.901	0.849	0.812	0.783	0.758	0.737	0.718	0.701	0.686	0.671
19.00	0.946	0.901	0.847	0.81	0.781	0.756	0.734	0.715	0.698	0.683	0.669
19.25	0.948	0.901	0.846	0.808	0.778	0.753	0.732	0.713	0.696	0.681	0.666
19.50	0.95	0.9	0.844	0.806	0.776	0.75	0.729	0.71	0.693	0.677	0.663
19.75	0.952	0.899	0.842	0.803	0.773	0.748	0.726	0.707	0.69	0.674	0.66
20.00	0.899	0.84	0.801	0.77	0.745	0.723	0.704	0.687	0.671	0.657
20.25	0.899	0.839	0.798	0.767	0.742	0.72	0.701	0.683	0.668	0.654
20.50	0.899	0.837	0.795	0.764	0.738	0.717	0.697	0.68	0.665	0.651
20.75	0.898	0.834	0.792	0.761	0.735	0.713	0.694	0.677	0.661	0.647
21.00	0.896	0.832	0.79	0.758	0.732	0.71	0.691	0.673	0.658	0.643
21.25	0.894	0.829	0.786	0.754	0.728	0.706	0.686	0.669	0.654	0.64
21.50	0.892	0.826	0.783	0.75	0.724	0.702	0.682	0.665	0.65	0.636
21.75	0.891	0.823	0.779	0.746	0.72	0.698	0.679	0.661	0.646	0.631
22.00	0.887	0.82	0.776	0.743	0.716	0.694	0.674	0.657	0.641	0.627

XVIII-1124 Example 4

Given: A safety valve having a certified rating of 8,000 lb/hr of steam for a pressure setting of 2,000 psi.

Problem: Find the relieving capacity under the following conditions:

- (a) Steam quality (dryness fraction) of 0.93
 (b) Superheat of 100°F (total temperature of 737°F)

Solution:

- (a) The wet saturated capacity:

$$W = \frac{W_s}{0.93} = \frac{8,000}{0.93} = 8,600 \text{ lb/hr}$$

- (b) The superheat capacity:

$$W = K_{sh} W_s = (0.9152)(8,000) = 7,320 \text{ lb/hr}$$

NOTE: K_{sh} interpolated from Table XVIII-1110-1 for flowing conditions.

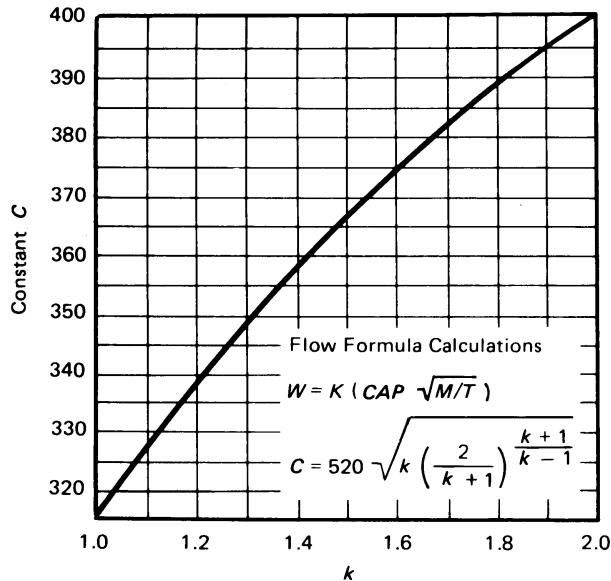
XVIII-1130 THEORETICAL FLOW

The theoretical flow for use in the establishment of the coefficient of discharge shall be calculated using eqs. XVIII-1110(1) through XVIII-1110(b)(4) with K being deleted.

XVIII-1140 SATURATED WATER CAPACITY

(a) Because the saturated water capacity is configuration sensitive, the following applies only to those safety valves that have a nozzle type construction (throat to inlet diameter ratio of 0.25 to 0.80 with a continuously contoured change and have exhibited a coefficient K_D in excess of 0.90). No saturated water rating shall apply to other types of construction.

Figure XVIII-1110-1
Constant C for Gas or Vapor Related to Ratio of Specific Heats ($k = c_p/c_v$)



k	Constant C	k	Constant C	k	Constant C
1.00	315	1.26	343	1.52	366
1.02	318	1.28	345	1.54	368
1.04	320	1.30	347	1.56	369
1.06	322	1.32	349	1.58	371
1.08	324	1.34	351	1.60	372
1.10	327	1.36	352	1.62	374
1.12	329	1.38	354	1.64	376
1.14	331	1.40	356	1.66	377
1.16	333	1.42	358	1.68	379
1.18	335	1.44	359	1.70	380
1.20	337	1.46	361	2.00	400
1.22	339	1.48	363	2.20	412
1.24	341	1.50	364

NOTE: The manufacturer, user, and Inspector are all cautioned that for the following rating to apply, the valve shall be continuously subjected to saturated water. If, after initial relief, the flow medium changes to quality steam, the valve shall be rated as per saturated steam. Valves installed on vessels or lines containing steam-water mixture shall be rated on saturated steam.

(b) To determine the saturated water capacity of a valve currently rated and meeting the requirements of (a) above, refer to Figure XVIII-1140-1. Enter the graph at the set pressure of the valve, move upward to the saturated water line, and read horizontally the relieving capacity. This capacity is the theoretical, isentropic value arrived at by assuming equilibrium flow and calculated values for the critical pressure ratio.

XVIII-1150 EQUATIONS FOR CONVERSION (INCOMPRESSIBLE FLUIDS)¹⁵

The capacity of a pressure relief valve in terms of a liquid other than the medium for which the valve was officially rated shall be determined by application of the following equation:¹⁶

$$W_t = C K A \sqrt{(P - P_d)W}$$

where

A = actual discharge area of valve, in.² (mm²)

C = 2,407 (5.092)

K = coefficient of discharge

P = (set pressure \times 1.10) plus atmospheric pressure, psia (MPa_{abs})

P_d = pressure at discharge from valve, psia (MPa_{abs})

W = density of liquid at valve inlet conditions, lb/ft³ (kg/m³)

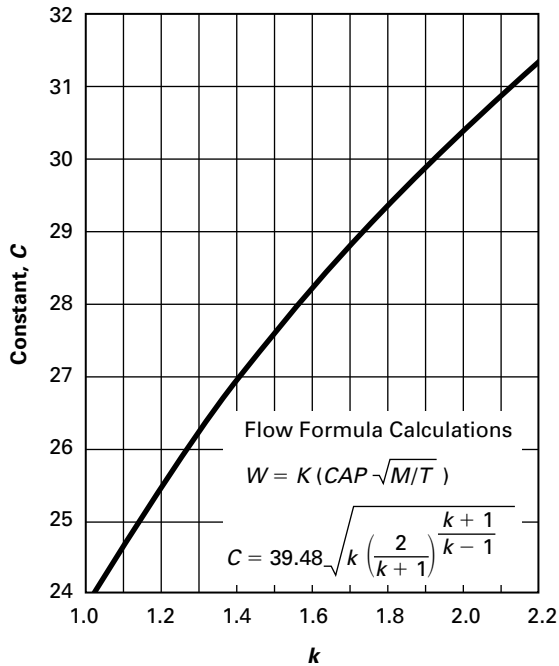
W_t = rated capacity, lb/hr (kg/h) of any liquid

W_w = rated capacity, lb/hr (kg/h) water @ 70°F

Table XVIII-1110(a)-1
Molecular Weights of Gases and Vapors

Gas or Vapor	Molecular Weight	Gas or Vapor	Molecular Weight
Air	28.97	Freon 22	86.48
Acetylene	26.04	Freon 114	170.90
Ammonia	17.03	Hydrogen	2.02
Butane	58.12	Hydrogen sulfide	34.08
Carbon dioxide	44.01	Methane	16.04
Chlorine	70.91	Methyl chloride	50.48
Ethane	30.07	Nitrogen	28.02
Ethylene	28.05	Oxygen	32.00
Freon 11	137.371	Propane	44.09
Freon 12	120.9	Sulfur dioxide	64.06

Figure XVIII-1110-1M
Constant C for Gas or Vapor Related to Ratio of Specific Heats ($k = c_p/c_v$)



k	Constant C	k	Constant C	k	Constant C
1.001	23.95	1.26	26.05	1.52	27.80
1.02	24.12	1.28	26.20	1.54	27.93
1.04	24.30	1.30	26.34	1.56	28.05
1.06	24.47	1.32	26.49	1.58	28.17
1.08	24.64	1.34	26.63	1.60	28.29
1.10	24.81	1.36	26.76	1.62	28.40
1.12	24.97	1.38	26.90	1.64	28.52
1.14	25.13	1.40	27.03	1.66	28.63
1.16	25.29	1.42	27.17	1.68	28.74
1.18	25.45	1.44	27.30	1.70	28.86
1.20	25.60	1.46	27.43	2.00	30.39
1.22	25.76	1.48	27.55	2.20	31.29
1.24	25.91	1.50	27.68

XVIII-1160 EXAMPLE

Given: Pressure relief valve bearing a certified rating of 1,500 gpm water @ 70°F with a set pressure of 120 psig.

Problem: Find the flow capacity of this pressure relief valve in gpm of kerosene ($G = 0.82$) at the same pressure rating.

Solution:

(a) For water at 70°F:

$$KA = \frac{W_W \times 500}{2,407 \sqrt{(p - 14.7)(62.3)}}$$

$$KA = \frac{1,500 \times 500}{2,407 \sqrt{(146.7 - 14.7)(62.3)}} = 3.436$$

(b) For kerosene:

$$W_l = 38 (3.436) \sqrt{\frac{(146.7 - 14.7)}{0.82}}$$

$$W_l = 1,656.60 \text{ gpm}$$

Figure XVIII-1140-1
Flow Capacity Curve for Rating Nozzle Type
Safety Valves on Saturated Water (Based on
10% Overpressure)

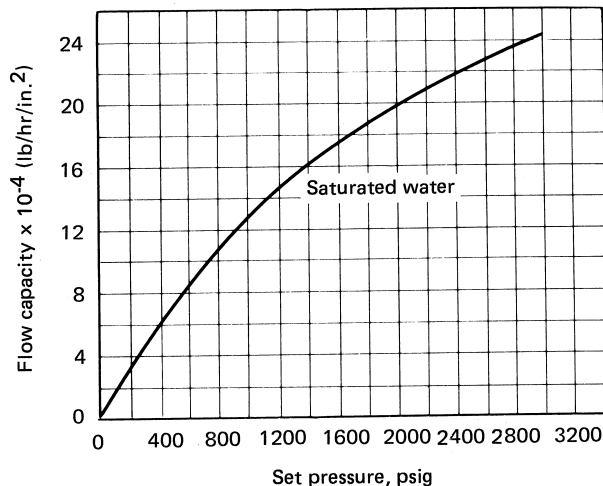
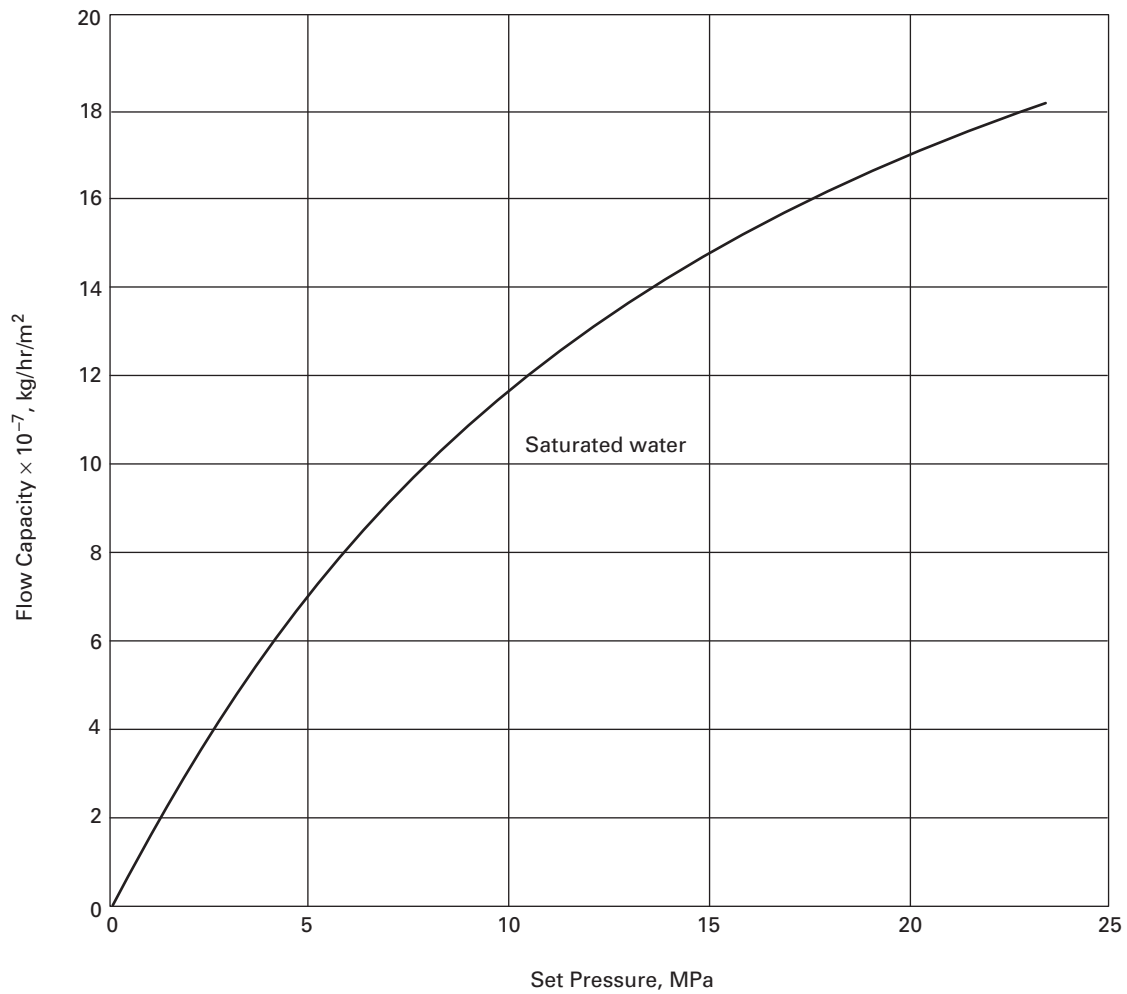


Figure XVIII-1140-1M
Flow Capacity Curve for Rating Nozzle Type Safety Valves on Saturated Water (Based on 10% Overpressure)



MANDATORY APPENDIX XIX

ARTICLE XIX-1000 INTEGRAL FLAT HEAD WITH A LARGE OPENING

XIX-1100 GENERAL REQUIREMENTS

XIX-1110 SCOPE

(a) Rules of this Appendix apply to flat heads which have a single, circular, centrally located opening that exceeds one-half of the head diameter and have a shell/head juncture which is integrally formed or integrally attached with a full penetration weld similar to those shown in [Figure XIX-1110-1](#). Heads of this type shall be designed according to the rules which follow and related parts of [Mandatory Appendix XI](#).

(b) A general arrangement of an integral flat head with and without a nozzle attached at the central opening is shown in [Figure XIX-1110-2](#).

XIX-1120 NOMENCLATURE

(a) Except as given below, the symbols used in the equations of [XIX-1200](#) are defined in [XI-3130](#):

- A = outside diameter of flat head and shell
- B_n = diameter of central opening (for nozzle, this is inside diameter and for opening without a nozzle, diameter of the opening)
- B_s = inside diameter of shell (measured below tapered hub if one exists)
- M_H = moment acting at shell/head junction
- P = internal design pressure
- t = flat head nominal thickness

**Figure XIX-1110-1
Applicable Configurations of Flat Heads**

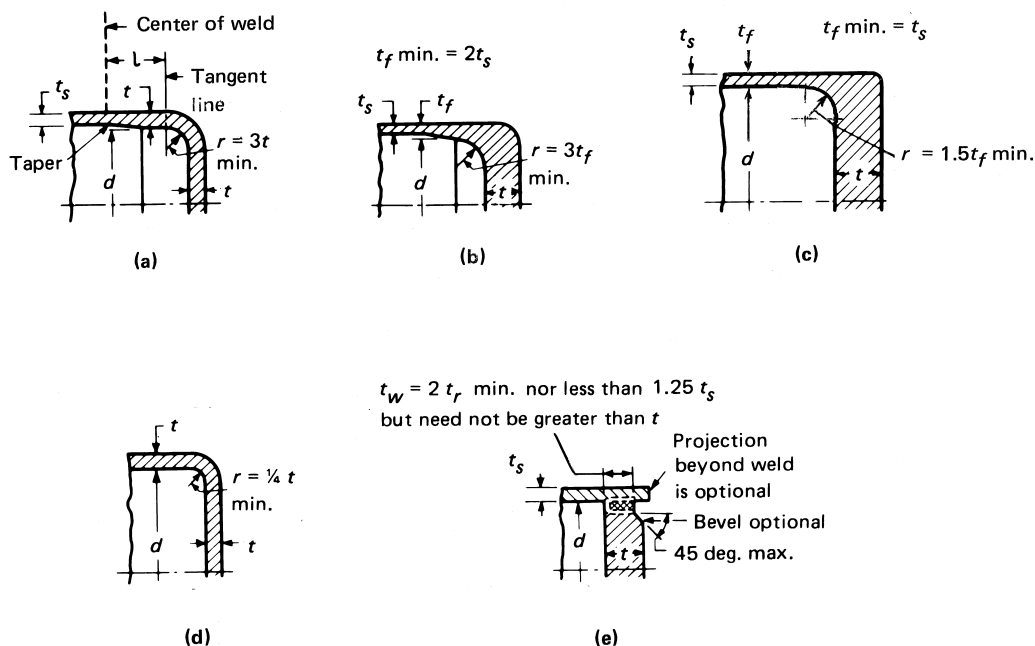
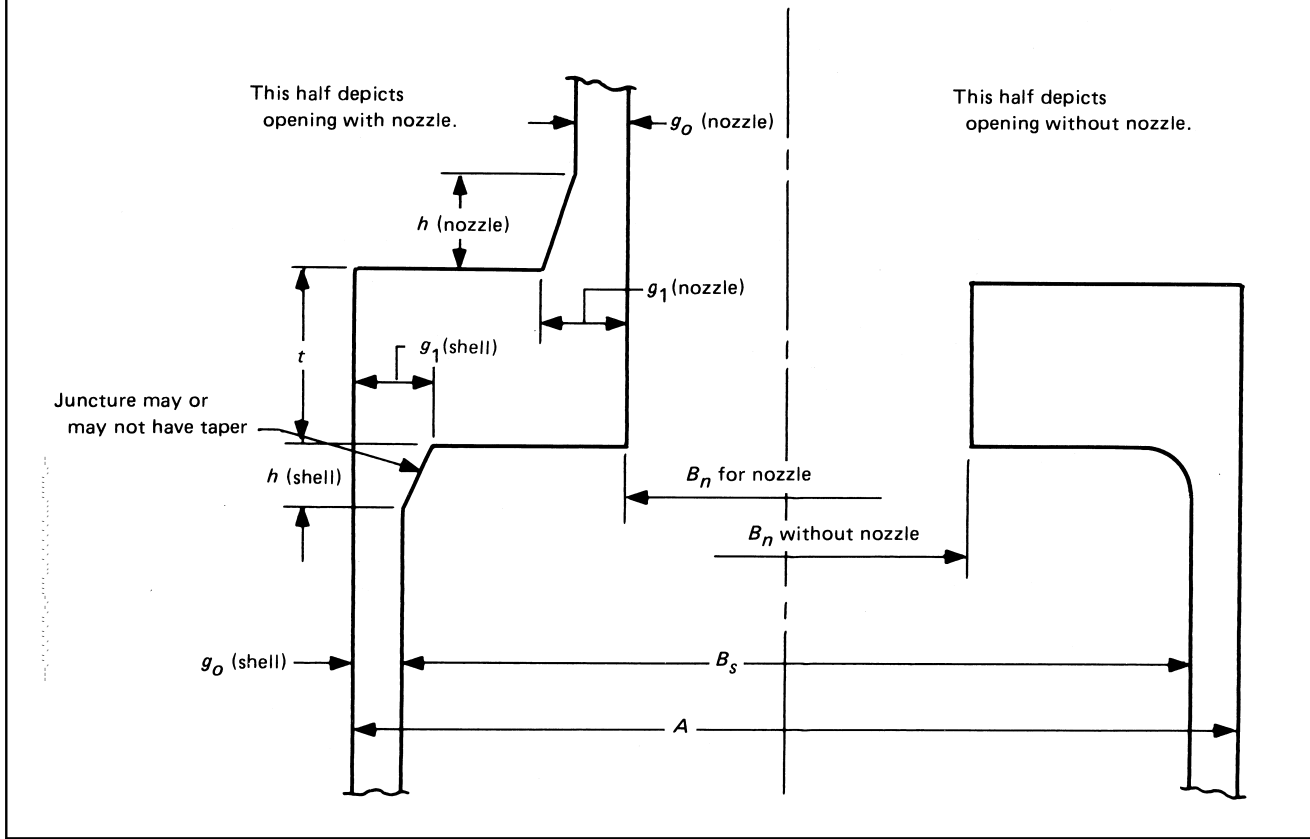


Figure XIX-1110-2
Integral Flat Head With Large Central Opening



B_1 , g_0 , g_1 , h_0 , F , V , and f are defined in XI-3130. These terms may refer to either the shell/flat head juncture or to the flat head/central opening juncture and depend upon details of those junctures.

XIX-1200 DESIGN PROCEDURE

(a) Disregard the shell attached to the outside diameter of the flat head and then analyze the flat head with a central opening (with or without nozzle) in accordance with these rules.

(1) Calculate the operating moment, M_0 , according to XI-3230 (there is no M_0 for gasket seating to be considered). The equations in Mandatory Appendix XI for loads (XI-3130) and moment arms (Table XI-3230-1) shall be used directly with the following designations and terms substituted for terms in Mandatory Appendix XI. Let

$C = G =$ inside diameter of shell B_s

$B = B_n$, where B_n is as shown in Figure XIX-1110-2 depending on the presence of an integral nozzle or opening without a nozzle

(2) With $K = A/B_n$, use XI-3240 to calculate stresses. Designate the calculated stresses S_H^* , S_R^* , and S_T^* .

(b) Calculate $(E\theta)^*$ as follows:

(1) for integrally attached nozzle:

$$(E\theta)^* = \frac{0.91(g_1/g_0)^2 B_1 V}{f h_0} S_H^*$$

(2) for an opening without a nozzle:

$$(E\theta)^* = (B_n/t) S_T^*$$

where g_0 , g_1 , B_1 , V , f , h_0 , and B_n all pertain to the flat head/opening as described in (a).

(c) Calculate $(E\theta)^*/M_0$.

(d) Calculate M_H :

$$M_H = \frac{(E\theta)^*}{\frac{1.74 h_0 V}{g_0^3 B_1} + \frac{(E\theta)^*}{M_0} (1 + Ft/h_0)}$$

where h_0 , V , g_0 , B_1 , and F refer to the shell attached to the outside diameter of the flat head.

(e) Calculate X_1 :

$$X_1 = \frac{M_0 - M_H(1 + Ft/h_0)}{M_0}$$

where F and h_0 refer to the shell.

(f) Calculate stresses at head/shell juncture as follows.

(1) For longitudinal hub stress in shell:

$$S_{HS} = (X_1)(E\theta)^* \frac{1.10h_0f}{(g_1/g_0)^2 B_s V}$$

where h_0, f, g_0, g_1, B_s , and V refer to the shell.

(2) For radial stress at outside diameter:

$$S_{RS} = \frac{1.91M_H(1 + Ft/h_0)}{B_s t^2} + \frac{0.64FM_H}{B_s h_0 t}$$

where B_s, F , and h_0 refer to the shell.

(3) For tangential stress at outside diameter:

$$S_{TS} = \frac{(X_1)(E\theta)^* t}{B_s} - \frac{0.57(1 + Ft/h_0)M_H}{B_s t^2} + \frac{0.64FZM_H}{B_s h_0 t}$$

where B_s, F , and h_0 refer to the shell, and

$$Z = \frac{K^2 + 1}{K^2 - 1}$$

(g) Calculate stresses at opening/head juncture as follows.

(1) For longitudinal hub stress in central opening:

$$S_{HO} = X_1 S_H^*$$

(2) For radial stress at central opening:

$$S_{RO} = X_1 S_R^*$$

(3) For tangential stress at diameter of central opening:

$$S_{TO} = X_1 S_T^* + \frac{0.64FZ_1 M_H}{B_s h_0 t}$$

where F, B_s , and h_0 refer to the shell, and

$$Z_1 = \frac{2K^2}{K^2 - 1}$$

(h) The preceding calculated stresses shall meet the allowable stresses in [XI-3250](#).

(13)

MANDATORY APPENDIX XX

DELETED

MANDATORY APPENDIX XXI

ARTICLE XXI-1000 ADHESIVE ATTACHMENT OF NAMEPLATES

XXI-1100 INTRODUCTION

XXI-1110 SCOPE

This Appendix provides minimum requirements for the use of adhesive systems for the attachment of nameplates, limited to

(a) the use of pressure sensitive acrylic adhesives that have been preapplied by the nameplate manufacturer to a normal thickness of at least 0.005 in. (0.13 mm) and that are protected with a moisture stable liner

(b) use on items with Design Temperatures within the range of -40°F to 300°F (-40°C to 150°C), inclusive

(c) application to clean, bare metal surfaces, with removal of antiweld spatter compound which may contain silicone

(d) nameplate nominal thickness not less than 0.020 in.

(e) use of prequalified application procedures as outlined in this Article

(f) use of the preapplied adhesive within an interval of 2 yr after adhesive application

XXI-1120 NAMEPLATE APPLICATION PROCEDURE QUALIFICATION

(a) The Certificate Holder's Quality Assurance Manual shall require that written procedures, acceptable to the Authorized Inspection Agency, for the application of adhesive backed nameplates shall be prepared and qualified.

(b) The application procedure qualification shall include the following essential variable, using the adhesive and nameplate manufacturers' recommendations where applicable:

(1) description of the pressure sensitive acrylic adhesive system employed, including generic composition

(2) the qualified temperature range [the cold box test temperature shall be -40°F (-40°C) for all applications]

(3) materials of nameplate and item when the mean coefficient of expansion at design temperature of one material is less than 85% of that for the other material

(4) finish of the nameplate and item surfaces to which the nameplate is to be attached

(5) the nominal thickness and modulus of elasticity at application temperature of the nameplate when nameplate preforming is employed. A change of more than 25% in the quantity [(nameplate nominal thickness)² × nameplate modulus of elasticity at application temperature] will require requalification

(6) the qualified range of preformed nameplate and companion item contour combinations when preforming is employed

(7) cleaning requirements for the item prior to attachment of the nameplate

(8) application temperature range and application pressure technique

(9) application steps and safeguards

(c) Each procedure used for nameplate attachment by pressure sensitive acrylic adhesive systems shall be qualified for outdoor exposure in accordance with Standard UL-969-82, Marking and Labeling Systems, with the following additional requirements.

(1) Width of nameplate test strip shall not be less than 1 in.

(2) Nameplates shall have an average adhesion of not less than 8 lb/in. (55 kPa) of width after all exposure conditions, including low temperature.

(d) Any change in (b) above shall require requalification.

(e) Each package of nameplates shall be identified with the adhesive application date.

MANDATORY APPENDIX XXII

ARTICLE XXII-1000 RULES FOR REINFORCEMENT OF CONE-TO-CYLINDER JUNCTION UNDER EXTERNAL PRESSURE

XXII-1100 INTRODUCTION

XXII-1110 SCOPE

(a) The equations of this Appendix provide for the design of reinforcement, if needed, at the cone-to-cylinder junctions for reducer sections and conical heads where all the elements have a common axis and the half-apex angle $\alpha \leq 60$ deg. Subparagraph XXII-1300(d) provides for special analysis in the design of cone-to-cylinder intersections with or without reinforcing rings where α is greater than 60 deg.

(b) In the design of reinforcement for a cone-to-cylinder juncture, the requirements of ND-3336 shall be met.

XXII-1200 NOMENCLATURE

The nomenclature given below is used in the equations of the following subparagraphs:

- A = factor determined from the applicable chart in Section II, Part D, Subpart 3 for the material used in the stiffening ring, corresponding to the factor B , below, and the design temperature for the shell under consideration
- A_e = effective area of reinforcement due to excess metal thickness
- A_{rL} = required area of reinforcement at large end
- A_{rS} = required area of reinforcement at small end
- A_s = cross-sectional area of the stiffening ring
- A_T = equivalent area of cylinder, cone, and stiffening ring
- where

$$A_{TL} = \frac{L_L t_S}{2} + \frac{L_C t_C}{2} + A_s, \text{ for large end}$$

$$A_{TS} = \frac{L_S t_S}{2} + \frac{L_C t_C}{2} + A_s, \text{ for small end}$$

B = factor determined from the applicable chart in Section II, Part D, Subpart 3 for the material used for the stiffening

D_L = outside diameter of large end of conical section under consideration

D_o = outside diameter of cylindrical shell (In conical shell calculations, the value of D_s and D_L should be used in calculations in place of D_o depending on whether the small end D_s , or large end D_L , is being examined.)

D_s = outside diameter at small end of conical section under consideration

E = lowest efficiency of the longitudinal joint in the shell or head or of the joint in the reducer; $E = 1$ for butt welds in compression

E_c = modulus of elasticity of cone material

E_R = modulus of elasticity of reinforcing material

E_s = modulus of elasticity of shell material

$E_x = E_o, E_R, \text{ or } E_s$

f_1 = axial load at large end (excluding pressure P), lb/in. (N/mm)

f_2 = axial load at small end (excluding pressure P), lb/in. (N/mm)

I_s = required moment of inertia of the stiffening ring cross section about its neutral axis parallel to the axis of the shell

I_s' = required moment of inertia of the combined ring-shell-cone cross section about its neutral axis parallel to the axis of the shell, in.⁴ (mm⁴). The width of shell which is taken as contributing to the moment of inertia of the combined section shall not be greater than $1.10 \sqrt{D_o T}$ and shall be taken as lying one-half on each side of the centroid of the ring. Portions of the shell plate shall not be considered as contributing area to more than one stiffening ring. If the stiffeners should be so located that the maximum permissible effective shell sections overlap on either or both sides of a stiffener, the effective shell section for that stiffener shall be shortened by one-half of each overlap.

$$k = \frac{S_s E_s}{S_R E_R} \text{ but not less than } 1.0$$

L = axial length of cone

L_c = length of cone between stiffening rings measured along surface of cone. For cones without intermediate stiffeners,

$$L_c = \sqrt{L^2 + (R_L - R_s)^2}$$

L_L = design length of a vessel section taken as the largest of the following:

(a) the center-to-center distance between the cone-to-large-shell junction and an adjacent stiffening ring on the large shell;

(b) the distance between the cone-to-large-shell junction and one-third the depth of head on the other end of the large shell if no other stiffening rings are used.

L_s = design length of a vessel section taken as the largest of the following:

(a) the center-to-center distance between the cone-to-small-shell junction and an adjacent stiffening ring on the small shell;

(b) the distance between the cone-to-small-shell junction and one-third the depth of the head on the other end of the small shell if no other stiffening rings are used.

P = external design pressure

$Q_L = \frac{PR_L}{2} + f_1$; axial compressive force at large end due to pressure and f_1 , lb/in. (N/mm)

$Q_s = \frac{PR_s}{2} + f_2$; axial compressive force at small end due to pressure and f_2 , lb/in. (N/mm)

R_L = inside radius of large cylinder

R_s = inside radius of small cylinder

S' = the lesser of twice the allowable stress at design metal temperature from Section II, Part D, Subpart 1, Tables 1A and 1B or 0.9 times the tabulated yield strength at design metal temperature from Section II, Part D, Subpart 2, Tables Y-1 and Y-2

S_R = allowable stress of reinforcing material

S_s = allowable stress of shell

T = minimum required thickness of cylinder at cone-to-cylinder junction, exclusive of corrosion allowance (see ND-3133.3)

T_c = nominal thickness of cone at cone-to-cylinder junction, exclusive of corrosion allowance (see ND-3121)

T_L = the smaller of $(T_s - T)$ or $(T_c - T_r)$

T_r = minimum required thickness of cone at cone-to-cylinder junction, exclusive of corrosion allowance

T_s = nominal thickness of cylinder at cone-to-cylinder junction, exclusive of corrosion allowance (see ND-3121)

α = one-half the included (apex) angle of the cone at the center line of the head

Δ = value to indicate need for reinforcement at cone-to-cylinder intersection having a half-apex angle $\alpha \leq 60$ deg. When $\Delta \geq \alpha$, no reinforcement is required at the junction (see Table XXII-1200-1).

XXII-1300 DESIGN PRESSURE

(a) Reinforcement shall be provided at the junction of the cone with the large cylinder for conical heads and reducers without knuckles when the value of Δ obtained from Table XXII-1200-1 using the appropriate ratio $P/S'E$, is less than α . Interpolation may be made in the Table.

The cross-sectional area of the reinforcement ring shall be at least equal to that indicated by the following equation:

$$A_{rL} = \frac{kQ_LR_L \tan \alpha}{S'E} \left[1 - \frac{1}{4} \left(\frac{PR_L - Q_L}{Q_L} \right) \frac{\Delta}{\alpha} \right] \quad (1)$$

When the thickness, less corrosion allowance, of both the reducer and cylinder exceeds that required by the applicable design equations, the minimum excess thickness may be considered to contribute to the required reinforcement ring in accordance with the following equation:

$$A_e = 4T_L \sqrt{R_L T_s} \quad (2)$$

Any additional area of reinforcement which is required shall be situated within a distance of $\sqrt{R_L T_s}$ from the junction of the reducer and the cylinder. The centroid of the added area shall be within a distance of $0.5 \sqrt{R_L T_s}$ from the junction.

The reinforcement ring at the cone-to-cylinder junction shall also be considered as a stiffening ring. The required moment of inertia of a circumferential stiffening ring cross section shall not be less than that determined by the following equation:

$$I_s = \frac{AD_L^2 A_{TL}}{14.0}$$

Table XXII-1200-1
Values of Δ for Junctions at the Large
Cylinder for $\alpha \leq 60$ deg

$P/S'E$	0	0.002	0.005	0.010	0.02
Δ , deg	0	5	7	10	15
$P/S'E$	0.04	0.08	0.10	0.125	0.15
Δ , deg	21	29	33	37	40
$P/S'E$	0.20	0.25	0.30	0.35	
Δ , deg	47	52	57	60	[Note (1)]

NOTE:

(1) $\Delta = 60$ deg for greater values of $P/S'E$.

The required moment of inertia of the combined ring-shell-core cross section shall not be less than that determined by the following equation:

$$I'_s = \frac{AD_L^2 A_{TL}}{10.9}$$

The moment of inertia for a stiffening ring at the large end shall be determined by the following procedure:

Step 1. Assuming that the shell has been designed and D_L , L_L , and T are known, select a member to be used for the stiffening ring and determine cross-sectional area A_{TL} . Then calculate factor B using the following equation:

$$B = 3/4 \left(\frac{F_L D_L}{A_{TL}} \right)$$

where

$$F_L = PM + f_1 \tan \alpha$$

$$M = \frac{-R_L \tan \alpha}{2} + \frac{L_L}{2} + \frac{R_L^2 - R_s^2}{3R_L \tan \alpha}$$

A_{TL} was defined previously.

Step 2. Enter the right-hand side of the applicable material chart in Section II, Part D, Subpart 3 for the material under consideration at the value of B determined by [Step 1](#). If different materials are used for the shell and stiffening ring, use the material chart resulting in the larger value of A in [Step 4](#), below.

Step 3. Move horizontally to the left to the material/temperature line for the design metal temperature. For values of B falling below the left end of the material/temperature line, see [Step 5](#).

Step 4. Move vertically to the bottom of the chart and read the value of A .

Step 5. For value of B falling below the left end of the material/temperature line for the design temperature, the value of A can be calculated using the formula $A = 2B/E_x$.

Step 6. Compute the value of the required moment of inertia from the equations for I_s or I'_s above.

Step 7. Calculate the available moment of inertia of the stiffening ring using the section corresponding to that used in [Step 6](#).

Step 8. If the required moment of inertia is greater than the moment of inertia for the section selected in [Step 1](#), a new section with a larger moment of inertia must be selected and a new moment of inertia determined. If the required moment of inertia is smaller than the moment of inertia of the section selected in [Step 1](#), that section is satisfactory.

The requirements of ND-4430 are to be met in attaching stiffening rings to the shell.

(b) Reinforcement shall be provided at the junction of the conical shell of a reducer without a flare and the small cylinder. The cross-sectional area of the reinforcement ring shall be at least equal to that indicated by the following formula:

$$A_{rS} = \frac{kQ_s R_s \tan \alpha}{S' E} \quad (3)$$

When the thickness, less corrosion allowance, of either the reducer or cylinder exceeds that required by the applicable design formula, the thickness may be considered to contribute to the required reinforcement ring in accordance with the following formula:

$$A_e = \sqrt{\frac{R_s T_c}{\cos \alpha}} (T_c - T_r) + \sqrt{R T_s} (T_s - T) \quad (4)$$

Any additional area of reinforcement which is required shall be situated within a distance of $\sqrt{R_s T_s}$ from the junction, and the centroid of the added area shall be within a distance of $0.5\sqrt{R_s T_s}$ from the junction.

The reinforcement ring at the cone-to-cylinder junction shall also be considered as a stiffening ring. The required moment of inertia of a circumferential stiffening ring cross section shall not be less than that determined by the following equation:

$$I_s = \frac{AD_s^2 A_{TS}}{14.0}$$

The required moment of inertia of the combined ring-shell-cone cross section shall not be less than that determined by the following equation:

$$I'_s = \frac{AD_s^2 A_{TS}}{10.9}$$

The moment of inertia for a stiffening ring at the small end shall be determined by the following procedure.

Step 1. Assuming that the shell has been designed and D_s , L_s , and T are known, select a member to be used for the stiffening ring and determine cross-sectional area A_{TS} . Then calculate factor B using the following equation:

$$B = 3/4 \left(\frac{F_s D_s}{A_{TS}} \right)$$

where

$$F_s = PN + f_2 \tan \alpha$$

$$N = \frac{R_s \tan \alpha}{2} + \frac{L_s}{2} + \frac{R_L^2 - R_s^2}{3R_L \tan \alpha}$$

A_{TS} was defined previously.

Step 2. Enter the right-hand side of the applicable material chart in Section II, Part D, Subpart 3 for the material under consideration at the value of B determined by

Step 1. If different materials are used for the shell and stiffening ring, use the material chart resulting in the larger value of A in **Step 4**, below.

Step 3. Move horizontally to the left to the material/temperature line for the design metal temperature. For values of B falling below the left end of the material/temperature line, see **Step 5**.

Step 4. Move vertically to the bottom of the chart and read the value of A .

Step 5. For values of B falling below the left end of the material/temperature line for the design temperature, the value of A can be calculated using the formula $A = 2B/E_x$.

Step 6. Compute the value of the required moment of inertia from the equations for I_s or I_s' above.

Step 7. Calculate the available moment of inertia of the stiffening ring using the section corresponding to that used in **Step 6**.

Step 8. If the required moment of inertia is greater than the moment of inertia for the section selected in **Step 1**, a new section with a larger moment of inertia must be

selected and a new moment of inertia determined. If the required moment of inertia is smaller than the moment of inertia for the section selected in **Step 1**, that section is satisfactory.

The requirements of ND-4430 are to be met in attaching stiffening rings to the shell.

(c) Reducers, such as those made up of two or more conical frustums having different slopes, may be designed in accordance with (d) below.

(d) As an alternative to the rules provided in the preceding (a) and (b) and when half the apex angle is greater than 60 deg, the design may be based on special analysis such as numerical methods or the beam-on-elastic-foundation analysis of Timoshenko, Hetenyi, or Watts and Lang. The stresses at the junction shall meet all of the allowable stress limits of this Division. The effect of shell and cone buckling on the required area and moment of inertia at the joint shall also be considered in the analysis. The theoretical buckling pressure of the junction shall be at least 3.3 times the allowable external design pressure of the junction.

MANDATORY APPENDIX XXIII

ARTICLE XXIII-1000 QUALIFICATIONS AND DUTIES OF SPECIALIZED PROFESSIONAL ENGINEERS

XXIII-1100 SCOPE

This Appendix presents minimum requirements for the qualification of personnel engaged in the certification activities. The personnel addressed are those who perform the following specialty fields:

(a) for Division 1

(1) certification of the Design Specification on behalf of the Owner

(2) certification of the Design Report on behalf of the N Certificate Holder

(3) certification of the Overpressure Protection Report on behalf of the Owner

(4) certification of the Load Capacity Data Sheet on behalf of the N Certificate Holder

(b) for Division 2

(1) certification of the Design Specification on behalf of the Owner

(2) certification of the Construction Specification, Design Drawings, and Design Report on behalf of the Designer

(c) for Division 3

(1) certification of the Design Specification on behalf of the N3 Certificate Holder

(2) certification of the Design Report on behalf of the N3 Certificate Holder

(3) certification of the Fabrication Specification on behalf of the N3 Certificate Holder

Also provided are the duties of these personnel in the performance of the activities described above.

XXIII-1200 QUALIFICATIONS

XXIII-1210 GENERAL

One or more Registered Professional Engineers (PE) shall be selected by the Owner, Designer, N Certificate Holder, or N3 Certificate Holder, as applicable, to perform Code activities in the appropriate specialized field(s), provided the qualifications of the PE in meeting the requirements of this Appendix have been evaluated and verified by the Owner, Designer, N Certificate Holder, or N3

Certificate Holder, as applicable, responsible for the activity being certified or reviewed. Guidelines for demonstrating PE qualifications are contained in Guide C. A record of the qualifications of the PE shall be maintained by the responsible organization or the PE.

XXIII-1220 REGISTRATION AND EXPERIENCE

He shall be a Registered Professional Engineer in at least one state of the United States or Province of Canada and have the following qualifications.

He shall have 4 years of varied application experience at least 2 of which have been in each specialty field for which he performs certifying or review activities as delineated in [XXIII-1230](#) through [XXIII-1250](#). In addition, he shall keep current his knowledge of Code requirements and continue his professional development in his specialty field through personal study and experience, or by attendance at appropriate courses, seminars, Society meetings, and technical committee meetings. The Owner, Designer, N Certificate Holder, or N3 Certificate Holder, as applicable, shall review the qualifications of the PE at least once every 3 years to assure that his qualifications have been maintained. A continuing record of all such activity shall be included in the qualification records of the PE.

XXIII-1230 CERTIFIER OF THE DESIGN SPECIFICATION (DIVISIONS 1 AND 2)

To qualify as certifier of the Design Specification on behalf of the Owner, the PE shall be experienced in the applicable field of design and related nuclear power plant requirements, and in the application of the requirements of the Code relating to the construction of nuclear power plant items. This experience shall indicate that the PE has sufficient knowledge of anticipated plant and system operating and test conditions and their relationship to Code design criteria pertinent to the applicable Code item. In addition, he shall be knowledgeable of the specific Code requirements pertaining to his specialty field. Guidelines reflecting the degree of knowledge appropriate for preparation of the Design Specification are contained in Guide B, [Table B1](#).

XXIII-1231 Certifier of the Design Specification (Division 3)

To qualify as certifier of the Design Specification on behalf of the N3 Certificate Holder, the PE shall be experienced in the applicable field of Division 3 containment design requirements. This experience shall indicate that the PE has sufficient knowledge of anticipated Division 3 containment systems operating and test conditions and their relationship to Code design criteria pertinent to the applicable Code item. In addition, he shall be knowledgeable of the specific Code requirements pertaining to his specialty field. Guidelines reflecting the degree of knowledge appropriate for preparation of the Design Specification are contained in Guide B, [Table B6](#).

XXIII-1240 CERTIFIER OF THE LOAD CAPACITY DATA SHEET, CONSTRUCTION SPECIFICATION, DESIGN DRAWINGS, AND DESIGN REPORT (DIVISIONS 1 AND 2)

To qualify as certifier of the Load Capacity Data Sheet, the Construction Specification, Design Drawings, or Design Report, the PE shall be experienced in the applicable field of design and analysis and in the application of the requirements of the Code. In addition, he shall be knowledgeable of the specific Code requirements pertaining to his specialty field. Guidelines reflecting the degree of knowledge appropriate for preparation of the Design Report (Division 1), the Load Capacity Data Sheet, and the Construction Specification, Design Drawings, and Design Report (Division 2) are contained in Guide B, [Tables B2, B4, and B5](#), respectively.

XXIII-1241 Certifier of the Fabrication Specification and Design Report (Division 3)

To qualify as certifier of the Fabrication Specification and Design Report on behalf of the N3 Certificate Holder, the PE shall be experienced in the applicable field of design, analysis, fabrication, and the application of Division 3 requirements. Guidelines reflecting degree of knowledge appropriate for the preparation of the Design Report and Fabrication Specification are contained in Guide B, [Tables B7 and B8](#), respectively.

XXIII-1250 CERTIFIER OF THE OVERPRESSURE PROTECTION REPORT (DIVISIONS 1 AND 2)

To qualify as certifier of the Overpressure Protection Report on behalf of the Owner, the PE shall be experienced in nuclear power plant systems design, and in plant operation and safety control. In addition, he shall be knowledgeable of the specific Code requirements pertaining to his specialty field. Guidelines reflecting the degree of knowledge appropriate for preparation of the Report on Overpressure Protection are contained in Guide B, [Table B3](#).

XXIII-1300 DUTIES**XXIII-1310 GENERAL**

The certification activities covered in this Appendix may be performed only if the PE has assured himself that he is qualified to do so by virtue of a self-review establishing that his qualifications meet those required by this Appendix. He shall be familiar with the Quality Assurance requirements of the organization responsible for providing the document as these requirements relate to his work. For certification activities, the document being certified must have been reviewed in detail by the certifying PE, or prepared by him or prepared under his responsible direction. He shall prepare a statement¹⁷ to be affixed to the document attesting to compliance with the applicable requirements of the Code.

XXIII-1320 CERTIFICATION OF THE DESIGN SPECIFICATION

It is the responsibility of the PE certifying the Owner's Design Specification (Divisions 1 and 2) or the N3 Certificate Holder's Design Specification (Division 3) to assure that the Design Specification is correct, complete, and in compliance with the requirements of the applicable Edition and Addenda of the Code. As a minimum, the certifier of the Design Specification shall assure that

- (a) the function of the item is properly specified
- (b) the design requirements, including identification of the item Design and Service Loadings or Operating Conditions and their combinations and associated Limits, are properly specified
- (c) the proper environmental conditions, including corrosion, erosion, and radiation, are specified
- (d) the Code classification is properly specified
- (e) the definition of the specific boundaries and load conditions on these boundaries for each item is specified, and that the boundaries and associated load conditions between adjacent components and structure are compatible with the overall system design
- (f) the specified materials for items covered by the Code are permitted by the Code for the applicable item
- (g) all requirements with regard to impact testing are specified
- (h) any restrictions on or additional requirements for heat treating are specified
- (i) any restrictions on cladding materials are specified
- (j) any reduction to design stress intensity values, allowable stress, or fatigue curves necessitated by the given environmental conditions are specified
- (k) the necessary information concerning the load is given carrying capacity of structures supporting Code items
- (l) when operability of a component is a requirement, the Design Specification shall make reference to other appropriate documents that specify the operating requirements
- (m) the overpressure protection requirements are specified

(n) the Code Edition, Addenda, and Code Cases to be used for construction are specified

XXIII-1330 CERTIFICATION OF THE DESIGN REPORT

It is the responsibility of the PE certifying the Design Report to assure that the design of the item complies with the requirements of the applicable Edition and Addenda of the Code for the Design, Service Loadings or Operating Conditions, and Test Loadings that have been specified in the Design Specification. As a minimum, the certifier of the Design Report shall assure that

(a) the Design Report reflects the design as shown by the drawings used for construction and that all modifications to the drawings and construction deviations have been reconciled with the Design Report

(b) the design as shown by the drawings is in accordance with the requirements of the Code

(c) the Design Report is in accordance with the requirements of the Code

(d) materials specified for Code items are permitted by the Code, and that any reduction of material impact properties from heat treatments, welding, and forming have been taken into account

(e) the Design Report is based on the Design, Service Loadings or Operating Conditions, and Test Loadings stated in the Design Specification

(f) the specified requirements for protection against nonductile fracture are specified

(g) all special nondestructive examinations required to validate unique features have been specified in appropriate documents/drawings

(h) the specified test pressure and temperature are in compliance with Code requirements

(i) adequate analytical techniques have been employed to assess the structural adequacy of the item of concern for the Design, Service Loadings or Operating Conditions, and Test Loadings specified

XXIII-1340 CERTIFICATION OF THE OVERPRESSURE PROTECTION REPORT

It is the responsibility of the PE certifying the Overpressure Protection Report to assure that the report has been reconciled with the system requirements and with the requirements of the applicable Subsection of the Code.

XXIII-1350 CERTIFICATION OF THE LOAD CAPACITY DATA SHEET

It is the responsibility of the PE certifying the Load Capacity Data Sheet on behalf of the N Certificate Holder to determine that the load capacity of the component or piping support is rated in accordance with Subsection NF of the Code. He shall assure that the design of the component or piping support complies with the requirements of the applicable Edition and Addenda of the Code for the Design,

Service, and Test Loadings specified in the Design Specification. In addition, his duties shall include the requirements of [XXIII-1330\(a\)](#) through [XXIII-1330\(i\)](#) for the data substantiating the Load Capacity Data Sheet.

XXIII-1360 CERTIFICATION OF THE CONSTRUCTION SPECIFICATION, DESIGN DRAWINGS, OR DESIGN REPORT FOR DIVISION 2

It is the responsibility of the PE certifying the Construction Specification, Design Drawings, or Design Report on behalf of the Designer for Division 2 to assure that each of the above principal Code documents is correct, complete, and in accordance with the Design Specification and Section III, Division 2. As a minimum, the certifier of each of the principal Code documents shall assure

(a) that the Design Drawings contain

(1) concrete and steel liner thicknesses

(2) size and location of reinforcing steel, prestressing tendons, and penetrations

(3) the latest revisions to reflect any change in design.

(b) that the Design Report includes the requirements of [XXIII-1330\(a\)](#) through [XXIII-1330\(i\)](#), as applicable

(c) that the Construction Specification has provided the following in accordance with the Code:

(1) material specifications

(2) material shipping, handling, and storage requirements

(3) requirements for personnel or equipment qualification

(4) material or part examination and testing requirements

(5) acceptance and leakage testing requirements

(6) requirements for shop drawings

(7) requirements for batching, mixing, placing, and curing of concrete

(8) requirements for the fabrication and installation of the prestressing system, reinforcing steel, and embedments

(9) identification of parts requiring a Code stamp

(10) design life for parts and materials where necessary to establish compliance with the Design Specification

(11) construction surveillance to be performed by the Designer as required by the Design Specification

(12) the latest revisions to reflect any change in design

XXIII-1370 CERTIFICATION OF THE FABRICATION SPECIFICATION FOR DIVISION 3

It is the responsibility of the PE certifying the Fabrication Specification on behalf of the N3 Certificate Holder for Division 3 to assure that the Fabrication Specification is correct, complete, and in accordance with the Design Specification, Design Output Documents, and Section III,

Division 3. The certifier of the Fabrication Specification shall assure that it contains sufficient detail to provide a complete basis for fabrication.

As a minimum, the Fabrication Specification shall contain the following:

- (a) material specifications
- (b) material shipping, handling, and storage requirements
- (c) requirements for personnel or equipment qualification
- (d) weld joint design requirements
- (e) fabrication dimensions and tolerances
- (f) identification of Code boundaries

- (g) identification of parts requiring a Code stamp
- (h) material or part examination and testing requirements
- (i) acceptance and leakage testing requirements
- (j) requirements for shop drawings
- (k) design life for parts and materials where necessary to establish compliance with the Design Specification
- (l) fabrication surveillance to be performed by the N3 Certificate Holder as required by the Design Specification
- (m) the latest revisions to reflect any change in design
- (n) requirements for as-built documentation
- (o) a listing of design drawings (by inclusion or reference)

GUIDE A NONMANDATORY SAMPLE STATEMENTS

Form A-1 Design Specification (Div. 1 and 2) CERTIFICATION

I, the undersigned, being a Registered Professional Engineer competent in the applicable field of design and related nuclear power plant requirements relative to this Design Specification, certify that to the best of my knowledge and belief it is correct and complete with respect to the Design and Service Conditions given and provides a complete basis for construction in accordance with NCA-3250 and other applicable requirements of the ASME Boiler and Pressure Vessel Code, Section III, Division _____, _____ Edition with Addenda (if applicable) up to and including _____.

The Specification and Revision being certified is:

Certified by _____ P.E.

Registration No. _____ State* _____

Date _____

* or Province of Canada

**Form A-2
Design Report**

CERTIFICATION¹

I, the undersigned, being a Registered Professional Engineer competent in the applicable field of design and using the certified Design Specification and the drawings identified below as a basis for design, do hereby certify that to the best of my knowledge and belief the Design Report is complete and accurate and complies with the design requirements of the ASME Boiler and Pressure Vessel Code, Section III, Division _____, _____ Edition with Addenda (if applicable) up to and including _____.

Design Specification and Revision:

Drawings and Revision:

Design Report and Revision:

Certified by _____ P.E.

Registration No. _____ State* _____

Date _____

¹ Similar statement may also be used for certification of Load Capacity Data Sheet when supplied in lieu of Design Report (NCA-3551).

* or Province of Canada

**Form A-3
Overpressure Protection Report (Div. 1 and 2)**

CERTIFICATION

I, the undersigned, being a Registered Professional Engineer competent in the applicable field of design and overpressure protection requirements, do hereby certify that to the best of my knowledge and belief the Overpressure Protection Report complies with the requirements of the ASME Boiler and Pressure Vessel Code, Section III, Division _____, _____ Edition with Addenda (if applicable) up to and including _____.

Overpressure Protection Report and Revision:

Design Specification and Revision:

Certified by _____ P.E.

Registration No. _____ State* _____

Date _____

* or Province of Canada

**Form A-4
Design Specification (Div. 3)**

CERTIFICATION

I, the undersigned, being a Registered Professional Engineer competent in the applicable field of Division 3 containment design requirements relative to this Design Specification, certify that to the best of my knowledge and belief it is correct and complete with respect to the Design and Operating Conditions given and provides a complete basis for construction in accordance with WA-3351 and other applicable requirements of the ASME Boiler and Pressure Vessel Code, Section III, Division 3, _____ Edition with Addend (if applicable) up to and including _____

The Specification and Revision being certified is:

Certified by _____ P.E.

Registration No. _____ State* _____

Date _____

* or Province of Canada

Form A-5
Fabrication Specification (Div. 3)
CERTIFICATION

I, the undersigned, being a Registered Professional Engineer competent in the applicable field of Division 3 containment fabrication requirements relative to this Fabrication Specification, certify that to the best of my knowledge and belief it is correct and complete with respect to the Design and Operating Conditions given and provides a complete basis for construction in accordance with WA-3361 and other applicable requirements of the ASME Boiler and Pressure Vessel Code, Section III, Division 3, _____ Edition with Addenda (if applicable) up to and including _____.

Design Specification and Revision:

Fabrication Specification and Revision:

Certified by _____ P.E.

Registration No. _____ State* _____

Date _____

* or Province of Canada

GUIDE B NONMANDATORY GUIDELINES FOR ESTABLISHING ASME CODE KNOWLEDGE

This Guide provides guidelines for establishing the degree of Code knowledge required by the certifying PE in his specialty field. In the paragraphs that follow, the degree of knowledge required by the PE of the requirements of the Code pertaining to his specialty field is indicated by the terminology “general knowledge” and “working knowledge.”

As used in this Guide, “general knowledge” signifies having sufficient acquaintance with the Code to be conversant with other persons involved in its applications, and to make prudent judgements in the application of Code requirements.

As used in this Guide, “working knowledge” signifies understanding by prior customary involvement in a specialty field of the Code requirements and of the principles on which the Code rules are based, to the extent that the PE may apply or direct others in the application of the requirements.

In this sense, “working knowledge” implies a more thorough understanding of the Code requirements and the ability to apply them than does “general knowledge” of the Code.

The degree of knowledge in the various areas of the Code cited in [Tables B1](#) through [B5](#) is based upon the more common Code items and activities. There may be special items or activities for which the degree of knowledge in a specific Code area must be more detailed than shown in the applicable table, or may require knowledge of specific Code areas that are not cited.

In the following tables, the degree of knowledge required by the PE of the various requirements of the Code pertinent to his specialty field is indicated by the letter G for “general knowledge” and the letter W for “working knowledge.”

Table B1
Design Specification — Divisions 1 and 2

NCA-1000	W	NCA-8000	G
NCA-2000	W	NX-1000	W
NCA-3100	W	NX-2100	G
NCA-3200	W	NX-2300	G
NCA-3300	W [Note (1)]	NX-2500	G
NCA-3400	W [Note (1)]	NX-2600	G
NCA-3500	W [Note (1)]	NX-3100	G
NCA-3600	W	NX-3200	G
NCA-3700	G	NX-3300	G [Note (1)]
NCA-3800	G	NX-3400	G [Note (1)]
NCA-3900	G [Note (1)]	NX-3500	G [Note (1)]
NCA-4000	G	NX-3600	G [Note (1)]
NCA-5000	G	NX-3700	G [Note (1)]

Table B1
Design Specification — Divisions 1 and 2
(Cont'd)

NX-3800	G [Note (1)]	NF-NG-2300	G [Note (1)]
NX-3900	G [Note (1)]	NF-NG-3000	G [Note (1)]
NX-4100	G	NF-NG-4100	G [Note (1)]
NX-4210	G	NF-NG-4200	G [Note (1)]
NX-4220	G	NF-NG-5000	G [Note (1)]
NX-4240	G		
NX-4620	G	CC-1000	W
NX-5100	G	CC-2000	G
NX-5200	G	CC-3000	G
NX-6000	W [Note (1)]	CC-4000	G
NF-NG-1000	W [Note (1)]	CC-5000	G
NF-NG-2100	G [Note (1)]	CC-6000	G

Legend:

G = General Knowledge

NX = NB/NC/ND/NE/NH, as applicable

W = Working Knowledge

NOTE:

(1) As applicable.

Table B2
Design Report — Division 1

NCA-1000	G	NX-3400	W
NCA-2000	G	NX-3500	W [Note (1)]
NCA-3100	G	NX-3600	W [Note (1)]
NCA-3200	G	NX-3700	W [Note (1)]
NCA-3300	W [Note (1)]	NX-3800	W [Note (1)]
NCA-3400	W [Note (1)]	NX-3900	W [Note (1)]
NCA-3500	W [Note (1)]	NX-4100	G
NCA-3600	G	NX-4210	G
NCA-3700	G	NX-4220	G
NCA-3800	G	NX-4240	G
NCA-3900	G	NX-4620	G
NCA-4000	G	NX-5100	G
NCA-8000	G	NX-5200	G
		NX-6000	G
NX-1000	W		
NX-2100	W	NF-NG-1000	W [Note (1)]
NX-2300	W	NF-NG-2100	W [Note (1)]
NX-2500	W [Note (1)]	NF-NG-2300	W [Note (1)]
NX-2600	G	NF-NG-3000	W [Note (1)]
NX-3100	W	NF-NG-4100	G [Note (1)]
NX-3200	W [Note (1)]	NF-NG-4200	G [Note (1)]
NX-3300	W [Note (1)]	NF-NG-5000	G [Note (1)]

Legend:

G = General Knowledge

NX = NB/NC/ND/NE/NH, as applicable

W = Working Knowledge

NOTE:

(1) As applicable.

Table B3
Overpressure Protection Report — Divisions
1 and 2

NCA-1000	G	NX-3521	G
NCA-2000	W	NX-3621	G
NCA-3100	G	NX-6200	G
NCA-3200	G	NX-6300	G
NCA-3500	G	NX-7000	W
NCA-3600	G		
NCA-4000	G	CC-1000	G
		CC-3100	G
		CC-3200	W
NX-1000	G	CC-6100	G
NX-3110	W	CC-6211	G
NX-3220	G	CC-7000	W
NX-3230	G		
NX-3414	G		

Legend:

G = General Knowledge

NX = NB/NC/ND/NE/NH, as applicable

W = Working Knowledge

Table B4
Load Capacity Data Sheet — Division 1

NCA All	G	NF-3600	W [Note (1)]
NCA-1250	W	Appendix	
NCA-2140	W	I	G
NCA-3550	W	II	G [Note (1)]
		II-1220	W [Note (1)]
NF All	G	II-1430	W [Note (1)]
NF-3100	W	F	G
NF-3200	W	F-1321	W
NF-3300	W [Note (1)]	F-1370	W
NF-3400	W		
NF-3500	W [Note (1)]		

Legend:

G = General Knowledge

W = Working Knowledge

NOTE:

(1) As applicable

Table B5
Construction Specification, Design Drawings, and Design Report — Division 2

	Construction Specification	Design Drawings	Design Report		Construction Specification	Design Drawings	Design Report
NCA-1000	W	W	W	CC-3300	G	W	W
NCA-2000	W	W	W				
NCA-3100	W	W	W	CC-3400	G	W	W
NCA-3200	W	W	W				
NCA-3300	W	W	W	CC-3500	W	W	W
NCA-3400	W	W	W				
NCA-3500	G	G	G	CC-3600	W	W	W
NCA-3600	G	G	G				
NCA-3700	G	G	G	CC-3700	W	W	W
NCA-3800	W	G	G				
NCA-3900	W	G	G	CC-3800	W	W	W
NCA-4000	W	G	W				
NCA-5000	G	G	G	CC-4100	W	W	W
NCA-8000	G	G	G				
				CC-4200	W	W	G
CC-1000	W	W	W				
				CC-4300	W	W	G
CC-2000	W	G	W				
				CC-4400	W	W	G
CC-3100	W	W	W				
				CC-4500	W	W	G
CC-3200	G	G	W	CC-5000	W	W	W
				CC-6000	W	G	G
CC-3300	G	W	W	CC-7000	W	G	G
				CC-8000	W	G	G
CC-3200	G	G	W				

Legend:

G = General Knowledge

W = Working Knowledge

Table B6
Design Specification — Division 3

WA-1000	W	WA-8000	W
WA-2000	W	WX-1000	W
WA-3100	W	WX-2000	G
WA-3300	W	WX-3000	G
WA-3400	W	WX-4000	G
WA-3800	G	WX-5000	G
WA-4000	W	WX-6000	W
WA-5000	G		

Legend:

G = General Knowledge

W = Working Knowledge

WX = WB and WC, as applicable

Table B7
Design Report — Division 3 (Cont'd)

WA-4000	W	WX-3000	W
WA-5000	G	WX-4000	G
WA-8000	G	WX-5000	G
WX-1000	W	WX-6000	W
WX-2000	W		

Legend:

G = General Knowledge

W = Working Knowledge

WX = WB and WC, as applicable

Table B8
Fabrication Specification — Division 3

WA-1000	W	WA-4000	W
WA-2000	W	WA-5000	G
WA-3100	W	WA-8000	W
WA-3300	W	WX-1000	W
WA-3400	W	WX-2000	W
WA-3800	W	WX-3000	W

Table B8
Fabrication Specification — Division 3
(Cont'd)

WX-4000	W		
WX-5000	W	WX-6000	W

Legend:

G = General Knowledge

W = Working Knowledge

WX = WB and WC, as applicable

GUIDE C NONMANDATORY GUIDELINES FOR DEMONSTRATING PE QUALIFICATIONS

This Guide provides suggested methods for demonstrating that the requirements for the qualification of personnel engaged in certification activities have been met. The Owner, Designer, N Certificate Holder, or N3 Certificate Holder, as applicable, responsible for the activity being certified should establish procedures or instructions for evaluating, verifying, and documenting the qualifications of the PE engaged in certifying activities as required by this Appendix.

The PE's qualifications for the requirements of [XXIII-1220](#) may be demonstrated as follows.

(a) PE registration in one or more states of the United States or provinces of Canada should be documented on records that, as a minimum, include

- (1) PE's identification
- (2) state or province of registration
- (3) registration number
- (4) expiration date

(b) The 4 years of varied application experience, including 2 years in his specialty field(s), should be documented in a resume describing the PE's Code experience and places and dates of employment.

(c) In order for the PE to keep current his knowledge of the Code requirements and to continue his professional development in his specialty field(s), as required by this Appendix, he should, in the 36-month period preceding the date of qualification, have performed Code activities requiring certification in his specialty field(s), or have been engaged in the application of Code requirements to an equivalent extent, but not necessarily including Certification. Alternatively, he should have done two or more of the following:

- (1) taught or attended an appropriate course or training program
- (2) taught or attended an appropriate seminar
- (3) attended an ASME or ASME/ACI Code meeting
- (4) attended a technical society meeting related to his specialty field

(d) The PE's participation in these activities should be documented in appropriate records that, as a minimum, include

- (1) PE's identification
- (2) description of Code activities performed
- (3) course or training program description, duration, and date completed
- (4) seminar description, duration, and date attended
- (5) ASME or ASME/ACI Code meeting(s) and date(s) attended
- (6) technical society meeting(s) and date(s) attended
- (7) the PE's function (i.e., attendee, member, speaker, chairman, etc.) indicating the nature of his participation

Guide B provides guidance regarding knowledge of the Code that the PE should have in each specialty field. The PE's qualifications regarding knowledge of the Code may be demonstrated by any one of the following methods:

(e) The Owner, Designer, N Certificate Holder, or N3 Certificate Holder, as applicable, upon review of the experience record of the PE, declares in writing that

- (1) the PE's knowledge of the Code in his specialty field meets the requirements, and
- (2) the PE's experience record reflects successful performance of the applicable Code activities in connection with the construction of ASME Code items.

(f) Another PE previously qualified to this Appendix, designated by the Owner, Designer, N Certificate Holder, or N3 Certificate Holder, as applicable, and familiar with the requirements of the Code, after reviewing the qualifications of the PE to be qualified, attests in writing that the PE's knowledge of the Code in his specialty field(s) meets the requirements of this Appendix.

(g) Attendance of the PE at appropriate courses or seminars that provide instruction in the Code for his specialty field(s) to import knowledge of the Code required by this Appendix. Training should be scheduled as required by this Appendix, at a frequency consistent with significant changes to the Code in his specialty field(s). Training may be accomplished by attending in-house courses or courses presented by others. Training should be documented on appropriate records that, as a minimum, include

- (1) attendee's identification
- (2) instructor's name and affiliation
- (3) outline or description of course or seminar
- (4) date and duration of course or seminar

(h) Examination of the PE in his specialty field(s), either written or oral, to verify his knowledge of the Code as required by this Appendix. The examination may be developed and/or administered either in-house or by others. Examinations should be documented on appropriate records that, as a minimum, include

- (1) attendee's identification
- (2) examiner's name and affiliation
- (3) outline or description of examination
- (4) date and results of the examination

MANDATORY XXIV

STANDARD UNITS FOR USE IN EQUATIONS

Table XXIV-1000
Standard Units for Use in Equations

Quantity	U.S. Customary Units	SI Units
Linear dimensions (e.g., length, height, thickness, radius, diameter)	inches (in.)	millimeters (mm)
Area	square inches (in. ²)	square millimeters (mm ²)
Volume	cubic inches (in. ³)	cubic millimeters (mm ³)
Section modulus	cubic inches (in. ³)	cubic millimeters (mm ³)
Moment of inertia of section	inches ⁴ (in. ⁴)	millimeters ⁴ (mm ⁴)
Mass (weight)	pounds mass (lbm)	kilograms (kg)
Force (load)	pounds force (lbf)	newtons (N)
Bending moment	inch-pounds (in.-lb)	newton-millimeters (N·mm)
Pressure, stress, stress intensity, and modulus of elasticity	pounds per square inch (psi)	megapascals (MPa)
Energy (e.g., Charpy impact values)	foot-pounds (ft-lb)	joules (J)
Temperature	degrees Fahrenheit (°F)	degrees Celsius (°C)
Absolute temperature	Rankine (R)	kelvin (K)
Fracture toughness	ksi square root inches (ksi√in.)	MPa square root meters (MPa√m)
Angle	degrees or radians	degrees or radians
Boiler capacity	Btu/hr	watts (W)

(13)

MANDATORY APPENDIX XXV ASME-PROVIDED MATERIAL STRESS-STRAIN DATA

ARTICLE XXV-1000 INTRODUCTION

XXV-1100 STRESS-STRAIN DATA

It is recognized that ASME-specified material property data would make implementation of the strain-based acceptance criteria easier for many users. However, at this time, the ASME true stress-strain curves (reflecting minimum yield and ultimate tensile strength values) and the true uniform and fracture strain limits (reflecting a 98%

exceedance probability) associated with the strain-based acceptance criteria of [Nonmandatory Appendix FF](#) are still under development. Until these data become available, the user shall develop the necessary material data based on tensile testing (see [Nonmandatory Appendix EE, EE-1222](#)), and their use shall be justified in the final Design Report.

NONMANDATORY APPENDICES

NONMANDATORY APPENDIX A

ARTICLE A-1000 STRESS ANALYSIS METHODS

A-1100 INTRODUCTION

A-1110 SCOPE

(a) The Articles of this Appendix illustrate some acceptable methods of analysis to determine the stresses and stress intensities required to ensure the adequacy of a design as defined in NB-3200.

(b) The methods presented here are not intended to exclude others such as computer programs working directly with shell equations or finite element breakdowns of the component under investigation.

ARTICLE A-2000 ANALYSIS OF CYLINDRICAL SHELLS

A-2100 INTRODUCTION

A-2110 SCOPE

(a) In this Article equations are given for stress and deformations in cylindrical shells subjected to internal pressure only. Refer to NB-3133.3 for cylindrical shells subjected to external pressure.

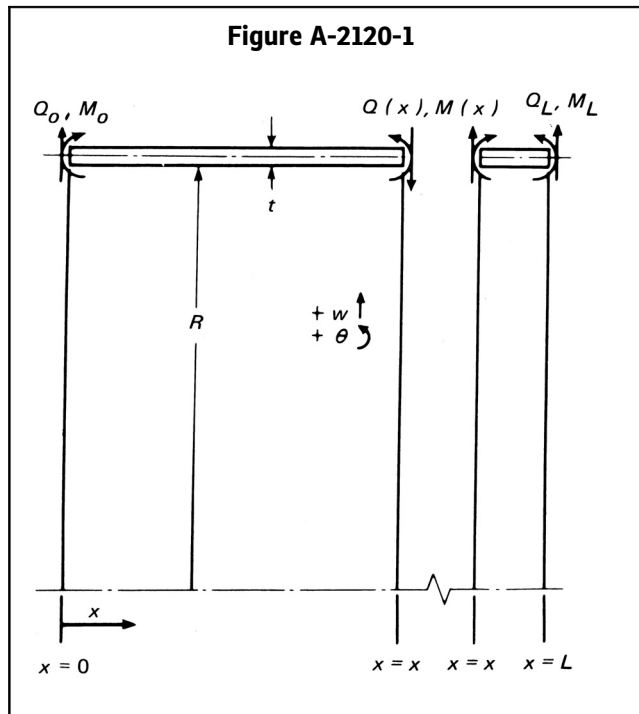
(b) Equations are given for bending analysis for uniformly distributed edge loads.

A-2120 SIGN CONVENTION AND NOMENCLATURE

The sign convention arbitrarily chosen for the analysis of cylindrical shells in this Article is as indicated in Figure A-2120-1. Positive directions assumed for pertinent quantities are indicated.

The symbols and sign convention adopted in this Article for the analysis of cylindrical shells are defined as follows:

$$\begin{aligned} B_{11} &= B_{11}(\beta L) \\ &= (\sinh 2\beta L - \sin 2\beta L)/2(\sinh^2 \beta L - \sin^2 \beta L) \\ B_{12} &= B_{12}(\beta L) \\ &= (\cosh 2\beta L - \cos 2\beta L)/2(\sinh^2 \beta L - \sin^2 \beta L) \\ B_{22} &= B_{22}(\beta L) \end{aligned}$$



$$= (\sinh 2\beta L + \sin 2\beta L)/2(\sinh^2 \beta L - \sin^2 \beta L)$$

$$D = Et^3/12(1 - \nu^2), \text{ in.-lb (N}\cdot\text{mm)}$$

E = modulus of elasticity

$$f_1(\beta x) = e^{-\beta x} \cos \beta x$$

$$F_{11}(\beta x) = (\cosh \beta x \sin \beta x - \sinh \beta x \cos \beta x)/2$$

$$F_{12}(\beta x) = \sinh \beta x \sin \beta x$$

$$F_{13}(\beta x) = (\cosh \beta x \sin \beta x + \sinh \beta x \cos \beta x)/2$$

$$F_{14}(\beta x) = \cosh \beta x \cos \beta x$$

$$f_2(\beta x) = e^{-\beta x} (\cos \beta x - \sin \beta x)$$

$$f_3(\beta x) = e^{-\beta x} (\cos \beta x + \sin \beta x)$$

$$f_4(\beta x) = e^{-\beta x} \sin \beta x$$

$$G_{11} = G_{11}(\beta L)$$

$$= -(\cosh \beta L \sin \beta L - \sinh \beta L \cos \beta L)/(\sinh^2 \beta L - \sin^2 \beta L)$$

$$G_{12} = G_{12}(\beta L)$$

$$= -2 \sinh \beta L \sin \beta L/(\sinh^2 \beta L - \sin^2 \beta L)$$

$$G_{22} = G_{22}(\beta L)$$

$$= -2 (\cosh \beta L \sin \beta L + \sinh \beta L \cos \beta L)/\sinh^2 \beta L - \sin^2 \beta L$$

L = length cylinder used as subscript to denote evaluation of a quantity at end of cylinder removed from reference end

M = longitudinal bending moment per unit length of circumference, in.-lb/in. (N·mm/mm)

o = used as subscript to denote evaluation of a quantity at reference end of cylinder, $x = 0$

p = internal pressure

Q = radial shearing forces per unit length of circumference, lb/in. (N·mm)

R = inside radius

S = stress intensity

t = thickness of cylinder

w = radial displacement of cylinder wall, in. (mm)

x = axial distance measured from the reference end of cylinder

Y = ratio of outside radius to inside radius

Z = ratio of outside radius to an intermediate radius

$$\beta = [3(1 - \nu^2)/(R + t/2)^2 t^2]^{1/4}, \text{ in.}^{-1} (\text{mm}^{-1})$$

θ = rotation of cylinder wall, rad

$$= dw/dx$$

ν = Poisson's ratio

σ_l = longitudinal (meridional) stress component

σ_r = radial stress component

σ_t = tangential (circumferential) stress component

A-2200 STRESS INTENSITIES, DISPLACEMENTS, BENDING MOMENTS, AND LIMITING VALUES

A-2210 PRINCIPAL STRESSES AND STRESS INTENSITIES DUE TO INTERNAL PRESSURE

A-2211 Loading Effects Considered

In this Subarticle equations are given for principal stresses and stress intensities resulting from uniformly distributed internal pressure in cylindrical shells. The effects of discontinuities in geometry and loading are not included and should be evaluated independently. The stresses resulting from all effects shall be combined by superposition.

A-2212 Principal Stresses

The principal stresses developed at any point in the wall of a cylindrical shell due to internal pressure are given by the following equations:

$$\sigma_1 = \sigma_t = p(1 + Y^2) / (Y^2 - 1) \quad (1)$$

$$\sigma_2 = \sigma_l = p / (Y^2 - 1) \quad (2)$$

$$\sigma_3 = \sigma_r = p(1 - Y^2) / (Y^2 - 1) \quad (3)$$

A-2220 STRESS INTENSITIES

A-2221 General Primary Membrane Stress Intensity

The general primary membrane stress intensity developed in a cylindrical shell as a result of internal pressure is given by the equation:

$$S = (pR/t) + (p/2) \quad (4)$$

A-2222 Maximum Value of Primary Plus Secondary Stress Intensity

The maximum value of the primary plus secondary stress intensity in a cylindrical shell as a result of internal pressure occurs at the inside surface and is given by the equation:

$$S = 2pY^2 / (Y^2 - 1) \quad (5)$$

A-2223 Values of Radial Stress Used

Note that in evaluating the general primary membrane stress intensity, the average value of the radial stress has been taken as $-p/2$. This has been done to obtain a result consistent with burst pressure analyses. On the other hand, the radial stress value used in A-2222 is $-p$, the value at the inner surface, since the purpose of that quantity is to control local behavior.

A-2230 BENDING ANALYSIS FOR UNIFORMLY DISTRIBUTED EDGE LOADS

A-2231 Behavior of Shells Subjected to Bending Moments

The equations in this Subarticle describe the behavior of cylindrical shells when subjected to the action of bending moments M , in.-lb/in. (N·mm/mm) of circumference, and radial shearing forces Q , lb/in. (N/mm) of circumference, uniformly distributed at the edges and acting at the mean radius of the shell. The behavior of shells due to all other loadings must be evaluated independently and combined by superposition.

A-2240 DISPLACEMENTS, BENDING MOMENTS, AND SHEARING FORCES IN TERMS OF CONDITIONS AT REFERENCE EDGE ($x = 0$)

A-2241 Equations for Conditions of Any Axial Location

The radial displacement $w(x)$, the angular displacement or rotation $\theta(x)$, the bending moments $M(x)$, and the radial shearing forces $Q(x)$ at any axial location of the cylinder are given by the following equations in terms of w_o , θ_o , M_o , and Q_o .

$$w(x) = (Q_o / 2\beta^3 D) F_{11}(\beta x) + (M_o / 2\beta^2 D) F_{12}(\beta x) + (\theta_o / \beta) F_{13}(\beta x) + w_o F_{14}(\beta x) \quad (6)$$

$$\begin{aligned} \theta(x) / \beta &= (Q_o / 2\beta^3 D) F_{12}(\beta x) + \\ &2(M_o / 2\beta^2 D) F_{13}(\beta x) + (\theta_o / \beta) F_{14}(\beta x) - \\ &2w_o F_{11}(\beta x) \end{aligned} \quad (7)$$

$$\begin{aligned} M(x) / 2\beta^2 D &= (Q_o / 2\beta^3 D) F_{13}(\beta x) + \\ &(M_o / 2\beta^2 D) F_{14}(\beta x) - (\theta_o / \beta) F_{11}(\beta x) - w_o F_{12}(\beta x) \end{aligned} \quad (8)$$

$$\begin{aligned} Q(x) / 2\beta^3 D &= (Q_o / 2\beta^3 D) F_{14}(\beta x) - \\ &2(M_o / 2\beta^2 D) F_{11}(\beta x) - (\theta_o / \beta) F_{12}(\beta x) - \\ &2w_o F_{13}(\beta x) \end{aligned} \quad (9)$$

A-2242 Equations When Cylinder Length $\geq 3/\beta$

In the case of cylinders of sufficient length, the equations in A-2241 reduce to those given below. These equations may be used for cylinders characterized by lengths not less than $3/\beta$. The combined effects of loadings at the two edges may be evaluated by applying the equations to the loadings at each edge, separately, and superposing the results.

$$w(x) = (Q_o / 2\beta^3 D) f_1(\beta x) + (M_o / 2\beta^2 D) f_2(\beta x) \quad (10)$$

$$\theta(x) / \beta = - (Q_o / 2\beta^3 D) f_3(\beta x) - 2(M_o / 2\beta^2 D) f_1(\beta x) \quad (11)$$

$$M(x) / 2\beta^2 D = (Q_o / 2\beta^3 D) f_4(\beta x) + (M_o / 2\beta^2 D) f_3(\beta x) \quad (12)$$

$$Q(x) / 2\beta^3 D = (Q_o / 2\beta^3 D) f_2(\beta x) - 2(M_o / 2\beta^2 D) f_4(\beta x) \quad (13)$$

A-2243 Edge Displacements and Rotations in Terms of Edge Loads

The radial displacement w_o and w_L and rotations θ_o and $-\theta_L$, developed at the edges of a cylindrical shell sustaining the action of edge loads Q_o , M_o , Q_L , and M_L , are given by the following equations:

$$w_o = (B_{11} / 2\beta^3 D) Q_o + (B_{12} / 2\beta^2 D) M_o + (G_{11} / 2\beta^3 D) Q_L + (G_{12} / 2\beta^2 D) M_L \quad (14)$$

$$-\theta_o = (B_{12} / 2\beta^2 D) Q_o + (B_{22} / \beta D) M_o + (G_{12} / 2\beta^2 D) Q_L + (G_{22} / 2\beta D) M_L \quad (15)$$

$$w_L = (G_{11} / 2\beta^3 D) Q_o + (G_{12} / 2\beta^2 D) M_o + (B_{11} / 2\beta^3 D) Q_L + (B_{12} / 2\beta^2 D) M_L \quad (16)$$

$$\theta_L = (G_{12} / 2\beta^2 D) Q_o + (G_{22} / 2\beta D) M_o + (B_{12} / 2\beta^2 D) Q_L + (B_{22} / \beta D) M_L \quad (17)$$

A-2250 LIMITING VALUE OF FUNCTIONS

A-2251 General Limiting Values of Influence Functions

The influence functions B and G , appearing in the equations in A-2243, rapidly approach limiting values as the length L of the cylinder increases. The limiting values are

$$B_{11} = B_{12} = B_{22} = 1$$

$$G_{11} = G_{12} = G_{22} = 0$$

(a) Thus, for cylindrical shells of sufficient length, the loading conditions prescribed at one edge do not influence the displacements at the other edge.

(b) In the case of cylindrical shells characterized by lengths not less than $3/\beta$, the influence functions B and G , are sufficiently close to the limiting values so that the limiting values may be used in the equations in A-2243 without significant error.

A-2252 Limiting Values of Influence Functions for Short Cylinders

In the case of sufficiently short cylinders, the influence functions B and G , appearing in the equations in A-2243, are, to a first approximation, given by the following expressions:

$$B_{11} = 2 / \beta L$$

$$B_{12} = 3 / (\beta L)^2$$

$$B_{22} = 3 / (\beta L)^3$$

$$G_{11} = -1 / \beta L$$

$$G_{12} = 3 / (\beta L)^2$$

$$G_{22} = -6 / (\beta L)^3$$

Introducing these expressions for the influence functions B and G into the equations in A-2243 yields expressions identical to those obtained by the application of ring theory. Accordingly, the resultant expressions are subject to all of the limitations inherent in the ring theory, including the limitations due to the assumption that the entire cross-sectional area of the ring $t \times L$ rotates about its centroid without distortion. Nevertheless, in the analysis of very short cylindrical shells characterized by lengths not greater than $1/2\beta$, the expressions may be used without introducing significant error.

A-2260 PRINCIPAL STRESSES DUE TO BENDING

The principal stresses developed at the surfaces of a cylindrical shell at any axial location x due to uniformly distributed edge loads (Figure A-2120-1) are given by the equations:

$$\sigma_1 = \sigma_t(x) = Ew(x) / (R + t/2) \pm 6\nu M(x) / t^2 \quad (18)$$

$$\sigma_2 = \sigma_l(x) = \pm 6M(x) / t^2 \quad (19)$$

$$\sigma_3 = \sigma_r = 0 \quad (20)$$

In these equations, where terms are preceded by a double sign \pm , the upper sign refers to the inside surface of the cylinder and the lower sign refers to the outside surface.

ARTICLE A-3000 ANALYSIS OF SPHERICAL SHELLS

A-3100 INTRODUCTION

A-3110 SCOPE

(a) In this Article equations are given for stresses and deformations in spherical shells subjected to internal or external pressure.

(b) Equations are also given for bending analysis of partial spherical shells under the action of uniformly distributed edge forces and moments.

A-3120 NOMENCLATURE AND SIGN CONVENTION

(a) The symbols and sign convention adopted in this Article are defined as follows:

$$A_o = \sqrt{1 + k_1^2}$$

$$B(\alpha) = [(1 + \nu^2) (K_1 + K_2) - 2K_2]$$

$$C(\alpha) = \sqrt{\sin \phi_o / \sin (\phi_o - \alpha)}$$

$$D = Et^3 / 12(1 - \nu^2), \text{ flexural rigidity, in.-lb (N}\cdot\text{mm)}$$

$$E = \text{modulus of elasticity}$$

$$F(\alpha) = \sqrt{\sin (\phi_o) \sin (\phi_o - \alpha)}$$

$$H = \text{force per unit length of circumference, perpendicular to center line of sphere, lb/in. (N}\cdot\text{mm)}$$

$$K_1 = 1 - [(1 - 2\nu) / 2\lambda] \cot (\phi_o - \alpha)$$

$$k_1 = 1 - [(1 - 2\nu) / 2\lambda] \cot \phi_o$$

$$K_2 = 1 - [(1 + 2\nu) / 2\lambda] \cot (\phi_o - \alpha)$$

$$k_2 = 1 - [(1 + 2\nu) / 2\lambda] \cot \phi_o$$

$$l = \text{used as a subscript to denote meridional direction}$$

$$M = \text{meridional bending moment per unit length of circumference, in.-lb/in. (N}\cdot\text{mm/mm)}$$

$$N = \text{membrane force, lb/in. (N}\cdot\text{mm)}$$

$$o = \text{used as a subscript to denote a quantity at the reference edge of sphere}$$

$$p = \text{uniform pressure internal or external}$$

$$Q = \text{radial shearing force per unit of circumference, lb/in. (N}\cdot\text{mm)}$$

$$R = \text{inside radius}$$

$$R_m = \text{radius of midsurface of spherical shell}$$

$$S = \text{stress intensity}$$

$$t = \text{thickness of spherical shell}$$

$$\text{used as a subscript to denote circumferential direction}$$

$$U = \text{ratio of inside radius to an intermediate radius}$$

$$w = \text{radial displacement of midsurface, in. (mm)}$$

$$Y = \text{ratio of outside radius to inside radius}$$

$$Z = \text{ratio of outside radius to an intermediate radius}$$

$$\alpha = \text{meridional angle measured from the reference edge, rad}$$

$$\beta = [3(1 - \nu^2) / R_m^2 t^2]^{1/4}, \text{ in.}^{-1} \text{ (mm}^{-1}\text{)}$$

$$\gamma_o = \tan^{-1} (-k_1)$$

$$\delta = \text{lateral displacement of midsurface, perpendicular to center line of spherical shell, in. (mm)}$$

$$\theta = \text{rotation of midsurface, rad}$$

$$\lambda = \beta R_m$$

$$\nu = \text{Poisson's ratio}$$

$$\sigma_l = \text{longitudinal (meridional) stress component}$$

$$\sigma_r = \text{radial stress component}$$

$$C + 2h_C$$

and the outside diameter of the flanges

$$C + 2h_{C_{\max}}$$

(c) The hub-flange interaction moment M_S , which acts on the flange, is expressed by [equations L-3242\(13\)](#), [L-3244\(a\)\(25\)](#), and [L-3244\(a\)\(26\)](#); for Category 3 flanges

$$M_S = 0$$

The contact force H_C is determined by [equation L-3242\(15\)](#) or [equation L-3244\(a\)\(33\)](#).

(d) The required bolt load for operating conditions is determined in accordance with the following equation:

$$W_{m1} = H + H_C + H_G$$

L-3222 Total Required and Actual Bolt Areas, and Flange Design Bolt Load

The total required cross-sectional area of bolts A_m equals W_{m1}/S_b . A selection of bolts to be used shall be made such that the actual total cross-sectional area of bolts A_b will not be less than A_m . The flange design bolt load W shall be taken equal to W_{m1} .

L-3230 CLASSIFICATION OF ASSEMBLIES AND CATEGORIZATION OF INDIVIDUAL FLANGES

It is necessary to classify the different types of flanged assemblies and to further categorize each flange which comprises the assembly under consideration.

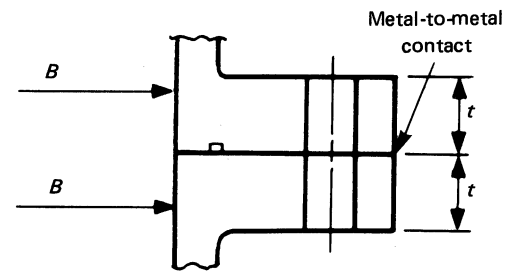
L-3231 Classification of a Class FF Flange Assembly

Since the flanges comprising an assembly are in contact outside the bolt circle, the behavior of one flange is influenced by the stiffness of the other. For the purpose of computation it is helpful to classify an assembly consisting of different types of flanges according to the way the flanges influence the deformation of the assembly.

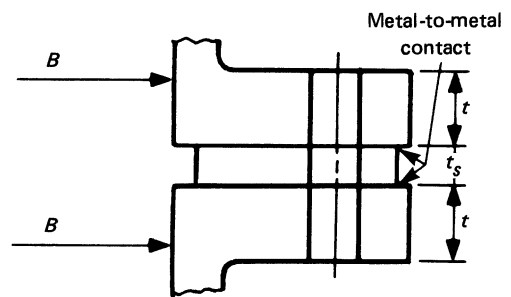
L-3231.1 Group 1 Assembly.³² A pair of flanges which are bolted together and which are nominally identical with respect to shape, dimensions, physical properties, and allowable stresses³³ except that one flange of the pair may contain a gasket groove. (A Group 1 assembly is also referred to as an identical flange pair.) [Figure L-3230-1](#) illustrates configuration of a Group 1 assembly.

L-3231.2 Group 2 Assembly. Any assemblage which does not fit the description of Group 1 where, in the case of reducers, the inside diameter of the reducing flange exceeds one-half of the bolt circle diameter. [Figure L-3230-2](#) illustrates configuration of a Group 2 assembly.

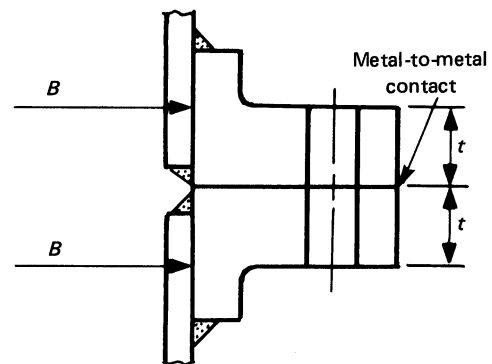
Figure L-3230-1
Group 1 Flange Assembly (Identical Flange Pairs)



(a) Integral Illustrated



(b) Integral With Spacer

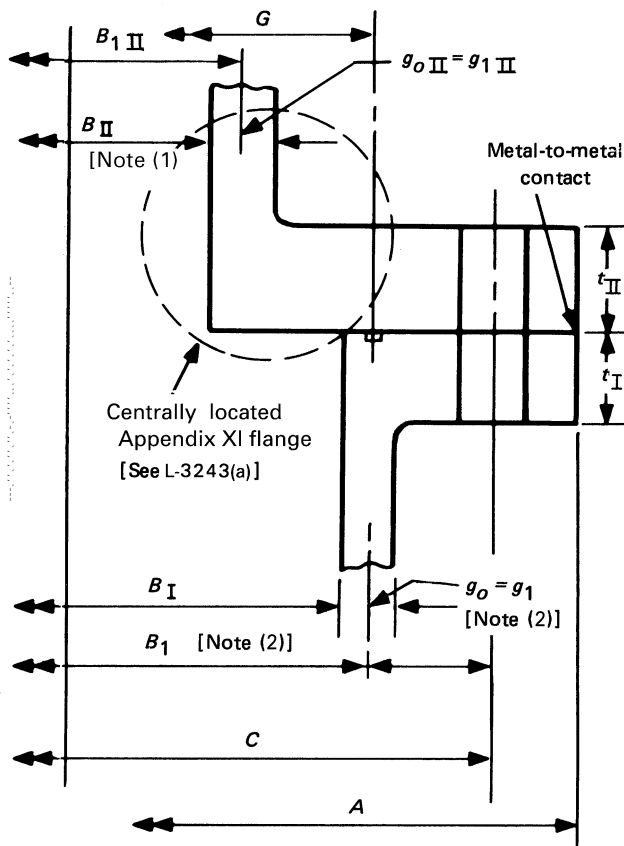


(c) Loose Type With Hub

GENERAL NOTES:

- (a) Category 1 flanges illustrated in sketch (a) and (b); Category 2 flanges illustrated in sketch (c).
- (b) Permitted weld details are in accordance with [Figures XI-3120-1](#) and [NC-4243-2](#), [ND-4243-1](#) or [NE-4243-1](#), as applicable.

Figure L-3230-2
Group 2 Flange Assembly



GENERAL NOTES:

- (a) Category 1 flanges illustrated. Categories II and III permitted.
- (b) For purposes of analysis of Flange II by method L-3243(a), assume $A_{II} = G_{II} = B_1$
- (c) Permitted weld details are in accordance with Figures XI-3120-1 and NC-4243-2, ND-4243-1 or NE-4243-1, as applicable.

NOTES:

- (1) $B_{II} > C/2$
- (2) See L-3192 and L-3193.

L-3231.3 Group 3 Assembly. Any assemblage consisting of a reducer or a flat circular head without an opening or with a central, reinforced opening provided the diameter of the opening in the reducing flange or flat cover is less than one-half of the bolt circle diameter. In the analysis the reducing flange is considered to be the equivalent of a flat circular head without an opening. Figure L-3230-3 illustrates configuration of a Group 3 assembly.

L-3232 Categorization of a Class FF Flange

In addition to classifying an assembly, the individual flanges (except the reducing flange or flat circular head) must be categorized for the purpose of computation as

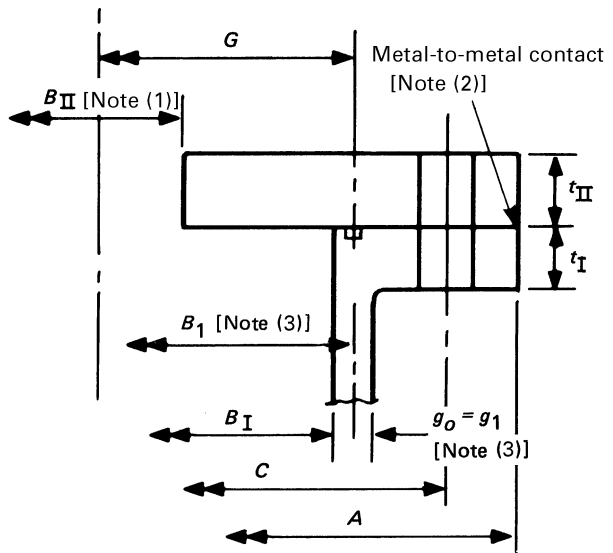
loose type, integral type, or optional type. This can be done using XI-3120; Figure XI-3120-1 is suitable by considering the flanges as flat faced (as a result of removing the raised gasket surface by machining and recessing the gasket in a groove) and by adding a flange-to-flange contact force H_C at some distance h_C outside the bolt circle. Since certain design options exist depending upon the category of the flange, the following categories include both the type of flange and the various design options.

(a) *Category 1 Flange.* An integral flange or an optional flange calculated as an integral flange.

(b) *Category 2 Flange.* A loose type flange with a hub where credit is taken for the strengthening effect of the hub.

(c) *Category 3 Flange.* A loose type flange with a hub where no credit is taken for the strengthening effect of the hub, a loose type flange without a hub, or an optional-type flange calculated as a loose type without a hub. Substitute B for B_1 in the applicable equation for this category of flange.

Figure L-3230-3
Group 3 Flange Assembly



GENERAL NOTE: Category I flange illustrated. Categories II and III permitted.

NOTES:

- (1) $B_{II} \leq C/2$
- (2) Permitted weld details are in accordance with Figures XI-3120 and NC-3225-3.
- (3) See L-3192 and L-3193.

L-3240 FLANGE ANALYSIS**L-3241 General Method**

(a) In order to calculate the stresses in the flanges and bolts of a flanged assembly, classify the assemblage in accordance with L-3231 and then categorize each flange per L-3232.

(b) The method of analyzing various groups and categories of flanges is basically the same. Although many equations appear to be identical, subtle differences do exist and care must be exercised in the analysis. To minimize the need for numerous footnotes and repetitive statements throughout the text, the equations to be used in analyzing the various groups of assemblies and categories of flanges are given in Table L-3240-1. In general, the terms should be calculated in the same order as they are listed in the table. It is important to refer to the table before starting an analysis since only a limited number of the equations contained in this Article are used in the design of a particular pair of flanges. Some of the numbered equations appear in L-3191 along with general purpose, unnumbered expressions.

Table L-3240-1
Summary of Applicable Equations for
Different Groups of Assemblies and Different
Categories of Flanges

Group	Category [Note (1)]	Applicable Formulas
1	1	L-3191(5a), L-3242(13) – L-3242(19), L-3242(20a), L-3242(21a), L-3242(22a)
1	2	L-3191(5b), L-3242(13) – L-3242(19), L-3242(20b), L-3242(21b), L-3242(22b)
1	3	L-3191(5c), L-3242(13) – L-3242(19), L-3242(20c), L-3242(21c), L-3242(22c)
2	All	See L-3243
3	1	L-3191(1) – L-3191(4), L-3191(6a), L-3244(a)(23) – L-3244(a)(37), L-3244(a)(38a), L-3244(a)(39a), L-3244(a)(40a), L-3244(a)(41) – L-3244(a)(44)
3	2	L-3191(1) – L-3191(4), L-3191(6a), L-3244(a)(23) – L-3244(a)(37), L-3244(a)(38b), L-3244(a)(39b), L-3244(a)(40b), L-3244(a)(41) – L-3244(a)(44)
3	3	L-3191(1) – L-3191(4), L-3191(6a), L-3244(a)(23) – L-3244(a)(37), L-3244(a)(38c), L-3244(a)(39c), L-3244(a)(40c), L-3244(a)(41) – L-3244(a)(44)

NOTE:

(1) Of the nonreducing flange in a Group 2 or Group 3 assembly.

(c) Subscripts I and II refer to the nonreducing flange and the reducer (or flat circular head), respectively, of a Group 3 assembly and of a Group 2 assembly designed using the method of L-3243(a).

L-3242 Analysis of a Group 1 Assembly

The following equations are used for the analysis of Category 1, 2, and 3 flanges of a Group 1 assembly in accordance with Table L-3240-1.

Flange Moment Due to Flange–Hub Interaction

$$M_S = - \frac{J_P F' M_P}{t^3 + J_S F'} \quad (13)$$

Slope of Flange at Inside Diameter Times E

$$E\theta_B = \frac{5.46}{\pi t^3} (J_S M_S + J_P M_P) \quad (14)$$

Contact Force Between Flanges at h_C

$$H_C = (M_P + M_S) / h_C \quad (15)$$

Bolt Load at Operating Conditions

$$W_{m1} = H + H_G + H_C \quad (16)$$

Operating Bolt Stress

$$\sigma_b = W_{m1} / A_b \quad (17)$$

Design Prestress in Bolts

$$S_i = \sigma_b - \frac{1.159 h_C^2 (M_P + M_S)}{a t^3 l r_E B_1} \quad (18)$$

Radial Flange Stress at Bolt Circle

$$S_R = \frac{6(M_P + M_S)}{t^2 (\pi C - nD)} \quad (19)$$

Radial Flange Stress at Inside Diameter

$$S_R = - \left(\frac{2Ft}{h_0 + Ft} + 6 \right) \frac{M_S}{\pi B_1 t^2} \quad (20a)$$

$$S_R = - \left(\frac{2F_L t}{h_0 + F_L t} + 6 \right) \frac{M_S}{\pi B_1 t^2} \quad (20b)$$

$$S_R = 0 \quad (20c)$$

Tangential Flange Stress at Inside Diameter

$$S_T = \frac{tE\theta_B}{B_1} + \left(\frac{2FtZ}{h_0 + Ft} - 1.8 \right) \frac{M_S}{\pi B_1 t^2} \quad (21a)$$

$$S_T = \frac{tE\theta_B}{B_1} + \left(\frac{2F_L tZ}{h_0 + F_L t} - 1.8 \right) \frac{M_S}{\pi B_1 t^2} \quad (21b)$$

$$S_T = \frac{tE\theta_B}{B_1} \quad (21c)$$

Longitudinal Hub Stress

$$S_H = \frac{h_0 E \theta_B f}{0.91 (g_1 / g_0)^2 B_1 V} \quad (22a)$$

$$S_H = \frac{h_0 E \theta_B}{0.91 (g_1 / g_0)^2 B_1 V_L} \quad (22b)$$

$$S_H = 0 \quad (22c)$$

L-3243 Analysis of a Group 2 Assembly

(a) The assembly may be analyzed using a variation of the analysis for a Group 3 assembly (L-3244) that accounts for the interaction of nonidentical flanges and the stiffening effect of an integral nozzle or hub centrally located in the reducing flange.

(1) The central nozzle of Flange II with diameter B_{II} shall be assumed for analysis purposes as an [Mandatory Appendix XI](#) flange with outside diameter A , bolt circle C , and gasket circle G all equal to B_1 of Flange I. See [Figure L-3230-2](#).

(2) In addition it is necessary to categorize the centrally located [Mandatory Appendix XI](#) flange (nozzle plus the associated over plate to diameter B_1) as a Category 1, 2, or 3 flange in accordance with L-3232.

(3) The moment due to pressure shall be designated M_p' where

$$M_p' = H_D' h_D' + H_T' h_T'$$

$$H_D' = 0.785 B_{II}^2 P$$

$$H_T' = 0.785 P (B_1^2 - B_{II}^2)$$

For Category 1, 2, or 3 flanges [see (2) above],

$$h_T' = \frac{B_1 - B_{II}}{4}$$

For Category 1 or 2 flanges [see (2) above],

$$h_D' = \frac{B_1 - B_{II} - g_{1II}}{2}$$

For Category 3 flanges [see (2) above],

$$h_D' = \frac{B_1 - B_{II}}{2}$$

(4) The rules in L-3244 and the summary of [Table L-3240-1](#) for the analysis of a Group 3 assembly apply to the analysis of a Group 2 assembly with the following additions and substitutions:

C_5 and C_6 and all the symbols in equations in (-a) and (-b) below pertain only to the centrally located [Mandatory Appendix XI](#) flange [nozzle plus the associated cover of thickness t_{II} to diameter B_1 defined in (1) above]. All terms in equations in (-c) and (-d) below, except C_5 and C_6 , refer to the nonreducing flange (Flange I). C_1 and C_2 of equations in (-c) and (-d) below replace C_1 and C_2 of eqs. L-3191(1) and L-3191(2).

(-a) Let

$$C_5 = M_p'$$

(-b) Let

$$C_6 = \frac{0.829}{\log(B_1 / B_{1II})}$$

for Category 3 flanges.³⁴ Let

$$C_6 = \frac{0.91 t_{II}^3 V}{L h_0 g_0^2}$$

for Category 1 or 2 flanges.³⁴

(-c) Let

$$C_1 = [1 - 2.095 J_S \log(A / B_1)] \div [-C_6 - 1.738 J_S]$$

(-d) Let

$$C_2 = (1.738 J_P M_p - C_5 C_6) \div (-C_6 - 1.738 J_S)$$

(-e) Replace eq. L-3244(a)(32) with:

$$E_{II} \theta_{BII} = \frac{5.46}{\pi t_{II}^3} (J_S M_{bII} + J_P M_p) + (E_{II}^* \theta_{rbII}) / t_{II}^3$$

(-f) Delete eq. L-3244(a)(44). Subparagraphs (1), (2), and (3) above apply only for calculating $C_5(M_p')$ and C_6 , and subsequently when using (5) below for calculating the stresses in and adjacent to the nozzle in Flange II.

(5) Stresses in the centrally located nozzle of Flange II shall be calculated in accordance with the following equations after M_{SII} has been found using (4) above. All terms, such as e , Y , and Z , apply to the centrally located [Mandatory Appendix XI](#) flange as defined in (1) and (2) above.

For Category 1 or 2 flanges [(2) above]:

Longitudinal Hub Stress

$$S_{HII} = \frac{f(M_p' - M_{SII})}{L g_{1II}^2 B_{II}}$$

Radial Flange Stress Adjacent to Central Nozzle

$$S_{RII} = \frac{(1.33t_{II}e + 1)(M_p' - M_{SII})}{Lt_{II}^2 B_{II}}$$

Tangential Flange Stress Adjacent to Central Nozzle

$$S_{TII} = \frac{Y(M_p' - M_{SII})}{t_{II}^2 B_{II}} - ZS_{RII}$$

For Category 3 Flanges [(a)(2) above]:

Tangential Flange Stress Adjacent to Central Nozzle

$$S_{TII} = \frac{Y(M_p' - M_{SII})}{t_{II}^2 B_{II}}$$

Radial and Longitudinal Hub Stress

$$S_{RII} = 0$$

$$S_{HII} = 0$$

(6) The stresses in Flange I and the remaining stresses in Flange II shall be calculated in accordance with L-3244 except as modified by (4).

(b) As an alternative to the method in (a) above and at the option of the designer, the assembly may be analyzed as if it is one flange of an identical pair in a Group 1 assembly using the procedure in L-3242. All stresses shall satisfy L-3250. The same value of h_c shall be used in both calculations and the strain length l of the bolts shall be based on the thickness of the flange under consideration. This method is more conservative and more bolting may be required than the method in (a) above.

(c) The central nozzle or opening in Flange II of a Group 2 assembly determined by the rules in (a) or (b) above meets the general requirements of this Division and of this Article. The rules for determining thickness and reinforcing requirements of NC-3225, NC-3325, ND-3325, and NE-3325, and NC-3233, NC-3333, ND-3333, and NE-3333, respectively, are not applicable.

L-3244 Analysis of a Group 3 Assembly

(a) The following equations are used for the analysis of Category 1, 2, and 3 nonreducing flanges and the reducer (or flat circular head) of a Group 3 assembly:

Rigid Body Rotation of Flanges Times E^*

$$E_I^* \theta_{rBI} = \frac{X(C_4 - C_2)}{1.206 \log(A/B_1) - XC_3 - (1 - X)C_1} \quad (23)$$

$$E_{II}^* \theta_{rBII} = -E_I^* \theta_{rBI} (E_{II}^* / E_I^*) \quad (24)$$

Total Flange Moment at Diameter B_1

$$M_{SI} = C_3(E_I^* \theta_{rBI}) + C_4 \quad (25)$$

$$M_{SII} = C_1(E_{II}^* \theta_{rBII}) + C_2 \quad (26)$$

Unbalanced Flange Moment at Diameter B_1

$$M_{uI} = 1.206 E_I^* \theta_{rBI} \log(A/B_1) \quad (27)$$

$$M_{uII} = 1.206 E_{II}^* \theta_{rBII} \log(A/B_1) \quad (28)$$

Balanced Flange Moment at Diameter B_1

$$M_{bI} = M_{SI} - M_{uI} \quad (29)$$

$$M_{bII} = M_{SII} - M_{uII} \quad (30)$$

Slope of Flange at Diameter B_1 Times E

$$E_I \theta_{BI} = \frac{5.46}{\pi t_I^3} (J_S M_{bI} + J_P M_P) + E_I^* \theta_{rBI} / t_I^3 \quad (31)$$

$$E_{II} \theta_{BII} = \frac{-1.337(M_{SII} - \pi P B_1^3 / 32)}{t_{II}^3} \quad (32)$$

Contact Force Between Flanges at h_c

$$H_C = (M_P + M_{bI}) / h_C \quad (33)$$

Bolt Load at Operating Conditions

$$W_{m1} = H + H_G + H_C \quad (34)$$

Operating Bolt Stress

$$\sigma_b = W_{m1} / A_b \quad (35)$$

Design Prestress in Bolts

$$S_i = \sigma_b - \frac{1.159 h_C^2 (M_P + M_{bI})}{2(1 - X) a t_I^3 l r_{E1} B_1} \quad (36)$$

Radial Stress in Flange I at Bolt Circle

$$S_{RI} = \frac{6(M_P + M_{SI})}{t_I^2 (\pi C - nD)} \quad (37)$$

Radial Stress in Flange I at Inside Diameter

$$S_{RI} = - \left(\frac{2F t_I}{h_0 + F t_I} + 6 \right) \frac{M_{SI}}{\pi B_1 t_I^2} \quad (38a)$$

$$S_{RI} = - \left(\frac{2F_L t_I}{h_0 + F_L t_I} + 6 \right) \frac{M_{SI}}{\pi B_1 t_I^2} \quad (38b)$$

$$S_{RI} = 0 \quad (38c)$$

Tangential Stress in Flange I at Inside Diameter

$$S_{TI} = \frac{t_I E_I \theta_{BI}}{B_1} + \left(\frac{2F t_I Z}{h_0 + F t_I} - 1.8 \right) \frac{M_{SI}}{\pi B_1 t_I^2} \quad (39a)$$

$$S_{TI} = \frac{t_1 E_1 \theta_{BI}}{B_1} + \left(\frac{2 F_L t_1 Z}{h_0 + F_L t_1} - 1.8 \right) \frac{M_{SI}}{\pi B_1 t_1^2} \quad (39b)$$

$$S_{TI} = \frac{t_1 E_1 \theta_{BI}}{B_1} \quad (39c)$$

Longitudinal Hub Stress in Flange I

$$S_{HI} = \frac{h_0 E_1 \theta_{BI} f}{0.91 (g_1 / g_0)^2 B_1 V} \quad (40a)$$

$$S_{HI} = \frac{h_0 E_1 \theta_{BI}}{0.91 (g_1 / g_0)^2 B_1 V_L} \quad (40b)$$

$$S_{HI} = 0 \quad (40c)$$

Radial Stress in Flange II at Bolt Circle

$$S_{RII} = \frac{6(M_P + M_{SII})}{t_{II}^2 (\pi C - nD)} \quad (41)$$

Radial Stress in Flange II at Diameter B_1

$$S_{RII} = \frac{6M_{SII}}{\pi B_1 t_{II}^2} \quad (42)$$

Tangential Stress in Flange II at Diameter B_1

$$S_{TII} = \frac{t_{II} E_{II} \theta_{BII}}{B_1} - \frac{1.8 M_{SII}}{\pi B_1 t_{II}^2} \quad (43)$$

Radial and Tangential Stress at Center of Flange II

$$S_{RII} = S_{TII} = \frac{0.3094 P B_1^2}{t_{II}^2} - \frac{6 M_{SII}}{\pi B_1 t_{II}^2} \quad (44)$$

(b) The thickness of Flange II of a Group 3 assembly determined by the above rules shall be used in lieu of the thickness that is determined by NC-3225, NC-3325, ND-3325, and NE-3325. However, any centrally located opening in Flange II shall be reinforced to meet the rules of [Mandatory Appendix XIX](#).

L-3250 ALLOWABLE FLANGE DESIGN STRESSES

The stresses calculated by the above equations, whether tensile or compressive (–), shall not exceed the following values for all groups of assemblies:³⁵

(a) operating bolt stress σ_b not greater than S_b for the design value of S_i

(b) longitudinal hub stress S_H not greater than S_f for Category 1 and 2 cast iron flanges except as otherwise limited by (1) and (2) below and not greater than $1.5 S_f$ for materials other than cast iron

(1) longitudinal hub stress S_H not greater than the smaller of $1.5 S_f$ or $1.5 S_n$ for Category 1 flanges where the pipe or shell constitutes the hub

(2) longitudinal hub stress S_H not greater than the smaller of $1.5 S_f$ or $2.5 S_n$ for integral flanges (Category 1) similar to the [Mandatory Appendix XI](#) flanges shown as [Figure XI-3120-1](#), sketches (6), (6a), and (6b)

(c) radial stress S_R not greater than S_f

(d) tangential stress S_T not greater than S_f

(e) also,

$$(S_H + S_R) / 2$$

not greater than S_f and

$$(S_H + S_T) / 2$$

not greater than S_f

(f) S_R and S_T at the center of the reducing flange in a Group 3 assembly [see [eq. L-3244\(a\)\(44\)](#)] shall not exceed S_f

L-3260 PRESTRESSING THE BOLTS

The design rules of this Article provide for tangential contact between the flanges at h_{Cmax} or some lesser value h_C beyond the bolt circle. As in the case of [Mandatory Appendix XI](#) flanges, a Class FF flange must be designed so that the calculated value of the operating bolt stress σ_b does not exceed S_b . Also, as in the case of [Mandatory Appendix XI](#) flanges, ordinary wrenching techniques without verification of the actual initial bolt stress (assembly stress) is considered to meet all practical needs with control and verification reserved for special applications. For the purposes of this Article, the use of

$$h_C < h_{Cmax}$$

to optimize stresses is considered to be a special application unless it is also shown that all of the requirements of this Article are also satisfied when

$$h_C = h_{Cmax}$$

L-3270 REFERENCES

Additional guidance on the design of flat faced metal-to-metal contact flanges can be found in the following references:

(a) Schneider, R. W., and Waters, E. O., The Background of ASME Code Case 1828: A Simplified Model of Analyzing Part B Flanges, *Journal of Pressure Vessel Technology*, ASME, Vol. 100, No. 2, May 1978, pp. 215–219;

(b) Schneider, R. W., and Waters, E. O., The Application of ASME Code Case 1828, *Journal of Pressure Vessel Technology*, ASME, Vol. 101, No. 1, February 1979, pp. 87–94.

It should be noted that the rules in [Nonmandatory Appendix L](#) were formerly contained in Code Case 1828, A Simplified Method for Analyzing Flat Face Flanges with Metal-to-Metal Contact Outside the Bolt Circle/ Section VIII, Division 1.

NONMANDATORY APPENDIX M

ARTICLE M-1000 RECOMMENDATIONS FOR CONTROL OF WELDING, POSTWELD HEAT TREATMENT, AND NONDESTRUCTIVE EXAMINATION OF WELDS

M-1100 INTRODUCTION

M-1110 SCOPE

This Appendix provides recommendations for control of welding, postweld heat treatment, and nondestructive examination of welds. The purpose of this Appendix is to identify the areas that should be controlled. It should not be construed that this Appendix describes the only controls that are necessary. Other controls may be necessary depending on the specific application.

M-1120 APPLICABILITY

The recommendations contained in this Appendix may be included in procedures, instructions, specifications, process sheets, shop travelers, checklists, drawings, or other documents that are part of the Quality Assurance or Control Program.

M-1200 WELDING PROCEDURE SPECIFICATIONS

M-1210 RESPONSIBILITY FOR PREPARATION

The responsibility for preparing and approving the Welding Procedure Specification, and the method of handling revisions should be defined in the Quality Assurance or Control Program.

M-1211 Content Requirements

Section IX identifies the essential and also the nonessential variables required to be included in a Welding Procedure Specification. Supplementary information that should be provided, as applicable, is listed in (a) through (k) below. This information may be provided in other specifications, drawings, or documents that are referenced in or used in conjunction with the Welding Procedure Specification, as described in [M-1213](#).

- (a) weld joint design, including fit-up
- (b) method of weld joint preparation
- (c) cleaning requirements prior to welding

(d) purge requirements in addition to QW-408.5 of Section IX, such as oxygen content, volume changes, or time of purging prior to welding, and minimum thickness of deposited metal prior to removal or purge

(e) cleaning and examination requirements between weld passes

(f) instructions for measuring preheat and interpass temperatures

(g) preheat temperature control

(h) interpass temperature control

(i) temperature maintenance and control after welding

(j) welding current and voltage control including method and frequency of measurement

(k) welding technique including, as applicable, method of starts and stops, method of arc initiation, and weave or stringer bead

M-1212 Format of Welding Procedure Specification

Welding Procedure Specifications may consist of a single document. Alternatively, there may be a general Welding Procedure Specification that applies to and is referenced by a number of detailed Welding Procedure Specifications. The general Welding Procedure Specification may also reference other specifications to define other requirements such as welding filler materials, purging, and postweld heat treatment.

M-1213 Application of Welding Procedures

Selection of qualified welding procedures should be based on the essential variables applicable for the production weld. Any additional requirements of the Design Specification, such as component classification, applicable Code edition and addenda, fracture toughness requirements, and manufacturing considerations, should be evaluated in the selection of welding procedures for specific weld joints.

M-1300 WELDING PERFORMANCE QUALIFICATION AND ASSIGNMENT

It is necessary that a system be established to ensure that properly qualified welders and welding operators are used for specific weld joints. The system should include confirmation that the welder and welding operator qualifications are up to date and valid for the application.

M-1400 CONTROL OF WELDING

A system should be established and maintained that is capable of

(a) maintaining the identification of acceptable electrodes, filler metal, consumable inserts, and fluxes until consumed, and

(b) minimizing moisture absorption of coated electrodes and of fluxes

The system should include requirements for receiving inspection, storage, handling, use of holding ovens, control of exposure time at ambient temperature, and reconditioning.

M-1500 NONDESTRUCTIVE EXAMINATION OF WELDS

M-1510 RESPONSIBILITY FOR PREPARATION

Nondestructive examination procedures should be prepared in accordance with the requirements of the particular Article of this Section used in fabrication, including acceptance standards. The responsibility for preparation and approval of procedures and the method of handling revisions should be defined in the Quality Assurance or Control Program.

M-1520 VERIFICATION OF APPLICABILITY

Prior to assignment of nondestructive examination procedures to specific weld joints, a review should be made of the Code classification of the component and the Code

edition and addenda to ensure that the type of examination, procedure and acceptance standards, frequency, and time of examination are all correct for the application.

M-1600 POSTWELD HEAT TREATMENT

A system should be established and maintained that is capable of meeting PWHT requirements for heating and cooling rates, metal temperature, metal temperature uniformity, and temperature control. The location of thermocouples, furnace loading to prevent direct impingement of flame on parts or components, and the furnace atmosphere should be considered in establishing procedures.

M-1700 EXAMINATION AND DIMENSIONAL INSPECTION

Examinations and dimensional inspections should be made to provide assurance that the requirements of this Section are met and that welding conforms with the procedures, specifications, and drawings. As applicable, consideration should be given to (a) and (b).

(a) Before welding

- (1) material identification
- (2) weld preparation surfaces
- (3) cleanliness
- (4) root opening
- (5) offset
- (6) alignment
- (7) socket engagement on socket welds

(b) After welding

- (1) cracks or linear indications
- (2) rounded indications
- (3) overlap
- (4) undercut
- (5) lack of required penetration
- (6) surface finish
- (7) weld profile

NONMANDATORY APPENDIX N

ARTICLE N-1000 DYNAMIC ANALYSIS METHODS

N-1100 INTRODUCTION AND SCOPE

Section III does not require dynamic analysis. However, the design of nuclear components requires consideration of the seismic and other dynamic inputs which are defined in the Design Specifications for the component. Component design may be based on the use of static forces resulting from equivalent earthquake acceleration acting at the centers of gravity of the extended masses, or a dynamic system analysis may be used to show how seismic loading is transmitted from the defined ground motions to all parts of the buildings, structures, equipment, and components. N-1100 through N-1200 of this Appendix are presented to illustrate one or more acceptable steps for seismic dynamic analysis. It is not intended that these steps are the only acceptable ones, since the seismic dynamic analysis involves a series of steps, and some of these steps have acceptable alternative methods. Dynamic analysis in general uses techniques which are illustrated in seismic analysis. Those technical areas of dynamic analysis used in nuclear component design which are not specifically illustrated by seismic analysis are included in N-1300 through N-1700.

N-1110 DEFINITIONS AND NOTATIONS

(a) The *magnitude* of an earthquake is a measure of the size of an earthquake and is related to the energy released in the form of seismic waves. Magnitude means the numerical value on a Richter scale. The *intensity* of an earthquake is a measure of its effects on man, on man-built structures, and on the earth's surface at a particular location. Intensity is measured by the numerical value of the modified Mercalli scale.

(b) The *Safe Shutdown Earthquake* (SSE) is that earthquake which is based upon an evaluation of the maximum earthquake potential considering the regional and local geology and seismology and specific characteristics of local subsurface material. It is that earthquake which produces the maximum vibratory ground motion for which those structures, systems, and components important to safety are designed to remain functional.

(c) The *Operating Basis Earthquake* (OBE) is that earthquake which, considering the regional and local geology and seismology and specific characteristics of local subsurface material, could reasonably be expected to affect the plant site during the operating life of the plant.

(d) The *response spectrum* is defined as a plot of the maximum response (acceleration, velocity, or displacement) of a family of idealized linear single-degree-of-freedom damped oscillators as a function of natural frequencies (or periods) of the oscillators to a specified vibratory motion input at their supports.

(e) The *design ground response spectrum* is a smooth response spectrum obtained by analyzing, evaluating, and statistically combining a number of individual response spectra derived from the records of significant past earthquakes.

(f) The *maximum (peak) ground acceleration* (for a given site) is defined as that value of the acceleration which corresponds to zero period in the design response spectra for that site. At zero period the design response spectra acceleration is identical for all damping values and is equal to the maximum (peak) ground acceleration specified for that site.

(g) *Normal mode — time history methods* use the normal mode theory and a time history of the input motion. When normal mode theory is used, the maximum response is determined by obtaining the combined response of all individual modes at a particular time.

(h) *Direct integration — time history methods* use numerical step-by-step integration of the equations of motion of a time history of the input motion.

(i) *Equivalent statical methods* means the use of a static loading coefficient which gives a definite upper limit to the response.

(j) *Coupled structures and plant equipment* include those structures and plant equipment which, because of their mass and stiffness properties, significantly influence the dynamic response of each other and must be considered together in a dynamic analysis.

(k) *Uncoupled* structures and plant equipment include those structures and plant equipment which, because of their mass and stiffness properties, do not significantly influence the dynamic response of each other and can be considered in separate dynamic analysis.

(l) *Seismic systems* are all those structures for which loads induced by earthquake should be considered.

(m) *Rigid range* is used to describe those frequencies of structures, systems, or components whose natural frequencies are greater than some value at which dynamic response acceleration is essentially the same as the impact acceleration. For example, a response to the seismic design response spectra of [Figures N-1211\(a\)-1](#) and [N-1211\(b\)-1](#) has unity amplification of acceleration above 33 Hz.

(n) *Spectrum consistent time history* is a time history which is artificially generated to essentially envelop a given design response spectrum.

N-1200 SEISMIC ANALYSIS

N-1210 EARTHQUAKE DESCRIPTION

N-1211 Ground Response Spectrum

(a) The *horizontal* component ground design response spectra of the Safe Shutdown Earthquake (SSE) or the Operating Basis Earthquake (OBE) on sites underlain by rock or by soil may be linearly scaled from [Figure N-1211\(a\)-1](#) in proportion to the maximum horizontal ground acceleration specified for the earthquake chosen. [[Figure N-1211\(a\)-1](#) corresponds to a maximum horizontal ground acceleration of $1.0g$ and accompanying displacement of 36 in. (915 mm).] The applicable multiplication factors and control points are given in [Table N-1211\(a\)-1](#). For damping ratios not included in [Figure N-1211\(a\)-1](#) or [Table N-1211\(a\)-1](#), a linear interpolation may be used.

(b) The *vertical* component ground design response spectra of the SSE or the OBE on sites underlain by rock or by soil may be linearly scaled from [Figure N-1211\(b\)-1](#) in proportion to the maximum horizontal ground acceleration specified for the earthquake chosen. [[Figure N-1211\(b\)-1](#) is based on a maximum horizontal ground acceleration of $1.0g$ and accompanying displacement of 36 in. (915 mm).] The applicable multiplication factors and control points are given in [Table N-1211\(b\)-1](#). For damping ratios not included in [Figure N-1211\(b\)-1](#) or [Table N-1211\(b\)-1](#), a linear interpolation may be used.

N-1212 Time History

N-1212.1 Frequency Content of Time History. The design ground response spectra used in the earthquake resistance design, such as those illustrated in [Figures N-1211\(a\)-1](#) and [N-1211\(b\)-1](#) for the design of nuclear power plant facilities, are generally based on multi-component time history motions from a number of major earthquakes. There are no single recorded earthquakes

ground motions which have such uniform frequency distribution. However, it is often necessary to generate a spectrum-consistent time history motion whose response spectrum matches the design response spectrum for a given damping value. The purpose of developing such a spectrum-consistent time history motion is to provide the analyst with an acceptable basis for generating floor (in structure) response spectra and performing time history analysis of systems and components.

Several acceptable approaches are presented below.

(a) *Modified Earthquake Records.* An acceptable approach is to modify the components of past earthquake records using spectral raising and suppressing techniques (refs. [71] and [72]).

Spectral raising is accomplished by adding to the original time history a harmonic function at the frequency of interest with a phase angle such that the response spectral value at this frequency will be increased to a desired amount; the time when the maximum vibratory motion occurred will be the same. In this way, the spectral characteristics of the modified time history will be similar to the original earthquake records. Consequently, statistical characteristics of past time history motions can be maintained as discussed in [N-1213](#).

When the time history response spectrum is higher than the design response spectrum at a frequency, spectral suppressing can be carried out by passing the time history through a linearly damped oscillator connected in series with a second damper. This damping arrangement will reduce the response spectral value, locally at the natural frequency of the oscillator, to the desired amount.

This usually requires an iterative procedure. A repetitious application of the raising and suppressing techniques may be used to arrive at a time history motion whose response spectrum is sufficiently close to the design spectrum.

(b) *Synthesized Time History Motions.* Several methods may be used to generate time history motions without the direct use of an actual earthquake record. One method uses power spectral density functions to generate the time history motions. Power spectral density (PSD) functions may be calculated for the strong motion portion of actual earthquake records by assuming that the strong motion portion is stationary and Gaussian. This (PSD) function may then be used to produce a sample of a pseudo-earthquake ensemble by filtering a white noise record with unit power density through a linear damped system (ref. [73]).

A second procedure which may be used is to pass Gaussian shot noise through selected filters (ref. [74]). This method may produce acceptable results if used in conjunction with the spectral raising and suppressing techniques described earlier.

Another acceptable procedure involves the superposition of continuous waves with the assumption that earthquake motion is stationary for the strong motion portion of the total time duration. The earthquake is characterized

Figure N-1211(a)-1
Horizontal Design Response Spectra Scaled to 1g Horizontal Ground Acceleration

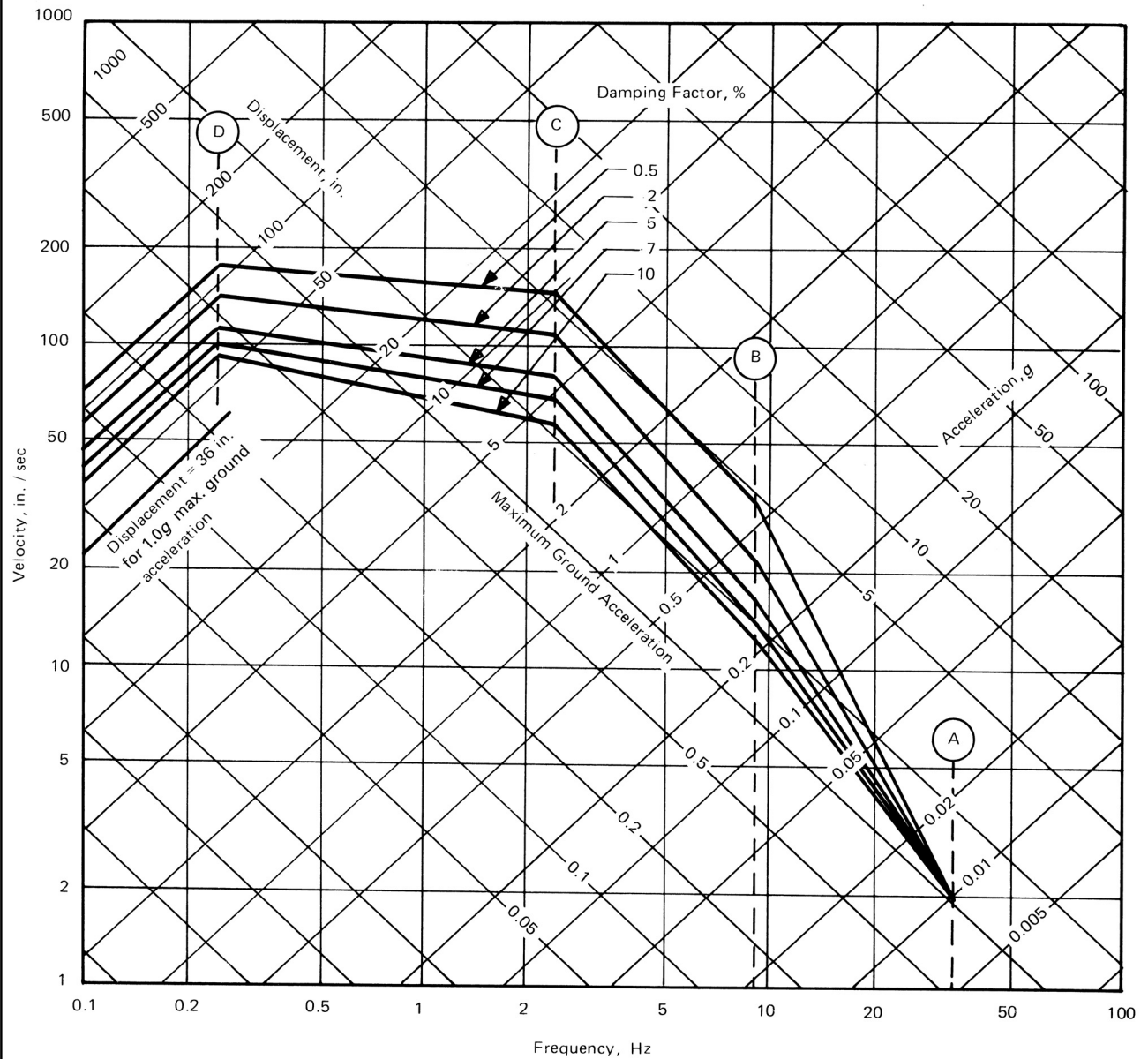


Table N-1211(a)-1
Horizontal Design Response Spectra Relative Values of Spectrum Amplification Factors for Control Points

Percent of Critical Damping	Amplification Factors for Control Points			
	Acceleration [Note (1)], [Note (2)]			Displacement [Note (1)], [Note (2)]
	A (33 Hz)	B (9 Hz)	C (2.5 Hz)	D (0.25 Hz)
0.5	1.0	4.96	5.95	3.20
2.0	1.0	3.54	4.25	2.50
5.0	1.0	2.61	3.13	2.05
7.0	1.0	2.27	2.72	1.88
10.0	1.0	1.90	2.28	1.70

NOTES:

- (1) Maximum ground displacement is taken proportional to maximum ground acceleration and is 36 in. (915 mm) for ground acceleration of 1.0g.
- (2) Acceleration and displacement amplification factors are taken from recommendations given in ref. [1] and discussed in refs. [2] and [3].

as the product of a normalized stationary process and a scaling factor that establishes the magnitude of the motion and its envelope shape which consists of three distinct parts: the buildup, the flat or stationary portion, and the motion decay. The random process is represented by a finite number of superimposed sine waves with distinct periods and randomly selected phase angles (ref. [30]).

Synthesized time history motions for the three perpendicular directions of an earthquake may not have time phase relationships similar to those of past earthquake records. The procedures of N-1213.1 may be used to check that time phasing is acceptable.

N-1212.2 Duration of Time History.

(a) It has been observed (ref. [51]) that the earthquake duration effect on the response spectrum shape is small for periods shorter than 0.5 sec, which is the period range significant for nuclear power plant structures, systems and components.

(b) Although actual long duration earthquakes tend to excite a much wider range of frequencies, this effect is included in the nuclear power plant design by using the smoothed, envelope type of design response spectra. The use of a design response spectrum in qualifying structures and components dictates that only maximum response amplitudes, be it displacement or stresses, are computed. The duration of ground motions, which is an influence in prolonging large amplitude structural response, may be important when fatigue life of the structures and components is considered. The fatigue effect is discussed in N-1214.

(c) The earthquakes used in producing the recommended response spectra [Figures N-1211(a)-1 and N-1211(b)-1] occurred almost exclusively in the California area originating along the Circumpacific Belt. They are referred to as type 2 earthquakes in ref. [51]. These type 2 earthquakes are associated with moderate distances from

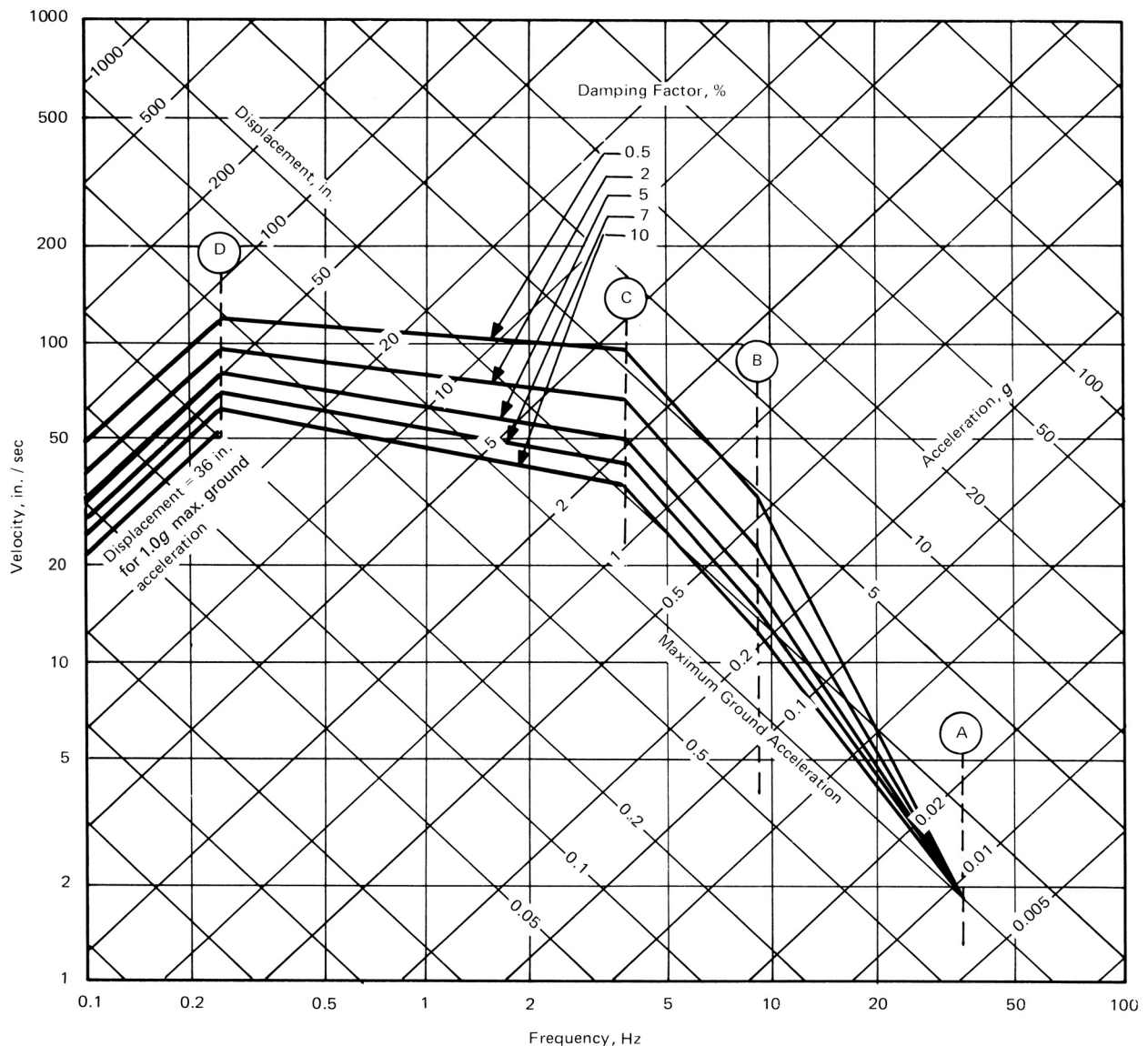
the focus and occur only on firm ground. They last for only a moderate time. The type 2 earthquakes are unlike those referred to as type 3 which occur on soft soil such as the Mexico City earthquakes of July 6, 1964, and which have a much longer duration.

(d) Maximum accelerations of type 2 earthquakes generally occur in the first 10 sec of strong shaking. For instance, the north-south component acceleration recorded at the El Centro earthquake, May 18, 1940, shows that the maximum horizontal ground acceleration of 0.33 g occurs at about 2 sec after the instrument started recording.

(e) Time history analysis is necessary to qualify systems and components which exhibit highly nonlinear characteristics, such as the opening and closing of large gaps between members. Long seismic inputs can waste computation resources. Duration of the artificial time histories should be sufficient to allow the structure enough time to achieve its maximum response. For developing response spectra, the input time history used should be chosen long enough so that the resultant response spectra do not significantly change if the time history is increased. For developing spectrum-consistent time histories, the resultant time history should be long enough so that further increases in its length will not produce significantly different response spectra.

(f) Generally the minimum duration of the strong seismic motion required may be taken as 6 sec for Code components. If this minimum 6 sec strong motion duration is used, then a buildup duration of about 4 sec is recommended to precede the strong motion. The buildup region may be taken as the time duration from where motion is zero to the time where stationarity or strong motion begins. In cases where it is uncertain as to the minimum strong motion duration required to produce maximum structural response, the analysis may be extended to longer strong motion durations such as 10 sec in order to

Figure N-1211(b)-1
Vertical Design Response Spectra Scaled to 1g Horizontal Ground Acceleration



evaluate whether significant response increases occur above those of the 6 sec duration. The method in ref. [71] may be used when required to include shorter or longer periods of vibratory energy in an artificial time history.

N-1213 Directional and Time Phase Considerations

Three orthogonal components of earthquake excitations are normally considered for nuclear power plant design. Rotational ground motion in the three directions, however, may be neglected. The triaxial input excitations are characterized by the relative magnitudes of the peak

accelerations of the three excitations, and the relative values of the response spectra over the frequency range of interest (N-1211). Time histories which are consistent with the design response spectra may be developed through methods discussed in N-1212 and which essentially envelop the design response spectra. Since these artificial time histories may not have time phase relationships similar to those of past earthquake records, the procedures of N-1213.1 may be used to check that time phasing is acceptable.

N-1213.1 Time Phase Relationships. The peak acceleration of the three orthogonal synthetic time histories generally need not occur at the same time. In order to

Table N-1211(b)-1
Vertical Design Response Spectra Relative Values of Spectrum Amplification Factors for Control Points

Percent of Critical Damping	Amplification Factors for Control Points			
	Acceleration [Note (1)], [Note (2)]			Displacement [Note (1)], [Note (2)]
	A (33 Hz)	B (9 Hz)	C (3.5 Hz)	D (0.25 Hz)
0.5	1.0	4.96	5.67	2.13
2.0	1.0	3.54	4.05	1.67
5.0	1.0	2.61	2.98	1.37
7.0	1.0	2.27	2.59	1.25
10.0	1.0	1.90	2.17	1.13

NOTES:

- (1) Maximum ground displacement is taken proportional to maximum ground acceleration and is 36 in. (915 mm) for ground acceleration of 1.0g.
- (2) Acceleration amplification factors for the vertical design response spectra are equal to those for horizontal design response spectra at a given frequency, whereas displacement amplification factors are $\frac{2}{3}$ those for horizontal design response spectra. These ratios between the amplification factors for the two design response spectra are in agreement with those recommended in ref. [1] and discussed in refs. [2] and [3].

stimulate natural earthquake occurrences, the correlation of the synthesized time histories may be evaluated by calculating the cross correlation coefficients and the coherence functions (refs. [4], [5], and [6]). The artificially generated time histories are acceptable if both their cross correlation coefficients and their coherence functions are approximately equal to the respective functions for past earthquake records. An absolute value of the correlation coefficient less than 0.16 is acceptable. For the coherence function the numerical values ranging between 0.0 and 0.3 with an average of approximately 0.2 are acceptable.

N-1214 Cyclic Criteria

(a) An acceptable cyclical load basis for fatigue analysis of earthquake loading of equipment and components is 10 equivalent maximum stress cycles per earthquake.

(b) The equivalent maximum stress cycle is defined as the full range including plus or minus seismic load calculated by equivalent static or response spectrum modal analysis techniques.

(c) The total usage factor contribution is based upon the number of earthquakes considered for the component.

N-1220 METHODS OF DYNAMIC ANALYSIS

N-1221 Modeling Techniques for Dynamic Analysis

In the course of preparation.

N-1222 Time History Method

(a) The structural system to be analyzed can be generally classified into two forms, linear and nonlinear. In the linear system the structure is idealized in such a way that the responses and motion retain a linear relation with the applied loads. A nonlinear structural dynamic system

may be caused by either a material nonlinearity (i.e., a nonlinear stress-strain relationship of the material), a geometrical nonlinearity (an excessive deformation, a support gap, etc. which significantly changes the geometry of the system), or a mixture of the two.

(b) Dynamic analysis for either a linear or nonlinear system is based on the solution of a set of simultaneous, differential equations of motion with given initial and boundary conditions. Acceptable methods are presented for solution to linear and nonlinear equations in N-1222.

N-1222.1 Linear.

(a) The response of a multi-degree-of-freedom linear structural system is described by the differential equation of motion expressed in matrix form

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{f\} \quad (1)$$

where

C = matrix of viscous damping coefficients

f = force vector and function of time representing external loading at mass points

K = stiffness matrix for the mass points of the linear elastic structure

M = mass matrix

x = displacement vector

\dot{x} = velocity vector

\ddot{x} = acceleration vector

In all typical linear matrix formulations, matrices M , C , and K are symmetric.

(b) For a seismic analysis where ground motion is the source of excitation, f in eq. (a)(1) can be replaced by $-[M]\{\ddot{Z}\}$, where \ddot{Z} is the acceleration of ground motion in the same direction of x which is the relative displacement

Figure N-1211(a)-1M
Horizontal Design Response Spectra Scaled to 1g Horizontal Ground Acceleration

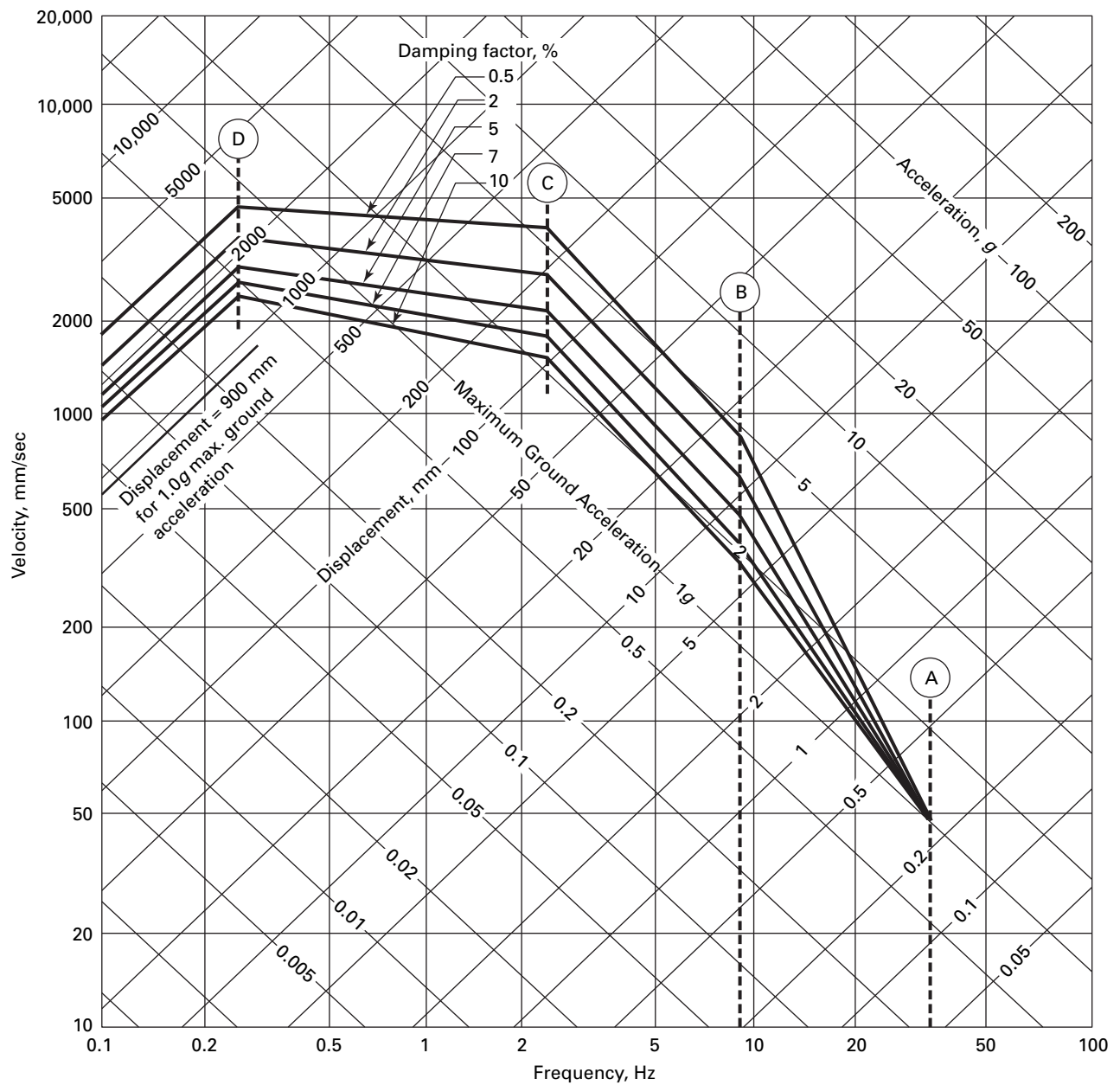
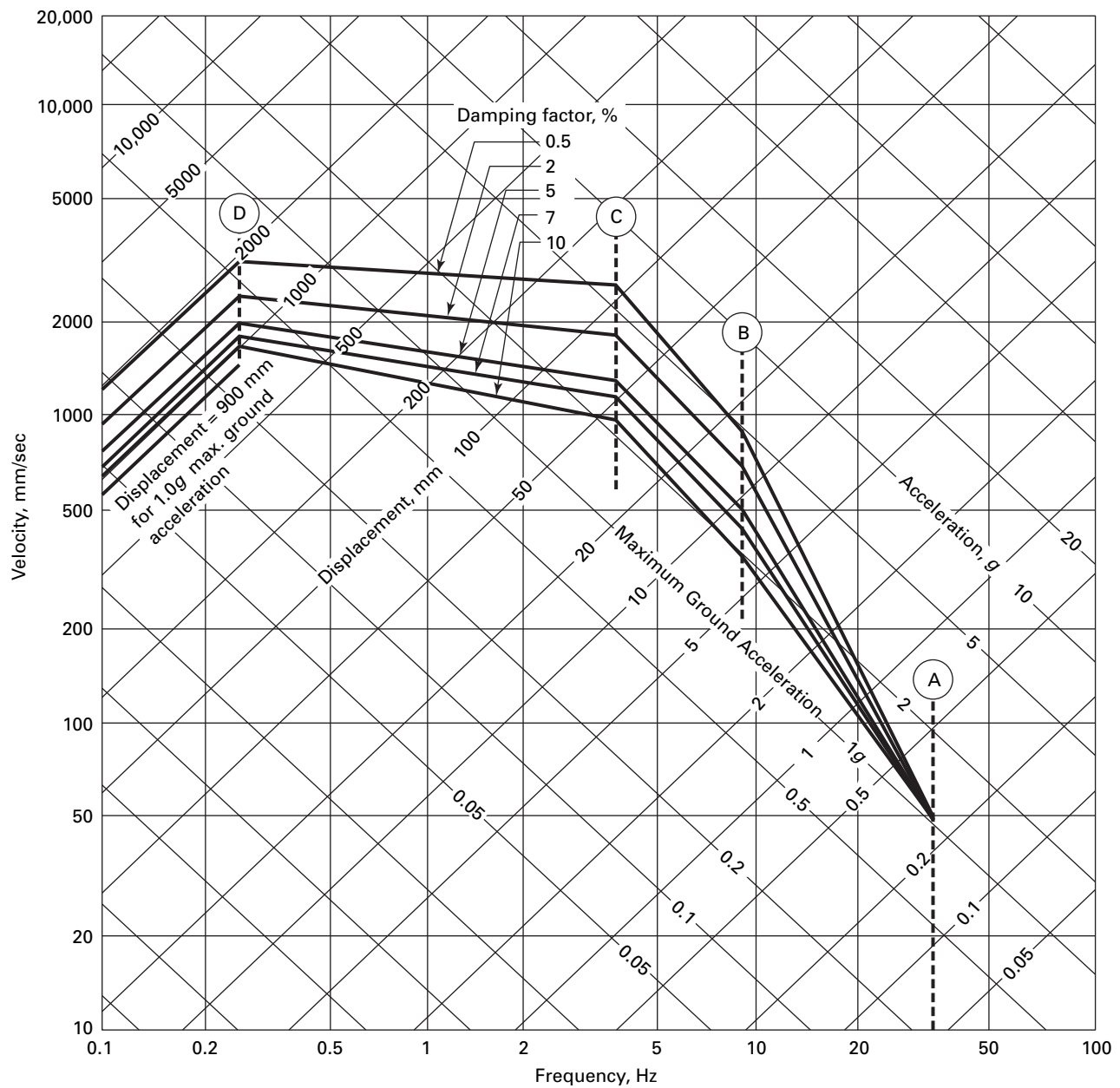


Figure N-1211(b)-1M
Vertical Design Response Spectra Scaled to 1g Horizontal Ground Acceleration



with respect to the ground motion. For systems excited at multiple support locations, further modification of the force vector is necessary (see N-1228).

(c) The differential equation of motion may be solved either analytically or numerically. A rigorous analytical solution of the simultaneous linear differential equations is often impractical and unnecessary. Simplification of the damping matrix may be made to facilitate computations.

(d) For example, in the special case of proportional damping (i.e., when the damping matrix is reduced to a linear combination of the mass and stiffness matrices), the classical method of modal analysis may be applied since eq. (a)(1) can be decoupled.

(e) See N-1230 for acceptable treatments of damping. A numerical solution using step-by-step integration method may be applied to a set of coupled as well as uncoupled differential equations. The direct integration method does not require the process of uncoupling modes; therefore, no calculation of natural frequencies and mode shapes is necessary. The direct integration method permits the handling of proportional damping where this treatment is required.

N-1222.1.1 Method of Modal Superposition (Ref. [8]).

N-1222.1.1.1 Modal Analysis. The method of modal analysis usually assumes that a modal matrix be defined such that

$$\begin{aligned} [\phi]^T [M] [\phi] &= [I] \\ [\phi]^T [K] [\phi] &= [\omega^2] \\ [\phi]^T [C] [\phi] &= [2\xi\omega] \end{aligned} \quad (2)$$

where the right-hand sides are diagonal matrices, ϕ is the modal matrix, ω is the natural frequency of the structure, and ξ is the modal fraction of critical damping. This formation uses damping matrices that are proportional.

Let $\{\eta\}$ be the normal coordinates such that the displacement $\{x\}$ can be defined by the transformation

$$\{x\} = [\phi]\{\eta\} \quad (3)$$

Equation N-1222.1(a)(1) can be written by means of this transformation as

$$\{\eta''\} + 2\xi\omega\{\eta'\} + \omega^2\{\eta\} = [\phi]^T \{f\} \quad (4)$$

which represents a decoupled system of individual modal equations. Response in each mode may be obtained by solving the differential equations individually for the corresponding mode. The total response of the structure is therefore the result of combining the responses of all its component modes.

The equations of motion for a linear system may be uncoupled by means of a transformation to a system of normal coordinates provided that the damping matrix C is a linear combination of M and K matrices.

N-1222.1.1.2 Complex Frequency Response Method. As an alternative approach to the method of modal analysis described above, the complex frequency response method may be used to determine dynamic response of the system represented by eq. N-1222.1(a)(1).

The complex frequency response method requires that the complex frequency responses of the system be determined first and the seismic excitations be transformed into its frequency domain. The time histories of the response may then be found by inverting the response transforms.

Corresponding to every normal coordinate η_r of eq. N-1222.1.1.1(4), a pair of Fourier transforms may be found:

$$\begin{aligned} G_r(\Omega) &= \int_{-\infty}^{\infty} \eta(t) e^{-i\Omega t} dt \\ Q_r(\Omega) &= \int_{-\infty}^{\infty} f_r(t) e^{-i\Omega t} dt \end{aligned} \quad (5)$$

where

$$f_r(t) = \{\phi_r\}^T \{f(t)\}$$

The Fourier transforms of the excitation and response are related according to

$$G_r(\Omega) = H_r(\Omega) Q_r(\Omega) \quad (6)$$

and

$$H_r(\Omega) = \frac{1}{1 - (\Omega/\Omega_r)^2 + i2\xi(\Omega/\Omega_r)} \quad (7)$$

The r th mode response is given by evaluating the Fourier integral:

$$\eta_r(t) = \frac{1}{2\pi} \int_{-\infty}^{\infty} H_r(\Omega) Q_r(\Omega) e^{i\Omega t} d\Omega \quad (8)$$

and the time history response of the system in the physical coordinate is given by

$$\{x(t)\} = \sum_{r=1}^n \{\phi_r\} \eta_r(t) \quad (9)$$

Standard fast Fourier transform techniques are available for expedient evaluation of $Q_r(\omega)$ and $\eta_r(t)$ of eqs. (6) and (8), respectively (ref. [9]).

N-1222.1.2 Method of Direct Integration.

(a) This is a numerical method directly applied to the solution of the differential equations of motion of linear structural system, eq. N-1222.1(a)(1), in a step-by-step manner. No uncoupling procedure is necessary to compute the responses. The damping matrix need not be proportional.

(b) There are many acceptable schemes available for numerical integration of the equations of motion such as Newmark β -method (ref. [10]), Houbolt method (ref. [11]), and Wilson θ -method (ref. [12]).

(c) Using the matrix formulation by Chan, Cox, and Benfield (ref. [13]) and Newmark β -method, eq. N-1222.1(a)(1) may be transformed into a finite difference equation in recurrence form involving displacement:

$$[D]\{x_{n+1}\} = [B]\{x_n\} - [F]\{x_{n-1}\} + \beta h^2 \left\{ f_{n+1} + \left(\frac{1}{\beta} - 2 \right) f_n + f_{n-1} \right\} \quad (10)$$

where

$$[B] = 2[M] - (1 - 2\beta)h^2[K]$$

$$[D] = [M] + (h/2)[C] + \beta h^2[K]$$

$$[F] = [M] - (h/2)[C] + \beta h^2[K]$$

h = length of the time interval used in the direct integration procedure

n = the number of time intervals lapsed

β = parameter of Newmark β -method in reflecting the type of acceleration function assumed between two time stations and numerical stability of the procedure

(d) The dynamic responses of a structure at any instant of time may be calculated from the time history of excitations and the previous structural responses.

(e) The direction integration method has the advantage of simplicity in computation and elimination of the eigenvalue problem. This method may also be extended to deal with the nonlinear problems.

N-1222.2 Nonlinear.

(a) The linear time history analysis discussed in N-1222.1 is based on the assumptions that external forces are independent of the displacement and velocity, the stress-strain relationship is linear, and the strain-displacement relationship is linear. Many problems of practical consequences exist for which these assumptions are not valid. Some acceptable numerical methods to analyze such nonlinear problems are discussed below.

(b) The following nonlinearities may be introduced into the system:

(1) material nonlinearities (plasticity);

(2) geometric nonlinearities (large displacement);

(3) combination of material and geometric nonlinearities (impact and friction).

(c) The differential equation of motion for nonlinear problems is

$$[M_t]\{U''\} + [C_t]\{U'\} + [K_t]\{U\} = \{P_t\} \quad (11)$$

where

C_t = damping

K_t = stiffness matrices

M_t = time-dependent mass

P_t = force arrays

U = displacement

U' = velocity

U'' = acceleration

The left-hand side of eq. (11) may be made linear by introducing

$$\begin{aligned} [M_t] &= [M] + [M_{nl}] \\ [C_t] &= [C] + [C_{nl}] \\ [K_t] &= [K] + [K_{nl}] \end{aligned} \quad (12)$$

where

C = proportional damping

C_{nl} = damping

K = stiffness matrices

K_{nl} = stiffness matrices

M = time-independent mass

M_{nl} = time-dependent mass

As an example, the compressible fluid flow in a pipe or a moving mass acting on a structure contributes to $[M_{nl}]$ and the closing and opening of gaps contribute to $[C_{nl}]$ and $[K_{nl}]$. Substituting eq. (12) into eq. N-1222.1(a)(1) yields

$$[M]\{U''\} + [C]\{U'\} + [K]\{U\} = \{F\} \quad (13)$$

where

$$\{F\} = \{P_t\} - [M_{nl}]\{U''\} - [C_{nl}]\{U'\} - [K_{nl}]\{U\}$$

Time history methods to solve eq. (11) or (13) fall into two major categories: (1) mode superposition, and (2) direct integration.

N-1222.2.1 Mode Superposition.

(a) The use of normal modes for linear time history method is discussed in N-1222.1. Mode superposition techniques may also be used for the nonlinear problems (refs. [14], [15], and [16]). A problem of a pendulum having a large amplitude is discussed in ref. [14]. Reference [15] presents an analysis of a structure including the following nonlinear characteristics: variable-stiffness elements, contact (compression only) elements, time-varying boundary conditions, and nonmodal and nonlinear damping elements. A ring problem involving large geometric nonlinearities and plastic strains is discussed in ref. [16].

(b) Let $\{\phi_i\}$ and $\{\omega_i\}$ be the i th orthonormal eigenvector and natural frequency, respectively:

$$[M]\{U''\} + [K]\{U\} = 0 \quad (14)$$

A transformation

$$\{U\} = [\phi]\{q\} \quad (15)$$

is introduced in eq. N-1222.2(c)(13). The resulting uncoupled equation is as follows:

$$[I]\{q''\} + [2\xi_j\omega_j]\{q'\} + [\omega_j^2]\{q\} = \{P\} \quad (16)$$

where

- I = identity matrix
- $= [\phi]^T[M][\phi]$
- P = generalized force
- $= [\phi]^T\{F\}$
- q = generalized displacement vector
- ξ_j = j th modal damping ratio
- ω_j = j th natural frequency
- ϕ = set of significant eigenvectors

(c) Eq. (b)(16) may be solved by either explicit numerical integration schemes such as the fourth order Runge-Kutta method (ref. [17]), Hamming's predictor-corrector method (ref. [17]), or by the analytical integration scheme (ref. [14]). These numerical methods are conditionally stable and available as library programs equipped with an automatic time step adjustment feature. These methods provide a stable solution if the integration time step is appreciably smaller than the smallest period associated with eq. (b)(16). In the analytical integration method, the force vector F is approximated over the time step Δt . This allows for Duhamel integral type solutions for eq. (b)(16).

(d) The generalized forces in eq. (b)(16) are updated at each time increment. If the initial stiffness matrix has a relatively large bandwidth and a large number of eigenvectors are not required for the analysis, it is likely that the mode superposition approach will be quite economical as compared with the direct integration procedures discussed in N-1222.2.2. In cases where the applied load distribution is extremely complex or its time variation contains significant high frequency components, or both, it is necessary to include many modes to obtain adequate accuracy by mode superposition. In these cases the direct integration procedure may be more efficient. This procedure is discussed in N-1222.2.2.

N-1222.2.2 Direct Integration. Direct numerical integration methods can be applied to eq. N-1222.2(c)(11) or eq. N-1222.2(c)(13). When applied to eq. N-1222.2(c)(11), the calculation of time-dependent matrices is necessary at each integration time step. When applied to eq. N-1222.2(c)(13), the time-independent matrices may be calculated once at the beginning of the numerical solution and the time-dependent force array is calculated at each time step. These methods are classified into two groups: explicit schemes and implicit schemes. The explicit schemes are hampered by both numerical

instability and convergence problems while the implicit schemes suffer only from convergence problems. These schemes are discussed in N-1222.2.2.1 and N-1222.2.2.2.

N-1222.2.2.1 Explicit Schemes. These schemes convert the differential equations of motion to a set of linear algebraic equations with unknown state variables (at the present time) which are independent of one another. Acceptable explicit schemes include: Runge-Kutta method (ref. [17]), predictor-corrector method (ref. [17]), Nordsieck integration method (ref. [18]), and central difference method (ref. [19]).

As the above methods are conditionally stable, they have a disadvantage of requiring small time step sizes. The first three methods can be made to operate with the variable time step, while the fourth one is a constant time-step procedure. Garnet and Armen (ref. [20]) demonstrate a one dimensional wave propagation problem solution with the aid of a predictor-corrector method, i.e., the modified Adams method. In ref. [21] the Nordsieck integration scheme is demonstrated to solve nonlinear vibration problems in reactor components. Wu and Witmer (ref. [22]) analyzed the problem of large transient elastic-plastic deformation of structures using the central difference method.

N-1222.2.2.2 Implicit Schemes. These schemes convert the differential equations of motion to a set of linear simultaneous algebraic equations and require a matrix inversion to step the solution forward. Acceptable implicit schemes include: Newmark's generalized acceleration method, Wilson's θ method, Nastran's integration method, and Houbolt's method.

(a) *Acceptable implicit schemes:*

(1) *Newmark's Generalized Acceleration Method* (Refs. [10] and [23]). The nodal point velocities and displacements are given by the following equations:

$$\{U'_{n+1}\} = \{U'_n\} + (1 - \gamma)(\Delta t)\{U''_R\} + \gamma(\Delta t)\{U''_{n+1}\} \quad (17)$$

and

$$\begin{aligned} \{U_{n+1}\} &= \{U_n\} + (\Delta t)\{U'_n\} + \\ &\quad \left(1/2 - \beta\right)(\Delta t)^2\{U''_n\} + \beta(\Delta t)^2\{U''_{n+1}\} \end{aligned} \quad (18)$$

Equations (17) and (18) are substituted in eq. N-1222.2(c)(13), which is expressed at the present time point $(n + 1)$. This gives the following:

$$\begin{aligned} &[M] + \gamma(\Delta t)[C] + \beta(\Delta t)^2[K] \left\{U''_{n+1}\right\} = \left\{F_{n+1}\right\} \\ &- \left[(1 - \gamma)(\Delta t)[C] + (1/2 - \beta)(\Delta t)^2[K] \right] \left\{U''_R\right\} \\ &- \left[[C] + (\Delta t)[K] \right] \left\{U_n\right\} - [K] \left\{U_n\right\} \end{aligned} \quad (19)$$

This set of algebraic simultaneous equations in the unknown accelerations is solved and used with eqs. (17) and (18) to obtain the velocity and displacement at the present time. The parameter γ in eq. (19) is a damping parameter. Artificial positive damping is introduced if $\gamma > 0.5$ and artificial negative damping if $\gamma < 0.5$. For linear problems, the method is unconditionally stable if $\beta > (2\gamma + 1)^2/16$ (ref. [12]). For $\gamma = 1/2$, $\beta > 1/4$ gives unconditional stability. For systems with nonlinearities or non-proportional damping or both, there is no analytical expression available for unconditional stability. So to provide a margin of stability for these systems $\beta > 1/4$ is considered for $\gamma = 1/2$. The value of γ slightly larger than 0.5 damp out the highest (and least important) modes while preserving the lower ones. The Newmark method for $\gamma = 1/2$ (no numerical damping) and $\beta = 1/6$ (linear acceleration) is conditionally stable. The Wilson modified method (to make it unconditionally stable) is discussed next.

(2) *Wilson's θ Method* (ref. [12]). To obtain the solution at the present time ($n + 1$), this method assumes that the acceleration varies linearly over the time interval $\tau = \theta \Delta t$, where $\theta > 1.0$. With the aid of eq. (1)(19), $\beta = 1/6$ and $\gamma = 1/2$, solution at time point ($n + \theta$) is obtained. Then the acceleration, velocity, and displacement at the time point $n + 1$ are given by the following expressions:

$$\begin{aligned} \{U''_{n+1}\} &= \{1 - 1/\theta\}\{U''_n\} + \{1/\theta\}\{U''_{n+\theta}\} \\ \{U'_{n+1}\} &= \{U'_n\} + \frac{\Delta t}{2}\left\{\left(U''_n + U''_{n+1}\right)\right\} \\ \{U_{n+1}\} &= \{U_n\} + \Delta t\{U'_n\} + \\ &\quad \frac{\Delta t^2}{6}\left\{U''_{n+1} + 2U''_n\right\} \end{aligned} \quad (20)$$

The last two equations are obtained from eq. (1)(18) with the value of β and γ specified above. This method is unconditionally stable for $\theta > 1.37$ (ref. [12]); for $\theta = 1.4$, the numerical damping is less than 1% provided 22 steps are taken within the natural period of the mode of interest (ref. [25]).

(3) *Nastran's Integration Method* (ref. [26]). In this method the acceleration and velocity are expressed in central finite difference form. They are expressed as follows:

$$\begin{aligned} \{U''_n\} &= \frac{\{U_{n+1}\} - \{2U_n\} + \{U_{n-1}\}}{(\Delta t)^2} \\ \{U'_n\} &= \frac{\{U_{n+1}\} - \{U_{n-1}\}}{2(\Delta t)} \end{aligned} \quad (21)$$

The displacement and the forcing function are expressed as the weighted average of the corresponding magnitudes at the three successive time points $n - 1$, n , $n + 1$. Substitution of these expressions in eq. N-1222.2(c)(13) expressed at time point n gives:

$$\begin{aligned} \left[\frac{1}{(\Delta t)^2} [M] + \frac{1}{2(\Delta t)} [C] + \frac{1}{3} [K] \right] \{U_{n+1}\} &= \\ \frac{1}{3} \{F_{n+1} + F_n + F_{n-1}\} + \\ \left[\frac{2}{(\Delta t)^2} [M] - \frac{1}{3} [K] \right] \{U_n\} + \\ \left[-\frac{1}{(\Delta t)^2} [M] + \frac{1}{2(\Delta t)} [C] - \frac{1}{3} [K] \right] \{U_{n-1}\} \end{aligned} \quad (22)$$

This set of algebraic simultaneous equations in the unknown displacements is solved and used to find velocity and acceleration.

This scheme is a special form of the Newmark method, eq. (1)(18) for $\beta = 1/3$ and $\gamma = 1/2$ (ref. [13]). The value $\beta = 1/3$ provided some margin of stability and for $\gamma = 1/2$, the artificial damping is zero. Note that the load vector F is averaged over three adjacent time steps in the same manner that K is averaged. This is done in order to provide statically correct solutions for massless degrees of freedom.

(4) *Houbolt's Method* (ref. [11]). In this method a third-order interpolating polynomial which fits the known displacements at the three previous time points and the unknown displacement at the present time point is formulated. The expressions for acceleration and velocity are given as follows:

$$\begin{aligned} \{U''_{n+1}\} &= \frac{1}{(\Delta t)^2} [2\{U_{n+1}\} - 5\{U_n\} + \\ &\quad 4\{U_{n-1}\} - \{U_{n-2}\}] \\ \{U'_{n+1}\} &= \frac{1}{6(\Delta t)} [11\{U_{n+1}\} - 18\{U_n\} + \\ &\quad 9\{U_{n-1}\} - 2\{U_{n-2}\}] \end{aligned} \quad (23)$$

The eqs. (23) are substituted in eq. N-1222.2(c)(13) which is expressed at the present time point $n + 1$. This gives:

$$\begin{aligned} \left[\frac{2}{\Delta t^2} \left(\frac{11}{6\Delta t} \right) [C] + [K] \right] \{U_{n+1}\} &= \{F_{n+1}\} + \\ \left[\frac{5}{\Delta t^2} [M] + \frac{3}{\Delta t} [C] \right] \{U_n\} &- \left[\frac{4}{\Delta t^2} [M] + \frac{3}{2\Delta t} [C] \right] \{U_{n-1}\} \\ + \left[\frac{1}{\Delta t^2} [M] + \frac{1}{3\Delta t} [C] \right] \{U_{n-2}\} \end{aligned}$$

This set of algebraic simultaneous equations in the unknown displacements is solved and used to find velocity and acceleration.

This method is unconditionally stable for linear problems and introduced artificial damping (the amount of such damping increases with the ratio of time step to natural period of the system). The artificial damping is less than 1% provided 50 time steps are taken within the natural period of the mode of interest (ref. [12]). Thus, the Houbolt method effectively removes higher mode response from the system.

(b) *Approximation of Load Vectors.* All four direct integration methods discussed in the previous section require that the forces at the time point $n + 1$ be known in order to calculate the displacements at that time. These loads, because of the presence of the nonlinear terms, are a function of the displacements which are to be calculated. So, it is not possible to evaluate these terms exactly. These forces may be approximated with the aid of the first-order Taylor series expansion about the motion at time point n as follows:

$$\begin{aligned}\{F(t, U)_{n+1}\} &= \{F(t, U)_n\} + \frac{\delta}{\delta t} \{F(t, U)_n\} (\Delta t) \\ &= 2\{F(t, U)_n\} - \{F(t, U)_{n-1}\}\end{aligned}$$

This expression has an inherent error of order $(\Delta t)^2$ which is the same as the Houbolt and Nastran method (ref. [27]). This expression corresponds to a linear extrapolation of the loads at the two previous time increments which is equivalent to a numerical differentiation. This introduces round off error (noise) which will set a low bound on the integration time step.

(c) *Convergence.* For nonlinear systems, the following two steps are recommended to ensure convergence of the solution to the proper value.

(1) Iterative schemes, based on the residual force array derived from the equation of motion at the present time point $n + 1$ may be used to obtain convergence. This residual, obtained by transferring all the terms in eq. N-1222.2(c)(13) to the right-hand side, is a measure of how well the dynamic equilibrium is satisfied at the present time point $n + 1$. The time-dependent matrices and force arrays, in eqs. N-1222.2(c)(11) and N-1222.2(c)(13), respectively, are modified based on the calculated solution at time point $n + 1$ and the iterative scheme is continued until the dynamic equilibrium is satisfied to a prescribed tolerance. References [25], [28], and [29] discuss this approach to improve efficiency in nonlinear solutions.

(2) Successive computation of the time history, employing successively smaller values of the integration time step, may also be used to ascertain convergence.

In the wave propagation problems, the response may contain the spurious oscillations (refs. [20] and [25]), which are due to the finite element discretization, and

are not eliminated by reducing the time step Δt . These oscillations may often be reduced by employing more uniform element size.

(d) *General Remarks.* No best choice of numerical method has been identified for nonlinear problems. Discussions which are helpful to indicate the advantages and limitations of these previously discussed methods are contained in refs. [13], [20], [22], [23], and [27].

Experience obtained from the solution of actual problems is the most reliable indicator of which of the different integration schemes is superior for a particular problem. Experience and checks of methods adopted for a similar type of physical problem where a correct solution is available is recommended to establish validity for a particular nonlinear method.

N-1222.3 Time History Broadening. To account for the effect of possible frequency variation of the structure, the same time history data may be used with at least three different scaled time intervals: Δt (the reference interval) and $(1 \pm \Delta f_j / f_j) \Delta t$, for the analysis of equipment, where f_j is the fundamental structural frequency and Δf_j is a parameter defining the frequency variation due to uncertainties. This variation of the time scale interval has a similar effect to widening the spectral peak when generating the smoothed response spectrum. If one of the equipment frequencies f_e is within the range $f_j \pm \Delta f_j$, it is recommended that the time history also be used with additional scaled time intervals of $[1 \pm (f_e - f_j) / f_j] \Delta t$.

N-1223 Response Spectrum Method

N-1223.1 Modal Combination. In the response spectrum method, the peak values of particular responses of interest (displacement, acceleration, shear, moment, etc.) are determined for each mode. The total response may be obtained by combining the peak modal responses by the square root of sum of squares (SRSS) method. Mathematically this is expressed as follows:

$$R = \pm \left[\sum_{k=1}^n R_k^2 \right]^{1/2} \quad (24)$$

where

R = the total response of interest (strain, displacement, stress, moment, shear, etc.) based on the SRSS method of combining the individual peak modal responses

R_k = the peak response of interest due to the k th mode

n = the number of significant modes

The above method may be used to determine the particular response for all significant modes regardless of frequency spacing.

N-1223.2 Combination of Effects Due to Triaxial Excitation.

(a) In the response spectrum method of analysis, the natural frequencies, mode shapes, and the load for each mode are first determined. The load for each mode for unit generalized response is the product of the stress matrix, the mode shapes, and the mode participation factors. When this product is multiplied by the generalized response determined from the spectrum curves, the load for each natural mode results.

(b) The following assumptions may be used in combining the loads of each natural mode and for each direction.

(1) The peak responses of the different modes due to any one excitation do not occur at the same time.

(2) The peak generalized responses due to the three different earthquake excitations for the same mode do not occur at the same time.

(3) The peak stresses due to different modes and due to different excitations generally do not occur at the same location on the structure nor at the same angular orientation.

(c) These assumptions are consistent with the use of the SRSS method to compute the resultant responses. It is necessary to use the SRSS method only on scalar components because the SRSS of components orthogonal to each other results in magnitudes equivalent to vector sums thus implying simultaneous occurrence.

(d) The following general procedure should be used to combine the seismic responses due to triaxial excitation:

$$R_{ij} = \pm \left[\sum_{k=1}^3 R_{ijk}^2 \right]^{1/2} \quad (25)$$

where

R_{ijk} = maximum, co-directional seismic response of interest (strain, displacement, stress, moment, shear, etc.) associated with coordinates i and j due to earthquake excitation in the k th direction

R_{ij} = seismic response of interest for design (strain, displacement, stress, moment, shear, etc.) obtained by the SRSS rule to account for the nonsimultaneous occurrences of R_{ijk}

N-1224 Component or Equipment Testing

In the course of preparation.

N-1225 Simplified Dynamic Analysis**N-1225.1 Seismic Load Coefficient Method for Piping System Analysis.****N-1225.1.1 Simplified Seismic Load Coefficient Method Using Floor or Amplified Response Spectra as the Seismic Design Basis Input.**

(a) *Piping System Loads.* A simplified seismic load coefficient method for earthquake resistant design for equivalent seismic inertia induced resultant piping stresses,

displacements, loads, and support reaction loads may be used to describe the earthquake input with a format as follows:

$$F_{phi} = K_{hi} S_{ami} W \quad (26)$$

$$F_{pv} = K_v S_{ami} W \quad (27)$$

where

F_{phi} = equivalent static inertia force applied to the piping and piping components in the i th horizontal direction as defined in N-1225.2

F_{pv} = equivalent static inertia force applied to the piping and piping components in the vertical direction as defined in N-1225.2

K_{hi} = the load coefficient applied to the piping and piping components in the i th horizontal direction. The value of K_{hi} shall be determined as follows:

(a) For Soil Sites: $V_s < 3500$ ft/sec (1100 m/s)

where

$$K_{hi} = 1.0; l_{hi} < 3.5 l_v$$

$$K_{hi} = 0.65; 3.5 l_v \leq l_{hi} \leq 5.0 l_v$$

$$K_{hi} = 0.4; l_{hi} > 5.0 l_v$$

(b) For Rock Sites: $V_s > 3500$ ft/sec (1100 m/s)

where

$$K_{hi} = 1.0; l_{hi} < 2.5 l_v$$

$$K_{hi} = 0.6; 2.5 l_v \leq l_{hi} \leq 4.0 l_v$$

$$K_{hi} = 0.4; l_{hi} > 4.0 l_v$$

Piping systems with l_{hi} in more than one of the categories listed above shall use the highest K_{hi} value applied to two lateral support spans on either side of the span requiring the highest K_{hi} value.

K_v = the load coefficient applied to the piping in the vertical direction. The value of K_v shall be determined as follows:

For Soil Sites: $V_s < 3500$ ft/sec (1100 m/s)

where

$$K_v = .75; l_{hi} < 4.0 l_v$$

$$K_v = 1.0; l_{hi} \geq 4.0 l_v$$

For Rock Sites: $V_s > 3500$ ft/sec (1100 m/s)

where

$$K_v = 1.0$$

Piping systems with l_{hi} in more than one of these categories listed above shall use the highest K_v values applied to two lateral support spans on either side of the span requiring the highest K_v value.

S_{ami} = peak acceleration of applicable amplified or floor response spectra in the i th direction (in g's)
 l_{hi} = span in the i th direction between lateral supports measured along the axis of the pipe. Lateral supports added to the piping system next to large inline components whose weights equal or exceed the weight of the pipe between nominal dead weight supports contained in Table NF-3611.1, such as valves, strainers, etc., do not need to be considered in the determination of l_{hi} value
 l_v = deadweight vertical support span from Table NF-3611.1 for the nominal pipe size for the piping system under consideration
 V_s = building foundation media shear wave velocity
 W = the total piping dead load (weight) which exists during the postulated seismic event. This includes piping weight, water weight and insulation. The units of the term W must be consistent with those of the term F_{phi} and F_{pv} above.
 When the piping system under consideration contains more than one pipe size K_{hi} and K_v factors shall be separately determined for and applied to each nominal pipe size in the piping system. In addition, if the K_{hi} and K_v values change as a result of a pipe size change, the highest K_{hi} , K_v values at the point of piping size change shall be applied to the first two lateral support spans at the point piping size change having the lower K_{hi} , K_v values.

(b) Special Considerations for Support Reactions. In using this method to determine piping supports loads for seismic events, the specification of a minimum lateral and vertical support design load is required. This minimum design load is a function of pipe size and shall be determined as follows:

$$R_{Lmin} = \left(\text{MSLF} \right) \left(w' \right) \left[\sum_{i=1}^2 \left(S_{ami} \right)^2 \right]^{\frac{1}{2}} \quad (28)$$

$$R_{Amin} = \left(\text{MSLF} \right) \left(w' \right) \left[\sum_{i=1}^2 \left(S_{ami} \right)^2 \right]^{\frac{1}{2}} \quad (29)$$

where

S_{ami} = values that are the peak accelerations of the applicable amplified or floor response spectra in each of two orthogonal horizontal directions (units are g's)
 $R_{Vmin} = (\text{MSLF})(w')(S_{ami})$

where

S_{ami} = peak acceleration of the applicable amplified or floor response spectra in the vertical direction.

The remaining parameters are defined as follows:

MSLF = the Minimum Support Load Factor as defined in Table N-1225.1.1(b)-1 [MSLF has the units of ft (m)]

R_{Lmin} = Minimum Lateral Seismic Support Design Support Load [R_{Lmin} has the units of lbf (N)]

R_{Amin} = Minimum Axial Seismic Support Design Load for these axial supports not in the vertical direction. For axial supports in the vertical direction the minimum support design load shall be determined as R_{Vmin} . [R_{Amin} has units of lbf (N)].

R_{Vmin} = Minimum Vertical Seismic Support Design Load [R_{Vmin} has the units of lbf (N)]

w' = the total piping dead load per unit length of the piping which exists during a seismic event. This includes piping weight, water weight, insulation weight, etc. [w' has the units of lbf/ft (N/m)].

In instances where Seismic Support Design Loads determined using the methodology of N-1225.1(a) are less than R_{Lmin} , R_{Amin} , or R_{Vmin} , the support design load shall be specified as R_{Lmin} , R_{Amin} , or R_{Vmin} , as determined in this subarticle.

(c) Inline Component Extended Structures. The seismic applied accelerations at the center of gravity (cg) of inline component extended structures may be determined as follows:

$$av_h = \left[\sum_{i=1}^2 \left(0.9 S_{ami} \right)^2 \right]^{\frac{1}{2}} \quad (30)$$

when the S_{ami} values are the peak accelerations of the applicable amplified or floor response spectra in each of two orthogonal horizontal directions.

$$av_v = 1.2 S_{ami} \quad (31)$$

when the S_{ami} value is the peak of the applicable amplified floor response spectra in the vertical direction.

The remaining parameters are defined as follows:

av_h = the total acceleration applied to the inline component extended structure center of gravity (cg) in the horizontal direction

av_v = the acceleration applied to the inline component extended structure center of gravity (cg) in the vertical direction

Table N-1225.1.1(b)-1
Minimum Support Load Factor

Nominal Pipe Size [ϕ], NPS (DN)	MSLF, ft (m)
$\phi \leq 2$ (50)	8.5 (2.6)
2 (50) < $\phi \leq 4$ (100)	8.0 (2.4)
4 (100) < $\phi \leq 6$ (150)	7.5 (2.3)

The application of the eqs. (30) and (31) to determine Inline Component Extended Structures applied accelerations shall be limited to $W_v h^2/w'$ values less than or equal to $185,000 \text{ ft-in.}^2$ ($36.4 \times 10^6 \text{ m-mm}^2$), when:

- h = height from the pipe centerline to inline component extended structure center of gravity (cg) [in. (mm)]
- W_v = weight of the inline component extended structure considered at its center of gravity (cg) location [lbf (N)]
- w' = is as previously defined

For inline components when the extended structure vertical (axial) axis does not correspond to global vertical axis, the method is still applicable but care must be taken to insure that S_{ami} and eqs. (30) and (31), when applied, correspond to the inline component local vertical (axial) and lateral axes.

This method for inline piping component extended structure acceleration prediction is applicable only to those inline piping components where the extended structure of the inline component has a frequency greater than 20 Hz. The inline piping component frequency determination can be made by analysis, test, or judgement, and when making this frequency determination, the inline piping component body shall be or shall be assumed to be rigidly constrained.

(d) This method is limited to NPS 6 (DN 150) and smaller.

N-1225.2 Analysis Using the Seismic Load Coefficient Method. When seismic forces F_{phi} or F_v are determined from the equations of N-1225.1.1(a), they shall be assumed as statically applied inertia loads acting along three orthogonal axes defined relative to the piping system with one of these axes being in the vertical direction. The piping system seismic analysis shall be performed independently for each direction of excitation with the loads applied in the same sense. Resultant internal forces, moments or stresses in the piping products and piping supports determined from seismic loads in the three directions, may be combined by the square root sum of squares basis and will be considered as having both a (+) and (-) sign. In developing the piping system analytical models for this method, consideration shall be given to appropriately simulating the effects of localized concentrated mass effects assumed from such piping products as flanges, inline instrumentation, valves (including extended structures), etc.

The seismic forces defined by the equations of N-1225.1.1(a) consider only inertia effects. They do not include the effects of relative or differential motions between piping supports and motions of equipment to which the piping is attached. These effects are typically referred to as seismic anchor motions.

For this piping analysis method the piping supports may be assumed to be zero displacement points in the direction of restraint.

N-1226 Floor Spectrum Generation

(a) Nuclear facility structures are approximated by mathematical models to permit analysis of responses due to earthquake motions. Considering the large number of degrees of freedom that would be necessary and the possible ill-conditioning of the resulting stiffness matrix if the complete plant were idealized as a single mathematical model, the plant is usually separated into several separate subsystems for analysis purposes. There will usually be one or more primary structural models which support one or more secondary systems. Also, different models of the same structure may be required for different purposes. Specifically, the dynamic model used to generate the seismic excitation data for subsequent, separate analyses of the secondary systems may not be suitable for the detailed, localized stress analysis of the primary structure.

(b) Most equipment will have negligible interaction effects on the primary structure as in the case of equipment with relatively small mass and high frequency, and will only need to be included in the mass distribution of the primary system model. There are, however, major equipment systems, such as a reactor coolant system, whose stiffness, mass, and resulting frequency range should be considered for representation in the building model to account for possible dynamic interaction effects. Some guidelines for determining the extent of interaction are given in refs. [31] through [33].

(c) Equipment may be analyzed by combining the complete equipment model with the support structure model and applying the proper excitation to the base of the support structure. In this method no separate equipment support excitations need be generated because the equipment will be excited directly through the structure (ref. [34]).

(d) For most equipment a separate analysis of the secondary system may be performed using output from the building analysis. If building to equipment interaction is significant, then the equipment should be included in the mathematical model of the structure. The representation of the equipment that is included in the building model should be adequate to consider major interaction effects, but need not be as detailed as the mathematical model used in a separate analysis of the equipment.

(e) For equipment which is not analyzed as part of the building structural model, the response may be obtained by separate analysis using floor response spectra curves, time history excitations, or an appropriately defined power spectral density function at the support locations of the equipment in the structure.

N-1226.1 Response Spectra.

(a) Both horizontal and vertical response spectra may be computed from the time history motions of the structure at the various floors or other equipment support locations of interest (ref. [35]). The spectrum ordinates should be computed at sufficient frequency intervals to produce complete and accurate response spectra. Spectrum peaks

would normally be expected to occur at the frequencies of the peaks on the ground motion spectrum and at the natural frequencies of the supporting structures. In cases involving equipment mounted on equipment, the frequencies of all supporting structures should be included.

(b) Table N-1226-1 provides some systematic methods which may be used for choosing spectrum frequencies. Another acceptable method is to choose a set of frequencies such that each frequency is within 10% of the previous one and then add the natural frequencies of the supporting structures to the set.

(c) A more simplified acceptable method for constructing floor response spectra from the ground spectrum is given in refs. [36] and [37].

N-1226.2 Spectral Peaks. Studies (refs. [38] and [39]) have shown that, in general, the calculation of response of a piece of equipment having a frequency equal to a frequency of the supporting structure will be conservative if the spectrum at the equipment support is generated using a model of the supporting structure that does not include the equipment. This is because the exact resonance of an uncoupled analysis is not possible in the coupled case where the resonant frequencies tend to shift away from each other when the models are coupled. Further, it has been shown in ref. [39] that undamped peak amplification for singly supported equipment structures cannot exceed a maximum $\sqrt{J/m}$, where J is the modal mass for each mode of the primary structure and m is the mass of the supported subsystem. Since this limit is a function of the mass of the particular subsystem, the peak response may be calculated on a case-by-case basis. This rule is recommended only for relatively simple systems.

N-1226.3 Spectrum Peak Broadening.

(a) To account for the effect on structural frequency variation of the possible uncertainties in the material properties of the structure and soil, and the approximations in the modeling techniques used in seismic analysis, the initially computed floor response spectra are usually smoothed, and peaks associated with the structural frequencies are widened. A recommended method of determining the amount of peak widening, associated with the structural frequency, is described below.

(b) Let f_j be the j th mode structural frequency that is determined from the structure model. The variation in the structural frequency is determined by evaluating the individual frequency variation due to the variation in each parameter that is of significant effect, such as the soil modulus, material density, etc. The total frequency variation $\pm\Delta f_j$ is then determined by taking the SRSS of a minimum variation of $0.05f_j$ plus the individual frequency variation $(\Delta f_j)_n$, that is

$$\Delta f_j = \sqrt{(0.05f_j)^2 + \sum (\Delta f_j)_n^2}$$

(c) A value of $0.10f_j$ is recommended if the actual computed value of Δf_j is less than $0.10f_j$. Figure N-1226-1 shows a sample of such a smoothed floor spectrum curve. Note that the broadened peak is bounded on each side by lines which are not vertical but parallel to the lines forming the original spectrum peak.

(d) An alternative method for the broadening of the structural peaks can be based on a probabilistic approach, as discussed in ref. [40]. In the particular case where there is more than one equipment or piping frequency located within the frequency range of a widened spectrum peak that is associated with a structural frequency f_j , the floor spectrum curve may be more realistically applied in accordance with the following criterion. Based on the fact that the actual natural frequency of the structure can possibly

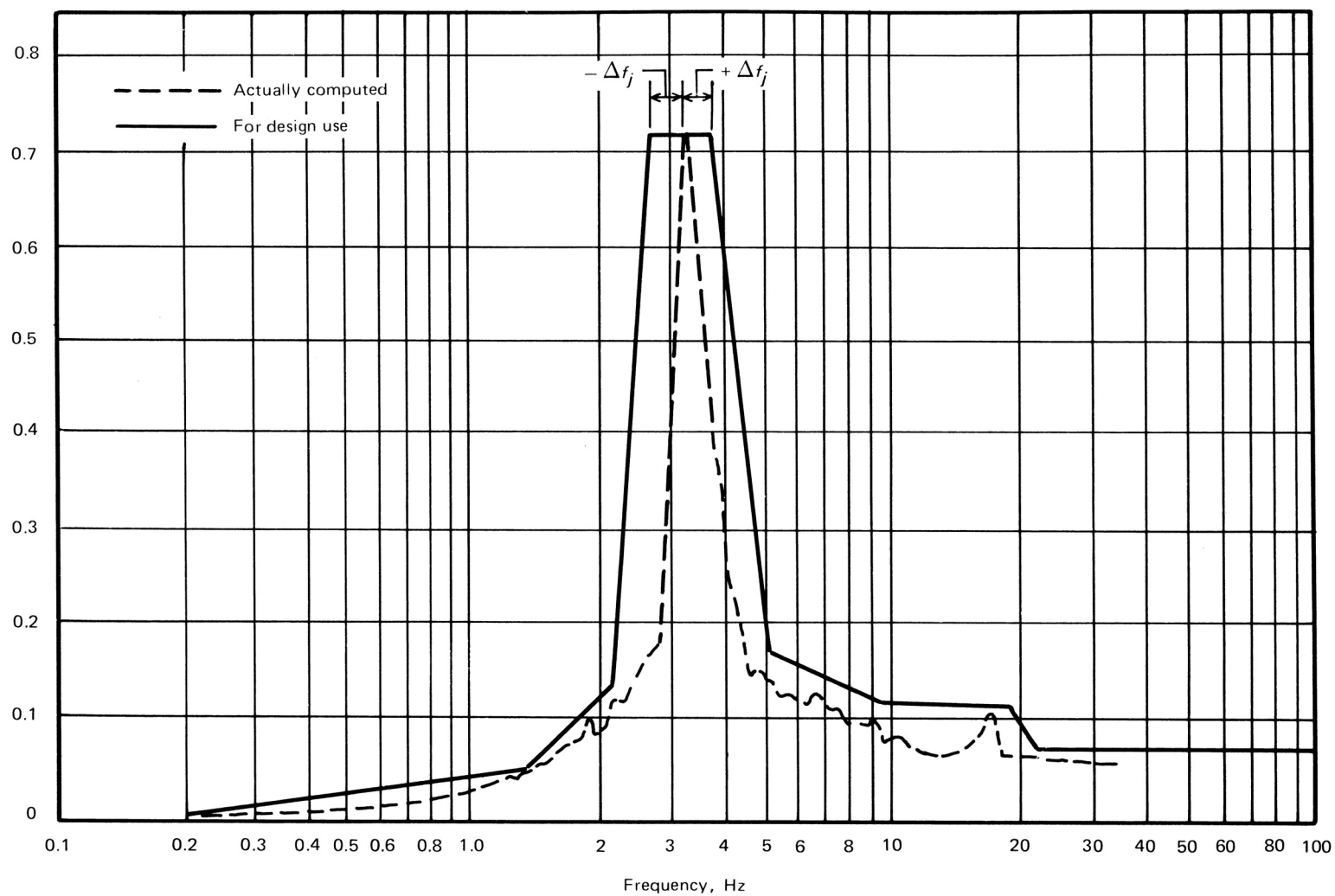
Table N-1226-1
Suggested Frequencies, Hz, for Calculation of Ground and Floor Response Spectra

Ground Spectra [Note (1)]		Floor Spectra [Note (2)]	
Frequency Range	Increment, Hz	Frequency Range	Increment, Hz
0.5–3.0	0.10	0.5–1.6	0.10
3.0–3.6	0.15	1.6–2.8	0.20
3.6–5.0	0.20	2.8–4.0	0.30
5.0–8.0	0.25	4.0–9.0	0.50
8.0–15.0	0.50	9.0–16.0	1.0
15.0–18.0	1.0	16.0–22.0	2.0
18.0–22.0	2.0	22.0–34.0	3.0
22.0–34.0	3.0		

NOTES:

- (1) Calculate response at all frequencies within the ranges shown, at the corresponding increments (results in 72 frequencies).
- (2) Calculate response at all frequencies within the ranges shown, at the corresponding increments (results in 46 frequencies); also calculate response at all natural frequencies of the supporting structures with the overall range.

Figure N-1226-1
Response Spectrum Peak Broadening and Peak Amplitude



assume only one single value within the frequency range defined by $f_j \pm \Delta f_j$, but not a range of values, only one of these equipment or piping modes can respond with a magnitude indicated by the peak spectral value.

Therefore, seismic analysis of the equipment or piping systems using the broadened floor design response spectra may be accomplished by the following alternative method which can be used for equipment when analytical techniques are justified for predicting the natural frequencies.

Determine the natural frequencies $(f_e)_n$ of the system to be qualified in the broadened range of the maximum spectrum acceleration peak.

If no equipment or piping system natural frequencies exist in the $\pm 15\%$ interval associated with the maximum spectrum acceleration peak, then the interval associated with the next highest spectrum acceleration peak shall be selected and used in the following procedure.

Consider all N natural frequencies in the interval

$$f_j - 0.15f_j \leq (f_e)_n \leq f_j + 0.15f_j$$

where

f_j = the frequency of maximum acceleration in the unbroadened spectra

$n = 1$ to N

The system shall then be evaluated by performing $N + 3$ separate analyses using the unbroadened floor design response spectrum and the unbroadened spectrum modified by shifting the frequencies associated with each of the spectral values by a factor of $+0.15$; -0.15 , and

$$\frac{(f_e)_n - f_j}{f_j}$$

where

$n = 1$ to N

The resultants of these separate seismic analyses shall then be enveloped to obtain the final resultant desired (e.g., stress, support loads, acceleration, etc.) at any given point in the system. If three different floor spectrum curves are used to define the response in the two horizontal and the vertical directions, then the shifting of the spectral values as defined above may be applied to these three spectrum curves.

The criterion is illustrated by the following example. Figure N-1226-2, sketch (a) represents the peak broadening on the floor spectrum curve associated with the j th mode structural frequency. Let there be two equipment or piping frequencies $(f_e)_1$ and $(f_e)_2$ that are within the frequency interval of $f_j \pm 0.15f_j$, Figure N-1226-2, sketch (b). Thus N equals two and therefore five separate floor response spectra must be considered. The unbroadened floor response spectrum as indicated in

Figure N-1226-2, sketch (a) would be the first floor spectrum considered. The unbroadened floor response spectrum modified by shifting all of the frequencies associated with the spectral acceleration values by a factor of -0.15 is illustrated in Figure N-1226-2, sketch (c). In Figure N-1226-2, sketches (d) and (e) illustrate the modifications made to the unbroadened floor response spectrum by shifting a factor of

$$\frac{(f_e)_1 - f_j}{f_j} \quad \text{and} \quad \frac{(f_e)_2 - f_j}{f_j}$$

respectively.

Figure N-1226-2, sketch (f) illustrates the modifications made to the unbroadened floor response spectrum by shifting by a factor of $+0.15$. For this example, the five separate seismic analyses are performed with the individually modified floor response spectrum curves.

The required resultants, stresses, support loads, accelerations, moments, forces, etc., associated with each of the five separate seismic analyses are then obtained. As any of the five modified spectra has an equal probability of occurrence, the resultant utilized to evaluate the seismic loadings shall be the envelope values from the five separate seismic analyses.

N-1227 Multiple Input Response Spectra Analysis

If a structural system is being analyzed by the response spectra method and is supported at intermediate locations with different characteristic response spectra, then the dynamic analysis performed should take into consideration the different response spectra. The effect of relative seismic anchor displacements between the intermediate anchor points should also be considered.

N-1227.1 Inertial Effects Due to Multiple Response Spectra Input.

The effect on a structural system of multiple supports with different characteristic response spectra may be accounted for by selecting a single spectra which will effectively produce the critical maximum responses due to different acceleration existing at anchor points and intermediate restraint points of the system. This may be conservatively accomplished by enveloping the response spectra for the different seismic anchor and restraint locations. Other procedures may be used, if applicable. Acceptable alternative procedures are given in refs. [7], [43], [44], and [45].

For a structural system which is restrained in different buildings, the response spectra of the restraint point should be enveloped.

N-1227.2 Stresses Due to Relative Seismic Support Displacements.

(a) When determining stresses the effects of relative seismic support movements should be considered. When these effects are considered significant, they may be obtained by performing a static structural analysis of the system including anchor movements. For structural systems

Figure N-1226-2
Use of Floor Spectra When Several Equipment Frequencies Are Within the Widened Spectral Peak

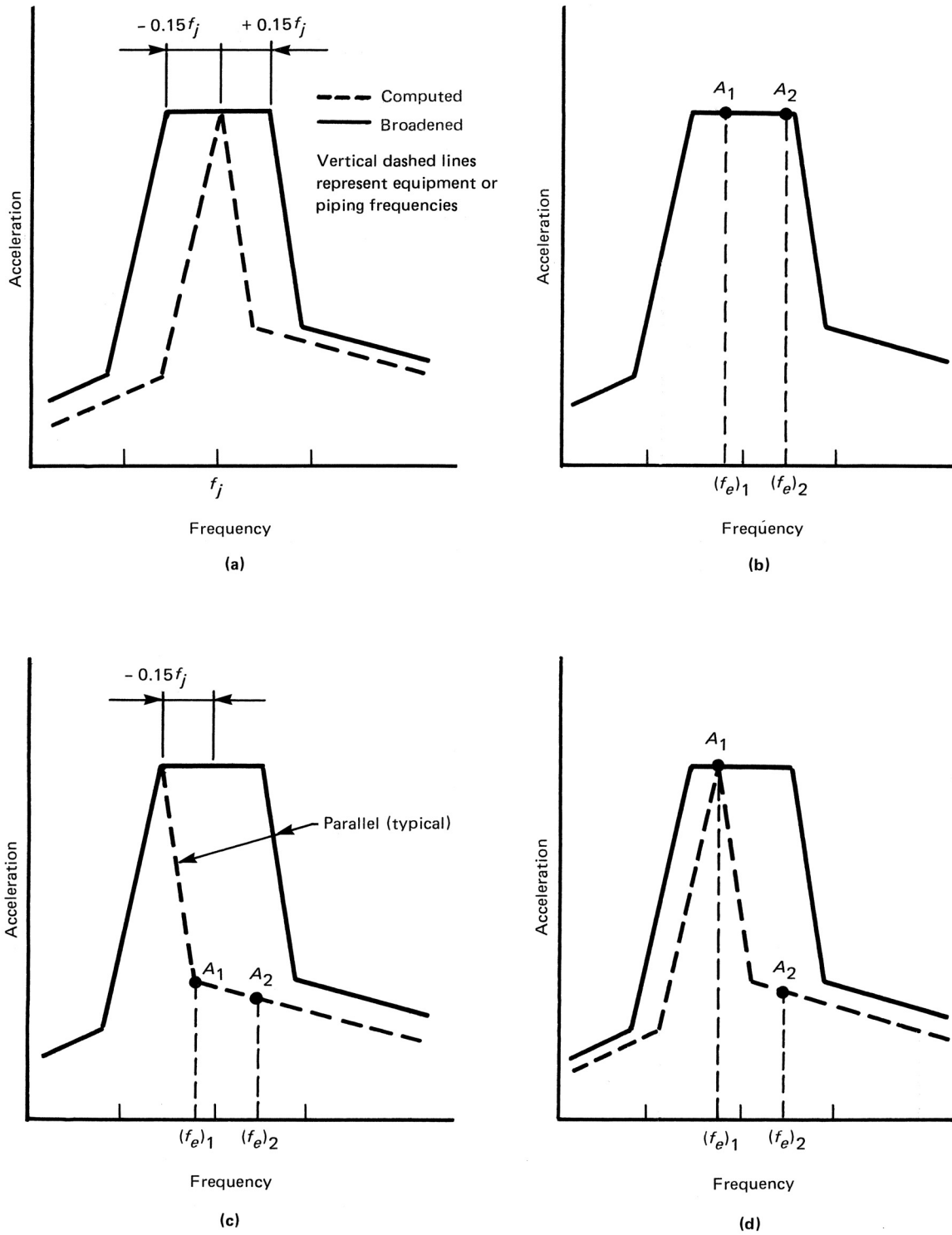
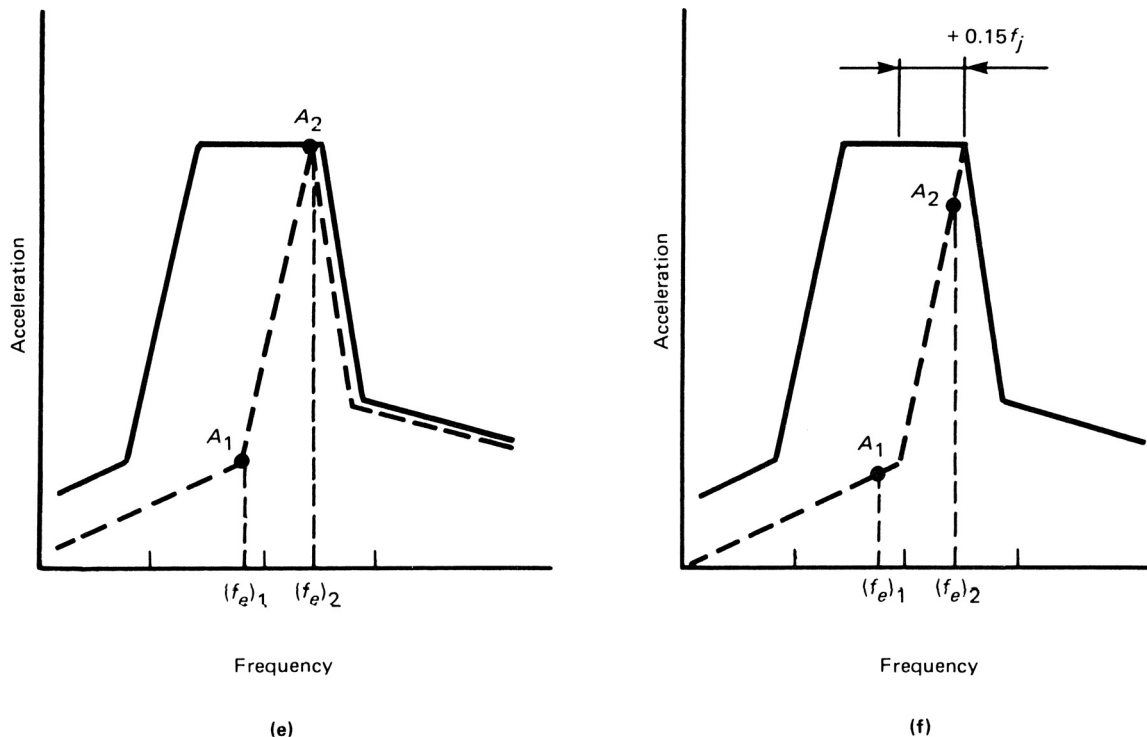


Figure N-1226-2
Use of Floor Spectra When Several Equipment Frequencies Are Within the Widened Spectral Peak (Cont'd)



with several possible combinations of anchor movements, any of the following methods may be used for determining the stresses due to relative support displacements.

(1) Stresses resulting from the differential movement of supports may be calculated for each significant mode using the modal displacement of the supports obtained from the structural response calculations. The maximum model responses are then combined following any of the acceptable methods described in this Appendix. If no support structure modal information is available, then the absolute sum of the displacements may conservatively be used.

(2) If the system is supported on independent structures, then the absolute sum of displacements in each direction should be used unless more detailed analyses are used in accordance with N-1700.

(3) If a displacement time history of the supporting structure is available, the procedures given in N-1228.4 may be used.

(b) Other detail considerations may be specified for dynamic analysis of Section III components, such as NB-3600 for piping, and these component related rules should be considered in the dynamic analysis.

N-1228 Multiple Time History Excitations

For structural systems supported at several locations, the responses due to simultaneous excitations by different motions at each support may be determined exactly by time history methods, as in N-1228.1. Having performed a time history response analysis, reaction loads and motions may also be determined on a time history basis, as in N-1228.2. Since the inertial effects may be considered to produce primary stresses and the differential support motions may be considered to produce secondary stresses, as in NB-3650, the inertial effects may be determined separately from the differential support movement effects, as in N-1228.3.

N-1228.1 Time History Response Analysis.

(a) Several different acceptable methods have been presented in the literature (refs. [7], [43], and [44]) for solving the equations of motion for a structural system excited by multiple simultaneous forcing functions. Since the basic problem to be solved must satisfy the fundamental principles of dynamics, the differences between the various methods are primarily in the approach of the solution process. One method of solution is summarized here to provide insight into the basics of time history analysis for multiple excitations.

(b) The general matrix form of the undamped coupled equations of motion may be written (refs. [7] and [45]) as follows:

$$MX'' + KU = F \quad (32)$$

where X'' represents the absolute acceleration of the mass point dynamic degrees of freedom, and U represents those displacements of the mass and support point dynamic degrees of freedom which would tend to cause distortions in the system. Rigid body translations or rotations are included in the mass acceleration terms on the left-hand side of the equation. Such rigid body motions would not tend to distort the system and accounted for by defining one of the system support points to be the *datum* support, and describing the motion of all other *nondatum* supports as motions relative to the datum support.

(c) Equation (b)(32) can be expanded to give

$$\begin{bmatrix} M_m & 0 \\ 0 & M_s \end{bmatrix} \begin{Bmatrix} X''_m \\ X''_s \end{Bmatrix} + \begin{bmatrix} K_{mm} & K_{ms} \\ K_{sm} & K_{ss} \end{bmatrix} \begin{Bmatrix} U_m \\ U_s \end{Bmatrix} = \begin{Bmatrix} 0 \\ F_s \end{Bmatrix} \quad (33)$$

where

F_s = the reaction forces at the system support points due to the response of the system to the motion of the support structure

K = the stiffness matrix of the system model condensed in a manner such that only mass point elements (subscript m) and active, nonreleased, nondatum support elements (subscript s) remain in the matrix

M_m = a diagonal submatrix of the system model lumped masses

M_s = a submatrix of inertia terms associated with the support joints of the system. For the purposes of this analysis, $M_s = 0$ because there is no mass lumped at support joints

U_m = displacement of mass point dynamic degrees of freedom relative to the datum support in each coordinate direction

U_s = displacement of nondatum support points relative to the datum support

X''_m = absolute acceleration of mass point dynamic degrees of freedom of the model

X''_s = absolute acceleration of the system support points

(d) The time history support motions imposed at the nondatum supports include only such displacements as would tend to cause distortions in the system. Rigid body translation or rotation of the supporting structure would not distort the system being analyzed; therefore, such

rigid motions of the supporting structure are removed by computing the nondatum support relative motions as follows:

$$U_s = X_s - X_d - R_s\theta \quad (34)$$

where

R_s = a matrix of three vectors representing distances from the datum support point of the nondatum support points

U_s = displacements of nondatum support points relative to the datum support in each direction of excitation

X_d = absolute displacements of datum supports in each direction of excitation

X_s = absolute displacements of nondatum supports in each direction of excitation

θ = rigid body rotations of the supporting structure about each of the three coordinate axes

(e) The first equation of the set of eq. (c)(33) yields:

$$M_m X''_m + K_{mm} U_m + K_{ms} U_s = 0 \quad (35)$$

(f) A separation of variables can be achieved by defining the absolute acceleration of a mass point in terms of acceleration relative to the datum support, such that:

$$U''_m = X''_m - \gamma X''_{sd} - R_m \theta'' \quad (36)$$

where

R_m = a matrix of three vectors representing distances from the datum support point to the mass points for each of the rotational components

X''_{sd} = the absolute accelerations of the datum support in each coordinate direction

γ = matrix defining the direction of each respective translational dynamic degree of freedom

where

$\gamma_{ij} = 1$, if the i th dynamic degree of freedom is in the direction of the j th direction of support translation

$\gamma_{ij} = 0$, if the i th dynamic degree of freedom is not in the direction of the j th direction of support translation

θ'' = rotational accelerations of the datum support

(g) Equation (e)(35) then becomes

$$M_m U''_m + K_{mm} U_m = -M_m X''_{sd} - M_m R_m \theta'' - K_{ms} U_s \quad (37)$$

(h) At this point it is to be noted that the equations of motion are in a form expressing three dimensional response of the system mass points due to multiple translational and rotational support excitations in one or more coordinate directions. Normally, the rotations will consist of two components representing rocking about two

orthogonal horizontal coordinate axes. The equations, however, are also valid for rotation about the vertical axes. It should be noted again that U_m and U_s are only the distortional portions of the total mass and support motions.

(i) The coupled equations of motion, eq. (g)(37), may be solved by one of the methods of direct integration, or, introducing the normal mode coordinate transformation:

$$U_m = \phi q \quad (38)$$

where

ϕ = the matrix of eigenvectors of the supported system
 q = the normal mode coordinate

(j) Then making the usual assumption of proportional damping for the modes of the supported system, the equations of motion can be uncoupled and written in the following form:

$$q'' + 2\xi\omega q' + \omega^2 q = -(\phi^t M_m \phi)^{-1} (\phi^t M_m \gamma X''_{sd} + \phi^t M_m R_m \theta'' + \phi^t K_{ms} U_s) \quad (39)$$

where

ω^2 = diagonal matrix of eigenvalues
 $2\xi\omega$ = diagonal matrix of modal damping terms

(k) The right-hand side of eq. (j)(39) contains no damping terms in a rigorous sense only if the modal damping is proportional to stiffness. However, Ref. [46] points out that the contribution of these terms to earthquake forces can be expected to be small. This term is usually neglected for systems exhibiting structural damping characteristics within the range of the values in Table N-1230-1.

N-1228.2 Time History Reaction Analysis.

(a) Having solved the dynamics problem for the time histories of the mass point motions, the resultant reactions at locations throughout the system can be determined for each time step by imposing the deflections on the structure in a series of static problems; one static solution for the deflected shape at each time step. Although this approach is analytically valid, it is generally not practical because of the many thousands of solutions required for the number of time steps in a typical seismic analysis. The problem may be reduced to a single static deflection analysis by applying unit displacements to each mass and support point to determine a set of influence coefficients for each desired reaction. The given support displacements and computed mass point displacements at each time step are multiplied by the set of influence coefficients to perform a complete reaction analysis of the system at each time step.

(b) The desired components of reaction (force, moment, stress, or deflection) are computed at each time step as follows:

$$R(t) = C_m U_m(t) + C_s U_s(t) \quad (40)$$

where

C_m = a matrix of mass point unit displacement influence coefficients (one column per mass point and one row per reaction component)

C_s = a matrix of nondatum support point unit displacement influence coefficients (one column per nondatum support and one row per reaction component)

$R(t)$ = a vector of reaction components at time t

$U_m(t)$ = a vector mass point relative displacements at time t

$U_s(t)$ = a vector of nondatum support relative displacements at time t

(c) In a similar manner, the absolute acceleration of any point in the system may be computed by multiplying the mass point and support relative accelerations by the influence coefficients for displacement reactions, and adding in the datum support rigid body rotational and translational absolute accelerations as follows:

$$R''(t) = \gamma X''_{sd}(t) + C_m U''_m(t) + C_s U''_s(t) + R_p \theta''(t) \quad (41)$$

where

C_m = a matrix of mass point unit displacement influence coefficients for components of displacement reactions

C_s = a matrix of nondatum support unit displacement influence coefficients for components of displacement reactions

R_p = a matrix containing distances to the reaction point from the datum

$R''(t)$ = a vector of absolute acceleration components at time t

$U''_m(t)$ = a vector of mass point accelerations relative to the datum at time t

$U''_s(t)$ = a vector of nondatum support relative accelerations at time t

$X''_{sd}(t)$ = the translational absolute acceleration of the datum support at time t

γ = a vector defining the direction of excitation,

where

$\gamma_{ij} = 1$, if the i th component of reaction is in the direction of j th support motion
 $= 0$, if the i th component of reaction is not in direction of the j th support motion

$\theta''(t)$ = the rotational absolute acceleration of the datum support at time t

(d) This method, therefore, permits the calculation of any desired force or moment, or nonmass point motion on a time history basis.

(e) The ability to calculate time history acceleration of nonmass locations in the system permits the subsequent generation of response spectra at any arbitrary point in the system without the necessity of lumping mass at these points.

N-1228.3 Separation of Inertial and Relative Anchor Movement Stress Effects.

(a) The terms of the C_s coefficients matrix can be found by applying a unit displacement to each nondatum support in turn, holding all other supports and mass points fixed, and calculating the desired reaction at a given location. The reactions found in this manner are the combined effect of inertial and relative displacement stress producing effects, suitable for meeting the appropriate Code requirements, such as eq. (10) of NB-3653.1 for Class 1 piping.

(b) Alternatively, one may break up the C terms into two terms C_{s1} and C_{s2} such that

$$R(t) = C_m U_m(t) + C_{s1} U_s(t) + C_{s2} U_s(t) \quad (42)$$

or

$$R_1(t) = C_m U_m(t) + C_{s1} U_s(t) \quad (43)$$

and

$$R_2(t) = C_{s2} U_s(t) \quad (44)$$

where

C_{s1} = a matrix of influence coefficients due to holding all supports fixed and applying the set of forces to all mass points which would be required to hold the mass point in place due to a unit displacement of each nondatum support in turn

C_{s2} = a matrix of influence coefficients to a unit displacement of each nondatum support in turn, allowing all mass points to respond freely as static degrees of freedom in the structure

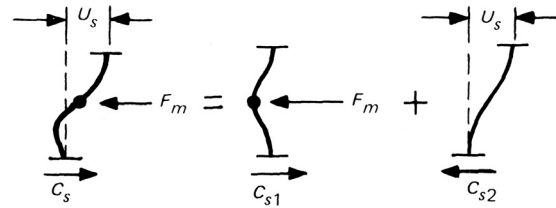
$R_1(t)$ = a vector of stress reaction components at time t due to inertial responses

$R_2(t)$ = a vector of stress reaction components at time t due to relative support motions

(c) The coefficients for a component of shear for a unit displacement of a nondatum support may be visualized as shown in Figure N-1228.3-1.

(d) The C_{s1} term therefore represents inertial effect produced by the differential support motion, and the C_{s2} term represents the relative displacement effect of the differential support motion. The separate stresses

Figure N-1228.3-1
Coefficients for a Component of Shear for a Unit Displacement of a Nondatum Support



determined in this manner may be used to meet the appropriate Code requirements, such as eq. (9) of NB-3652 for Class 1 piping.

N-1228.4 Envelope Excitations.

(a) Where the responses are calculated using a single time history excitation for a multiply supported system, the response spectrum of the single time history used should envelop the spectra for the individual support motions. The secondary stress effects of differential support motions may then be determined by a separate static analysis where either

(1) the set of maximum displacements over all time for each support are applied simultaneously in a single analysis, or

(2) the displacements of each support are applied to perform an analysis for each time step, accounting for sign.

(b) Where the maximum displacements over all time are used in a single analysis, then

(1) for systems supported at multiple locations on the same floor/wall where one flexural mode dominates, the relative displacement set for the dominant mode may be used, where justified

(2) for systems supported at multiple locations on different floor/walls or on different structures where different flexural modes can dominate, the displacement may be applied statically as prescribed in N-1227

(c) Note that, in general, different support displacement sets may be required to obtain conservative secondary stress effect at different locations within a system, depending on the complexity of a given system.

N-1230 DAMPING

N-1231 Damping Values

(a) Motions in a structural system will dissipate energy from the system. The phenomenon of this kind of energy loss is called damping. In a structural system, sources of energy loss may be due to: a structural damping, which is caused by internal friction within the material or at connections between elements of a structural system; a viscous damping, which is caused by motions in a fluid; Coulomb damping which results from the sliding friction

motion of a body on another surface. The following equation of motion is generally used for structural systems and components

$$[M]\{X''\} + [D]\{X'\} + [K]\{X\} = \{F\} \quad (45)$$

where

D = damping force vector
 F = force vector
 K = stiffness matrix
 M = mass matrix
 X = displacement vector
 X' = velocity vector
 X'' = acceleration vector

(b) The damping force $\{D\}$ represents the energy losses of the system. To evaluate these losses, the following alternative mathematical expressions are recommended for the vector $\{D\}$.

(1) Viscous damping:

$$\{D\} = [C]\{X'\} \quad (46)$$

where $[C]$ is a viscous damping matrix. The damping force is proportional to the velocity.

(2) Hysteretic damping:

$$\{D\} = iv[K]\{X\} \quad (47)$$

where v represents the structural damping factor (ref. [47]). This damping is also called complex damping. The damping force is proportional to the amplitude of the displacement and opposite in direction to the velocity:

$$\{D\} = g[abs[K]\{X\}] sgn\{X'\} \quad (48)$$

where g is a damping coefficient (ref. [48]). This damping is also called the Reid's damping. The damping force is proportional to the resisting force due to deformation and in the opposite direction of the velocity in harmonic motion.

(3) Coulomb damping:

$$\{D\} = \pm \mu\{N\} \quad (49)$$

where μ is the coefficient of friction and N is the normal force vector. The sign is chosen to be opposite the velocity. The damping force is due to friction and proportional to the normal force.

(4) General dashpot damping:

$$\{D\} = \alpha\{X'\}[abs\{X'\}]^{n-1} \quad (50)$$

where α is a damping constant and n is an integer. This damping force is due to the turbulence and the dashpot geometry.

(c) These expressions permit the simulation of the energy loss for a particular problem. It is extremely difficult to give an accurate analytical expression for the different

forms of damping in complex structural systems. The analyst may select a simple expression, such as viscous or hysteretic damping, to solve the problem by assuming that the energy losses are equivalent; i.e., viscous damped system introduced in the analysis will have the same energy loss per cycle as the real structure.

(d) To establish an equivalent viscous damping matrix $[C]$ for the system, experimental data is usually required. Important experimental results, which indicate that energy losses increase with stress or displacement amplitude, or both, have been compiled in existing literature (refs. [49] and [50]). The incorporation of this behavior to the equivalent viscous damping in the transient analysis will usually lead to an expression of the damping force which will make the equations of motion nonlinear. To avoid this inconvenience, conservative modal damping values may be used for nuclear systems and structures for OBE and SSE condition loads, respectively. Table N-1230-1 shows the modal damping values recommended. Damping values higher than the ones given in Table N-1230-1 may be used in a dynamic analysis if the basis is justified.

(e) Methods of incorporating the damping in structural dynamics are usually dependent on the mathematical convenience. The energy loss per cycle is simulated by a convenient mathematical scheme; frequently viscous damping expressions are adopted in the analysis. In N-1232 and N-1233, methods of proportional and nonproportional dampings are discussed and the physical interpretations associated with each method is also examined.

N-1232 Proportional Damping

(a) Consider a viscously damped system of the following form:

$$[M]\{X''\} + [C]\{X'\} + [K]\{X\} = \{F\} \quad (51)$$

Table N-1230-1
Damping Values
Percent of Critical Damping

Structure or Component	Earthquake Magnitude	
	Operating Basis Earthquake	Safe Shutdown Earthquake
Equipment	2	3
Piping systems	5	5
Welded steel structures	2	4
Bolted steel structures	4	7
Prestressed concrete structures	2	5
Reinforced concrete structures	4	7

Let $[\phi]$ be the modal matrix of the undamped system of

$$[M]\{X''\} + [K]\{X\} = \{0\} \quad (52)$$

such that

$$[\phi]^T[M][\phi] = [I] \quad (53)$$

and

$$[\phi]^T[K][\phi] = [\omega^2]$$

where

$[\phi]^T$ = transpose of the matrix $[\phi]$

I = identity matrix

ω^2 = diagonal matrix

ω_i = circular frequencies

(b) If the matrix $[C]$ in eq. (a)(51) can be diagonalized by the undamped normal modes (i.e., $[\phi]^T[C][\phi]$ is a diagonal matrix), then the damping matrix $[C]$ is called the proportional damping matrix; otherwise, it is called the nonproportional damping matrix.

(c) Reference [52] has illustrated that the matrix $[C]$ can be diagonalized when it is a linear combination of $[M]$ and $[K]$ matrices. Caughey (ref. [53]) has derived more general conditions under which the matrix $[C]$ can be diagonalized.

(d) For proportional damping the equations of motion may be uncoupled into a set of independent one degree of freedom systems. In this case, the damping ratio associated with the different modes can be determined.

(e) Among the category of proportional dampings, two types of damping matrix $[C]$ are commonly adopted; namely, the mass and stiffness damping and the orthogonal modal damping. Both types of damping are discussed in N-1232.1 and N-1232.2.

N-1232.1 Mass and Stiffness Damping (Ref. [53]).

(a) In this damping, the matrix $[C]$ is assumed to be proportional either to the mass matrix $[M]$ or to the stiffness matrix $[K]$ or to a linear combination of the two. That is, it can be written as follows:

$$[C] = \alpha[M] + \beta[K] \quad (54)$$

where

α and β = two real constants

$\alpha[M]$ = mass damping

$\beta[K]$ = stiffness damping

(b) In using the mass and stiffness damping, damping values of the entire system are determined by the two constants α and β . To determine the value of α and β , one can control the damping ratios for two frequencies.

(c) Let ω_r and ω_s be the two frequencies that we want to have a damping ratio of ξ_r and ξ_s , respectively. Then α and β may be determined (ref. [54]) as follows:

$$\begin{aligned} \alpha &= 2\omega_r\omega_s(\xi_s\omega_r - \xi_r\omega_s) / (\omega_r^2 - \omega_s^2) \\ \beta &= 2(\xi_r\omega_r - \xi_s\omega_s) / (\omega_r^2 - \omega_s^2) \end{aligned} \quad (55)$$

Then, for an arbitrary frequency ω_i the damping ratio ξ_i can be computed by eliminating α and β and is given as:

$$\xi_i = \frac{1}{(\omega_r^2 - \omega_s^2)} \left[\frac{\omega_r\omega_s}{\omega_i} (\xi_s\omega_r - \xi_r\omega_s) + \omega_i(\xi_r\omega_r - \xi_s\omega_s) \right] \quad (56)$$

where

$$\frac{1}{(\omega_r^2 - \omega_s^2)} \times \frac{\omega_r\omega_s}{\omega_i} (\xi_s\omega_r - \xi_r\omega_s)$$

is due to mass damping and

$$\frac{1}{(\omega_r^2 - \omega_s^2)} \times \omega_i(\xi_r\omega_r - \xi_s\omega_s)$$

is due to stiffness damping.

(d) The application of the mass and stiffness damping concept is illustrated by a following numerical example. Consider a structural system with frequencies of $f_n = n$ Hz for $n = 1, 2, \dots, 25$. For a damping ratio of $\xi_{10} = 0.05 = \xi_{20}$ at frequencies $f_{10} = 10$ Hz and $f_{20} = 20$ Hz and by using eq. (c)(56), the damping ratios for the other modes may be computed. For example, at the frequencies of 5, 15, and 25 Hz, the corresponding damping ratios are $\xi_5 = 0.075$, $\xi_{15} = 0.047$, and $\xi_{25} = 0.055$.

(e) It should be noted that in eq. (c)(56), the damping ratio is the sum of the contributions due to the mass and stiffness damping. Using mass damping $\alpha \neq 0$, $\beta = 0$, the damping ratio will decrease with increase of the frequency. On the other hand, using stiffness damping $\alpha = 0$, $\beta \neq 0$, the damping ratio will increase with the increase of the frequency.

(f) Mass damping introduces the damping forces that are proportional to the velocities of each mass point in the system. Mass damping may be used to represent the energy loss due to impact and friction. Stiffness damping introduces the damping forces that are proportional to the time rate of deformation. Stiffness damping can be used to represent the structural damping. While the mass damping introduces the damping forces due to the rigid body motion (displacements without deformations) of the system, the stiffness damping does not.

(g) In seismic analysis of a structural system, the equation of motion is generally formulated in terms of the displacements with respect to the base. If the damping ratios

are established for the two significant frequencies, the mass and stiffness dampings may be adopted. The damping ratios for the other frequencies are computed with the aid of eq. (c)(56). If only stiffness damping is used, and for the significant frequency Ω_r , the established damping ratio is ξ_r , then

$$\beta = \frac{2\xi_r}{\omega_r} \text{ and } \xi_i = \xi_r \frac{\omega_i}{\omega_r}$$

where ξ_i is the damping ratio for the modal frequency ω_i .

N-1232.2 Orthogonal Modal Damping (Ref. [54]). In the orthogonal modal damping method, the damping matrix $[C]$ in eq. N-1232(a)(51) is not formulated in advance. After solving the eigenvalue problem, the system or uncoupled equations are assumed to have the form

$$q''_i + \xi_i(2\omega_i)q'_i + \omega_i^2 q_i = f_i \quad (57)$$

$$i = 1, 2, \dots, N$$

where the second term in eq. (57) is added to account for the damping. The damping ratio for each frequency is free to be determined by the analyst without any mathematical restriction. The damping ratio assigned to one mode will have no effect on the damping ratios of the others; the modes are orthogonal. These ratios can be obtained from the experimental results. This method is used often in solving viscously damped systems. The example problems for the various damping methods discussed here are given in ref. [55].

N-1233 Nonproportional Damping

In the foregoing discussion of the proportional damping (N-1232), the damping of a structural system was assumed to have a form that couplings do not exist between the classical modes of vibrations; i.e., the mode shapes obtained from the solution of a free vibration of the undamped system. This approach is appropriate for the correlation of test data for a structural system composed of a single material, or damping mechanism which is homogeneous throughout the system. However, for a system which is composed of different materials, such as a reactor coolant loops, containment building, and soil foundation, the damping mechanism of one type of material may be considerably different from the other. In order to have a better simulation of the damping, it is useful to construct a damping matrix which reflects the material composition rather than the mathematical convenience (refs. [56] and [57]). The damping matrix so constructed will no longer be diagonalized by the classical modes. The term *nonproportional damping* is used for this type of damping. In nonproportional damping, the matrices $[M]$, $[K]$, and $[C]$ are used to solve the complex frequencies and mode shapes. From the complex frequencies, damping ratios associated with each mode can readily be computed. Unlike proportional damping, the matrix $[C]$ of a nonproportional damping system cannot be formulated simply by

specifying a set of damping ratios, unless a mathematical process of successive approximations is performed. Due to the mathematical complexity involved in the complex frequencies and mode shapes solution, the process of assigning a set of damping ratios with respect to the complex modes becomes impractical. For certain types of structures such as reactor coolant loops, the system exhibits a pronounced modal behavior (ref. [58]). Instead of using a set of damping ratios based on the complex frequencies and modes, the classical modes and frequencies may be used to avoid the mathematical complexity. N-1233.1 and N-1233.2 describe acceptable methods for the formulation of a nonproportional damping matrix for a structural system which is composed of different materials.

N-1233.1 Composite Mass Damping and Stiffness Damping.

(a) In the proportional damping, eq. N-1232(a)(51), the system damping matrix is assumed to be a linear combination of the system mass and stiffness matrices, without any distinction of the material damping property. Different sets of proportional constants for mass and stiffness may be assumed to formulate a system damping matrix. Using the concept of mass and stiffness damping, the energy dissipation function D for the system can be written as follows:

$$D = \frac{1}{2} \sum_{i=1}^{nc} \left(\alpha_i \{X'\}_i^T [M]_i \{X'\}_i + \beta_i \{X'\}_i^T [K]_i \{X'\}_i \right) \quad (58)$$

where

$[K]_i$ = stiffness matrix of subsystem i

$[M]_i$ = mass matrix of subsystem i

nc = number of subsystems

α_i = mass proportional constant of subsystem i

β_i = stiffness proportional constant of subsystem i

For the systems with constant damping values, α_i and β_i are the same for all elements. However, when the system is composite, which consists of subsystems with different damping values, α_i and β_i are different for different subsystems but constant within each subsystem. These constants may be established for each subsystem as discussed in N-1232.1.

(b) This decomposition to allow different damping constants α_i and β_i to be used for each different subsystem may be based on the damping values recommended in Table N-1230-1.

(c) To determine the constants α_i and β_i , the frequencies and modes of the free undamped vibration of the i th subsystem are used as a basis. The mathematical process involved is similar to the one of stiffness and mass damping described in N-1232.1 except the constants α_i and β_i are different for each material or subsystem.

(d) Using eq. (a)(58), the equation of motion of a viscously damped system has the following form:

$$\begin{bmatrix} M \end{bmatrix} \begin{Bmatrix} X' \end{Bmatrix} + \sum_{i=1}^{nc} \left(\alpha_i [M]_i + \beta_i [K]_i \right) \begin{Bmatrix} X' \end{Bmatrix} + \begin{bmatrix} K \end{bmatrix} \begin{Bmatrix} X \end{Bmatrix} = \begin{Bmatrix} F \end{Bmatrix} \quad (59)$$

(e) equation (d)(59) may be integrated directly to obtain the response. If normal mode method is used, eq. (d)(59) may be solved by neglecting the modal coupling effect to decouple the equations of motions on the normal mode basis.

(f) Let $\{\phi\}$ be the modal matrix of the undamped system of eq. (d)(59) and $\{\phi_j\}$ be the j th modal vector. With the aid of the orthogonality conditions, the equation of motion for the j th mode can be written in the following form:

$$\begin{aligned} & \{\phi_j\}^T [M] \{\phi_j\} q''_j + \{\phi_j\}^T \sum_{i=1}^n \left(\alpha_i [M]_i + \beta_i [K]_i \right) \{\phi_j\} q'_j \\ & + \{\phi_j\}^T [K] \{\phi_j\} q_j = \{\phi_j\}^T \{F\} \end{aligned} \quad (60)$$

where q''_j is the j th generalized coordinate. The second term on the left-hand side of eq. (60) contains the coupling terms. If these coupling terms are neglected, eq. (60) reduces to the following equation:

$$\begin{aligned} & \{\phi_j\}^T [M] \{\phi_j\} q''_j + \{\phi_j\}^T \sum_{i=1}^n \left(\alpha_i [M]_i + \beta_i [K]_i \right) \{\phi_j\} q'_j \\ & + \{\phi_j\}^T [K] \{\phi_j\} q_j = \{\phi_j\}^T \{F\} \end{aligned} \quad (61)$$

or

$$\begin{aligned} & q''_j + \frac{\sum_{i=1}^{nc} \{\phi_j\}^T \alpha_i [M]_i \{\phi_j\}}{2w_j \{\phi_j\}^T [M] \{\phi_j\}} + \frac{\sum_{i=1}^{nc} w_j \{\phi_j\} \beta_i [K]_i \{\phi_j\}}{2\{\phi_j\}^T [K] \{\phi_j\}} q_j \\ & + w_j^2 q_j = \frac{\{\phi_j\}^T \{F\}}{\{\phi_j\}^T [M] \{\phi_j\}} \end{aligned} \quad (62)$$

Consequently, the effective damping for the j th mode is as follows:

$$\xi_j = \frac{\sum_{i=1}^{nc} \{\phi_j\}^T \alpha_i [M]_i \{\phi_j\}}{2w_j \{\phi_j\}^T [M] \{\phi_j\}} + \frac{\sum_{i=1}^{nc} w_j \{\phi_j\} \beta_i [K]_i \{\phi_j\}}{2\{\phi_j\}^T [K] \{\phi_j\}} \quad (63)$$

or

$$\begin{aligned} \xi_j &= \frac{\sum_{i=1}^{nc} \alpha_i \left(\frac{\text{max. kinetic energy for } i\text{th}}{\text{subsystem in the } j\text{th mode}} \right)}{2w_j \left(\frac{\text{max. kinetic energy for the system}}{\text{subsystem in the } j\text{th mode}} \right)} \\ &+ \frac{\sum_{i=1}^{nc} \beta_i \left(\frac{\text{max. strain energy for } i\text{th}}{\text{subsystem in the } j\text{th mode}} \right)}{2 \left(\frac{\text{max. strain energy for the system}}{\text{subsystem in the } j\text{th mode}} \right)} \end{aligned} \quad (64)$$

(g) Equation (f)(63) can be greatly simplified if either α_i or β_i is assumed to be zero. This assumption may be used in the structural analysis for seismic loads where damping is generally assumed proportional either to the kinetic energy distribution of the subsystems or to the strain energy stored.

(h) Once the damping ratio ξ_j is known, the analysis can be carried out using a time history or response spectrum technique.

N-1233.2 Subregional Modal Damping.

(a) In dynamic analysis of soil-structure interaction, the modal damping value of the soil is often higher than that of the superstructure. In addition, the modal damping values of the various subsystems, building, loop, etc., of the superstructure may be different. There are several acceptable approximate methods to simulate this subregional modal damping characteristic. These methods are briefly discussed here. One method from ref. [15] is to relate composite damping to absolute displacement eq. (65), while another method (ref. [59]) is to relate it to relative displacement eq. (66):

$$\xi_j = \frac{\sum_{i=1}^{nc} \{\phi_j\}^T \beta_i [M]_i \{\phi_j\}}{\{\phi_j\}^T [M] \{\phi_j\}} \quad (65)$$

$$\xi_j = \frac{\sum_{i=1}^{nc} \{\phi_j\}^T \beta_i [K]_i \{\phi_j\}}{\{\phi_j\}^T [K] \{\phi_j\}} \quad (66)$$

where

- $[K]$ = total system stiffness matrix
- $[K]_i$ = stiffness matrix of i th subsystem
- $[M]$ = total system mass matrix
- $[M]_i$ = mass matrix of i th subsystem
- nc = number of subsystems
- $\{\phi_j\}$ = j th eigenvector
- β_i = % critical damping associated with i th subsystem
- ξ_j = % critical damping for j th mode

(b) Another acceptable approximate method employs mass and stiffness damping for each subsystem. This method calculates coefficients α and β , eq. N-1232.1(c)(55), for a subsystem such that its modal damping is simulated in the frequency range of interest.

The damping matrices of all the subsystems are assembled to formulate a total system damping matrix. Another acceptable approach to generate a total damping matrix is discussed in ref. [62].

(c) All the methods discussed above are approximate. More accurate techniques are also acceptable. A more rigorous method to formulate a damping matrix for a total system is based on the modal synthesis procedure. This procedure is explained by Hurty (ref. [56]). Various investigators have developed another method (ref. [63]) modifying this procedure. In this procedure, the total system having nonproportional damping is divided into subsystems, each having proportional damping. Each subsystem is represented by its modal mass, damping, and stiffness matrices. As each subsystem has a proportional damping, the associated modal damping matrix is formulated without any approximation. The modal matrices of all the subsystems are assembled and the displacement compatibility at subsystem interfaces are satisfied. The resulting equations of motion are solved and with the aid of a transformation matrix, actual response is obtained. References [57], [64], and [65] give the details of the application of this method.

(d) The total damping matrix developed by the preceding methods represents the viscous damping. An acceptable alternative formulation for damping, derived in ref. [66], is an equivalent modal damping ratio which represents the viscous damping in the swaying spring and the hysteretic damping in the rest of the soil-structure system (the superstructure and the rocking spring).

N-1234 Linear Hysteretic Damping

(a) In this paragraph the linear hysteretic damping of the Reid's model (ref. [48]) will be discussed. This nonlinear model leads to well posed mathematical problems for transient and steady-state oscillations of all kinds. The damping forces of the Reid's model are assumed to be proportional to the internal forces of the structure but in phase with the associated velocities. By using the Reid's damping model, the equation of motion [eq. N-1231(a)(45)] becomes:

$$\begin{aligned} [M]X'' + g\{[abs[K]\{X\}] \times [sgn\{X'\}]\} + \\ [K]\{X\} = \{F\} \end{aligned} \quad (67)$$

where

$abs[K]\{X\}$ = the absolute value of each component

g = damping coefficient

$sgn\{X'\}$ = the algebraic sign of the velocity $\{X'\}$

(b) The Reid's damping model has the following characteristics (ref. [67]):

(1) the energy loss per cycle in a harmonic motion is proportional to the coefficient g and square of the amplitude of displacement and independent of the frequency;

(2) for light damping ($g^2 \ll 1$) the Reid's damping model will be equivalent to the classical modal damping with a damping ratio of $\xi = g/\pi$ for each mode of vibration.

(c) Similar to the case of composite mass damping and stiffness damping, the damping coefficient g in eq. (a)(67) may also be considered as the elemental basis for a structural system which is composed of materials with different damping values.

N-1300 FLOW-INDUCED VIBRATION OF TUBES AND TUBE BANKS

N-1310 INTRODUCTION AND SCOPE

The flow-induced vibration (FIV) potential of structures has been known for a long time (refs. [79] through [84]). FIV analyses are required to determine the adequacy of a design, or in areas of uncertainty, to be aware of the need for experimental verification (refs. [85] and [86]) if high reliability of the component is a necessity. FIV may be due to any one of several excitation mechanisms because power systems include many types of flexible components subject to a variety of fluid flows, such as pipe, channel, and jet flows followed by mixing in plenums and heat exchangers. Since a single component is often subjected to different turbulent flows from several directions because of the influence of adjacent structures and boundaries, FIV analyses for more than one excitation mechanism is not unusual.

The quantitative data and correlations available to perform FIV analyses are unique to the flow geometry created by each component. More quantitative information and design methods are available for some components than others. In particular, the circular cylinder has been studied most. N-1320 through N-1340 of this Appendix are presented to illustrate one or more acceptable steps for the FIV analysis of arrays of cylinders subject to the three most significant excitation mechanisms. The general methods employed are applicable to other types of components, but the data are specifically for single cylinders and cylindrical arrays. Because of the large number of FIV mechanisms, the methodology of analysis is referenced, but enough information is given to understand a mechanism and make design calculations. Because of the developing nature of the subject, more than one set of design data or methods may be recommended with the implication to the designer to use either the more appropriate or the more conservative predictions.

Semiempirical correlations based on experimental data, but guided by the equations of motion, often form the basis of a design method. The state-of-the-art regarding description of the FIV mechanisms is that many mathematical models have been proposed for fluid-structure coupling forces, but general agreement on the physics of many of the phenomena has not been attained, although models simulating the behavior may be available.

N-1311 Definitions

In this section some commonly used terminologies in flow-induced vibration analysis are defined and briefly described.

(a) *Fluid forces* can be defined into two broad categories to describe FIV excitation mechanisms (refs. [79] through [83]). Fluid excitation forces are created by the incident flow on a structure, and they would occur, in some form, even without structural motion. Fluid-structure coupling forces are induced by structural motion, and they occur in both flowing and nonflowing fluids.

(b) *Added mass and added damping* have been successfully used to characterize the fluid-structure coupling forces created by the motion of a structure in a nonflowing fluid (refs. [87] through [93]). Added mass and added damping increase the effective mass and damping of a structure vibrating in a fluid. In addition, the presence of a dense fluid between otherwise unconnected, adjacent structures can couple their vibrations and result in significantly different natural frequencies, mode shapes, and damping from those obtained in a vacuum. For a low density fluid (e.g., air); the added mass is often negligible. Added mass is a function of the geometry of the structural surface exposed to fluid and the presence of adjacent structures, if any. Table N-1311-1 gives equations for added mass for two-dimensional sections and rigid bodies in two-dimensional motion. These equations were determined from exact solution of inviscid potential flow with a moving structural boundary. See N-1400 for dynamics of coupled fluid shells.

(c) In a *weakly coupled fluid-structure system*, the FIV excitation mechanism causes small structural motion and the fluid forces induced by the structural motion can be linearly superimposed onto the fluid excitation forces which are largely independent of the structural motion. The fluid-structure coupling forces can be expressed to a first order of approximation in terms of added mass, stiffness, and damping matrices. The fluid excitation forces can be determined separately from the coupling forces either by analysis or by model tests with only the hydraulics simulated.

Examples of FIV excitation mechanisms producing weakly coupled fluid-structure systems are incident flow turbulence and turbulent boundary layers over rods, plates, and shells (refs. [81] and [82]); some wake flows produced by flow across bluff bodies; and many sources of acoustic noise (refs. [80] and [95]). In these cases, the fluid excitation energy is generated at some point in the fluid circuit and the structure is the recipient of the energy. The forces due to flow turbulence and attached boundary layers typically are broadband random, while separated wake flows that roll into periodically shed vortices can produce very discrete frequency forces (refs. [82], [87], and [97]).

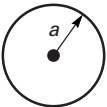
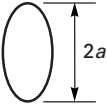
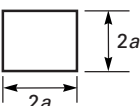
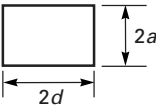

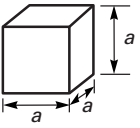
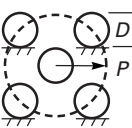
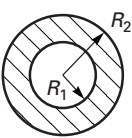
(d) In a *strongly coupled fluid-structure system*, the FIV excitation mechanism causes the structural motion to become large enough to change the flow field; some of the fluid forces amplify, rather than inhibit, the structural motion that produced them. Clearly distinguishing between fluid-structure coupling forces and fluid excitation forces is difficult in strongly coupled fluid-structure systems. In general, the coupling forces are highly nonlinear functions of structural motion and flow velocity.

(e) *Fluid-elastic instability* of closely packed heat exchanger tube bundles (refs. [80], [81], [82], and [88]) is an example of a strongly coupled fluid-structure system. The motion of each tube affects the fluid forces and the motion of the other tubes to produce self-excitation. The occurrence of the instability has been interpreted as due to adverse changes in the structural mass, damping, and fluid-structure coupling force (ref. [88]). However, most of the expressions for predicting the onset of instability are based on compilations of direct measurements of the critical velocities at the onset of instability.

(f) *Cross flow* is a flow perpendicular to the structural longitudinal axis. Cross flow is one example where an FIV mechanism is produced that can create either a weakly or a strongly coupled fluid-structure system. Vortex shedding in the wake of a tube in cross flow produces both fluid excitation forces and fluid-structure coupling forces that amplify structural motion. For ideal cross flow, where a long, smooth surface tube is isolated in uniform (2-D) cross flow with little or no turbulence in the approaching flow stream, very periodic, two-dimensional vortices are shed. These vortices produce alternating lift forces normal to the tube axis and flow and are nearly as large as the steady, flow direction drag forces, if the Reynolds number, based on the tube diameter, is below 2×10^5 (refs. [82], [87], and [89]). If the vortex shedding frequency is sufficiently different from the structural natural frequencies, the alternating lift forces act as fluid excitation forces only. However, if the vortex shedding frequency and one of the structural natural frequencies are sufficiently close to each other and the fluid excitation forces can produce large enough motion, then coupled fluid-structure forces occur, which apparently further amplify the motion. Enough experimental data are available to bound the fluid excitation forces, but the representation of the coupled fluid-structure forces is still being researched. Most of the representations are based on highly phenomenological models that stimulate, to various degrees, a small amount of data covering only a narrow range of idealized conditions.

(g) The *joint acceptance* is a measure of the probability that a structure vibrating in one mode will remain in the same mode when excited by a random force; the *cross acceptance* is a measure of the probability that a structure vibrating in one mode will change to another mode when excited by a random force. For many applications only the joint acceptance is assumed to be important. When mode shapes are normalized to unity, the sum of the joint

Table N-1311-1
Added Mass for Lateral Acceleration of Structures in a Fluid Reservoir

Geometry	Added Mass for Lateral Acceleration (Acceleration left to right) [Note (1)]	
1. Circular Section		$\rho \pi a^2 b$ [159]
2. Elliptical Section		$\rho \pi a^2 b$ [159]
3. Square Section		$1.51 \rho \pi a^2 b$ [159]
4. Rectangular Section		$k \rho \pi a^2 b$ [159]
		a/d
		k
		0.1 2.23
		0.2 1.98
5. Sphere		$\frac{2}{3} \rho \pi a^3$ [159]
6. Cube		$0.7 \rho a^3$ [159]
7. Cylinder Section in an Array of Fixed Cylinders		$\frac{\rho \pi D^2 b}{4} \left[\frac{(D_e/D)^2 + 1}{(D_e/D)^2 - 1} \right]$
	where	
	$D_e/D = (1 + 0.5 P/D)P/D$	
	Approximate solution from ref. [160]. See refs. [161] and [162] for arrays of flexible cylinders.	
8. Circular Section With a Fluid Filled Annulus		$\rho \pi R_1^2 b \left[\frac{1 + (R_1/R_2)^2}{1 - (R_1/R_2)^2} \right], \text{ inner cylinder}$ $\rho \pi R_2^2 b \left[\frac{1 + (R_1/R_2)^2}{1 - (R_1/R_2)^2} \right], \text{ outer cylinder}$ [163] See N-1451.1.
GENERAL NOTES:		
(a) b = length of section; ρ = fluid mass density. (refs. [159], [160], [161], [162], and [163].)		
(b) See N-1400 for finite length shells.		
NOTE:		
(1) The bracketed numbers refer to the references list following this Appendix.		

acceptances is equal to 1. (See ref. [112].) Therefore, the assumption that the joint acceptance is equal to 1 gives conservative estimates of structural responses.

(h) *Damping* is the result of energy dissipation during structural vibration. Damping limits resonant vibration amplitude and delays the onset of fluid elastic instability. Damping is the result of material damping within a structure, motion of trapped fluid within joints, and impact, scraping, and friction within joints. The material damping of nickel, copper, and steel is low, less than 0.1% critical damping. Damping at joints and supports dominates the damping of tubes that are supported by passing through oversized holes in support baffles. Table N-1311-2 gives general guidelines for damping in flow-induced vibration. Testing is required to establish more precise estimates of damping for specific designs.

N-1312 Nomenclature

C_n = reduced damping in n th mode
 C_L = lift coefficient
 D = cylinder diameter
 E = Young's modulus
 f_n = natural frequency of n th vibration mode, hertz
 f_s = frequency of periodic vortex shedding, hertz
 F = force
 G_f = single-sided power spectral density of the forcing function, in (force/length)² per Hz
 G_f^i = G_f spectrum for the i th span of a multi-span tube
 G_y = single-sided power spectral density of response
 H_j = transfer function of j th vibration mode
 I = area moment of inertia
 J^2 = joint acceptance
 J_{jk}^2 = cross acceptance for the j th and k th vibration modes

$(J_{jk}^2)^2$ = acceptance for the i th span
 ℓ_c = axial correlation length
 $= 2 \int_0^L r(x') dx'$ where $r(x')$ is the correlation function and x' is the separation distance
 ℓ_c^i = correlation length in the i th span
 L_e = cylinder length subject to vortex shedding
 L_t = span length
 m = mass per unit length
 m_A = added fluid mass per unit length
 m_c = contained fluid mass per unit length
 m_f = cylinder displaced fluid mass/length
 m_s = structural mass per unit length
 m_t = total mass per unit length of tube
 $= m_A + m_c + m_s$
 M_j = modal mass
 M_n = effective modal mass/length for n th vibration mode
 n = vibration mode, $n = 1$ is fundamental mode
 p = pressure
 P = tube pitch
 $=$ distance between tube centers
 q = dynamic pressure, $(1/2)\rho V^2$
 Re = Reynolds number, VD/ν
 R_p = Cross correlation of the pressure field
 S = Strouhal number, $f_s D/V$
 S_f = cross spectral density of the forcing function on a cylinder, (force/length)² per Hz
 S_{fo} = power spectral density of the forcing function
 S_p = cross spectral density of the pressure field
 S_y = power spectral density of cylinder response
 t = time
 U_c = convection velocity

Table N-1311-2
Guidelines for Damping of Flow-Induced Vibration

Description of Tube Installation	Fluid Surrounding Tube	Critical Damping Ratio, ζ		
		Low [Note (1)]	Typical Design Value	High [Note (2)]
Thermowells and single span tubes supported by welded or rolled in ends	Liquid and gas	0.0005	0.002	0.005
Multispan heat exchanger tubes supported by passing through oversized holes in plates	Low density gas	0.008	0.017	0.03 [Note (3)]
	Water and other liquids	0.01	0.02	0.03 [Note (3)]

GENERAL NOTE: This table applies to metallic tubes with 0.5 in. (13 mm) to 2.0 in. (50 mm) outside diameter. For tubes passing through oversized holes in support plates, this table applies to typical diametrical clearance between tube outside diameter and tube support inside diameter of 0.010 in. (0.2 mm) to 0.030 in. (0.8 mm).

NOTES:

- (1) Low value: For midspan rms vibration amplitude less than 1% of tube diameter and smaller than the diametrical clearance between the tube and the support plate.
- (2) High value: For midspan rms amplitudes comparable to or larger than the diametrical clearance between the tube and the support plate. Tube wear can result.
- (3) Critical damping ratios $0.03 < \zeta < 0.05$ can be used if justified by applicable experimental data.

V = mean velocity
 x = axial distance
 y_n^* = maximum displacement in n th vibration mode
 \bar{y}^2 = mean square response of a cylinder
 α_n = amplification factor in n th vibration mode
 γ_n = mode shape factor in n th vibration mode
 Γ = coherence of forcing function on a cylinder
 Γ^i = coherence for i th span
 δ_m = mass-damping parameter, $2\pi\xi_n m_t/\rho D^2$
 δ_n = log decrement for n th vibration mode
 $= 2\pi\xi_n$
 ξ_n = fraction of critical damping for n th mode
 ρ = fluid mass density
 ϕ_n = n th vibration mode shape
 ϕ_n^* = maximum value of ϕ_n
 θ = angle between direction of flow and normal to tube axis
 ν = kinematic viscosity
 ω = frequency, radian/sec

N-1320 VORTEX SHEDDING

N-1321 Vortex Shedding From a Fixed Bluff Body

For a bluff body in uniform cross flow, the wake behind the body is no longer regular, but contains distinct vortices of the pattern shown in Figure N-1321-1 for a circular cylinder. The vortices are shed alternatively from each side of the body in a regular manner and give rise to an alternating lift force. Experimental studies of this vortex shedding process have shown (refs. [94] and [95]) that the frequency in hertz of the alternating lift force can be expressed as:

$$f_s = SV/D \quad (68)$$

Some common types of bodies or structures for which vortex shedding occurs are shown in Figure N-1321-2. The following discussions are based on the circular cylinder; however, the concepts apply equally well to other bluff bodies.

The oscillating lift force produced on an isolated single cylinder of diameter D and length L by uniform cross flow can be expressed as (refs. [96] and [97]):

$$F = C_L J q D L [\sin(2\pi f_s t)] \quad (69)$$

where C_L , f_s , and J are functions of the Reynolds number

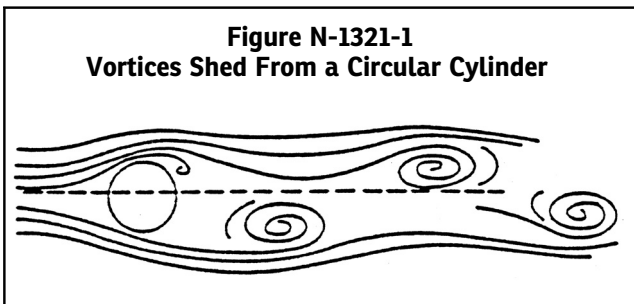
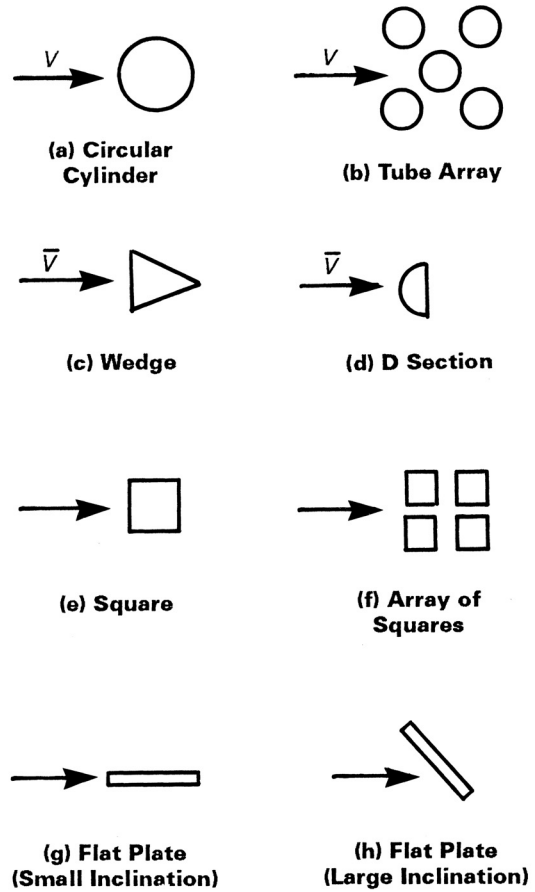


Figure N-1321-2
Some Typical Cross Sections of Bluff Bodies That Can Experience Vortex Shedding



Re and must be determined experimentally. In uniform cross flow, the energy of vortex shedding occurs over a very narrow frequency band with a center frequency f_s , except over a transition band of Reynolds number (2×10^5 to 3×10^6) where the character of the frequency content may vary from almost periodic to completely random. The measured Strouhal number is $S \approx 0.2$ for $10^3 < Re < 2 \times 10^5$; for larger Re , experimental values of S and C_L show considerable scatter.

The alternating vortex fluid forces are not generally correlated over the entire cylinder length L . As a consequence, two limiting cases of the joint acceptance exist for a uniform rigid-body-mode (ref. [97]).

$$\begin{aligned}
 J^2 &= \ell_c / L \quad \text{if } \ell_c \ll L \\
 &= 1 \quad \text{if fully correlated}
 \end{aligned} \quad (70)$$

The correlation length in the lift direction for stationary cylinders has been found to be approximately 3 to 7 diameters ($3D < \ell_c < 7D$), for $10^3 < Re < 2 \times 10^5$ (ref. [87]). For larger Reynolds numbers, the correlation lengths for stationary cylinders can be expected to be even smaller because the attached boundary layer becomes fully turbulent. J^2 is usually much less than 1 for long stationary cylinders. Motion of the cylinder at the frequency of vortex shedding substantially increases the correlation length (refs. [82] and [87]) as discussed in N-1323 and N-1324.

Vortex shedding also induces a force in the streamwise or drag direction. The drag force occurs at twice the vortex shedding frequency for single cylinders (ref. [87]). However, the magnitude of the oscillating drag force is typically an order of magnitude smaller than the oscillating lift force.

N-1322 Practical Cross Flow

The case of ideal cross flow is rarely found except in the laboratory. Many practical conditions reduce the effectiveness and strength of vortex shedding as an excitation mechanism:

(a) If the body is located in a turbulent flow, or if the tube surface is rough, the turbulence tends to widen the band of shedding frequencies and decrease the energy at the dominant shedding frequency (ref. [129]).

(b) If the cylinder is inclined to the flow, the shedding frequency can be adequately predicted by employing the component of flow velocity normal to the cylinder axis

$$f_s = (SV/D) \cos \theta \quad (71)$$

where θ is the angle between the direction of flow and the normal to the cylinder axis. Inclined flow tends to reduce the magnitude of the vortex shedding forces (ref. [98]).

(c) Spanwise variations in flow velocity imply that the vortex shedding frequency also varies in the spanwise direction. This effect will generally reduce the magnitude of the net vortex shedding excitation.

(d) There is some evidence (refs. [99] and [100]) that vortex shedding does not occur in two-phase flow and that vortex shedding is only a concern in single-phase flows.

(e) While the vortex shedding characteristics discussed above have general applicability, the effects of adjacent bodies have not been specifically included. Studies of two (ref. [101]) or more circular cylinders show that vortex shedding does occur, but its character is very sensitive to the relative location and spacing of the cylinders. For the important case of tube arrays, the values of S , J , and C_L to be employed in eq. N-1321(69) are much more uncertain than for single cylinders, as is evident by the considerable scatter in the experimental data (refs. [100], [102], [103], [104], [105], [139], and [140]).

N-1323 Flexible Cylinders

When the vortex shedding frequency f_s is sufficiently different from the structural natural frequencies, a condition called off-resonance, the representation of the vortex shedding lift force by F given in eq. N-1321(69) is valid, and is conservative if $C_L = 1$ and $J = 1$ is chosen. This conservative representation of the force can be extended to nonuniform loading, and modal response can be employed to simplify the analysis of cylinders where many modes are active. Normally, off-resonance response is small. However, as resonance is approached, large motions are encountered.

In the case of a single flexible or resiliently supported tube, once vibration begins the shedding frequency and the tube natural frequency can become synchronized if the two are sufficiently close. For a spring-supported cylinder in an air stream, it was shown (ref. [106]) that the velocity range over which synchronization persists depends upon the damping parameter, $m_t \delta_n / \rho D^2$. In Figure N-1323-1, the shaded area is the region of synchronization. The ordinate, $V/f_n D$, is a reduced velocity, where f_n is the natural frequency of the spring-mounted cylinder. Note, in particular, that with increasing $m_t \delta_n / \rho D^2$ the reduced velocity range over which synchronization persists decreases, and no synchronization occurs for $m_t \delta_n / \rho D^2 > 32$. Outside the shaded area, the cylinder experiences an alternating lift force at the vortex shedding frequency for a stationary cylinder, as given previously in eq. N-1321(69).

The consequences of synchronization are many. As the flow velocity is either increased or decreased so that the vortex shedding frequency approaches the structure frequency, the following will occur.

(a) The vortex shedding frequency shifts to the structural natural frequency, i.e., it synchronizes with or "locks-in" to the structural frequency even if the flow velocity or the structural frequency is varied within the range of synchronization as indicated in Figure N-1323-1.

(b) The spanwise correlation of the vortex shedding forcing function increases rapidly as structural response increases.

(c) The lift force becomes a function of structural amplitude.

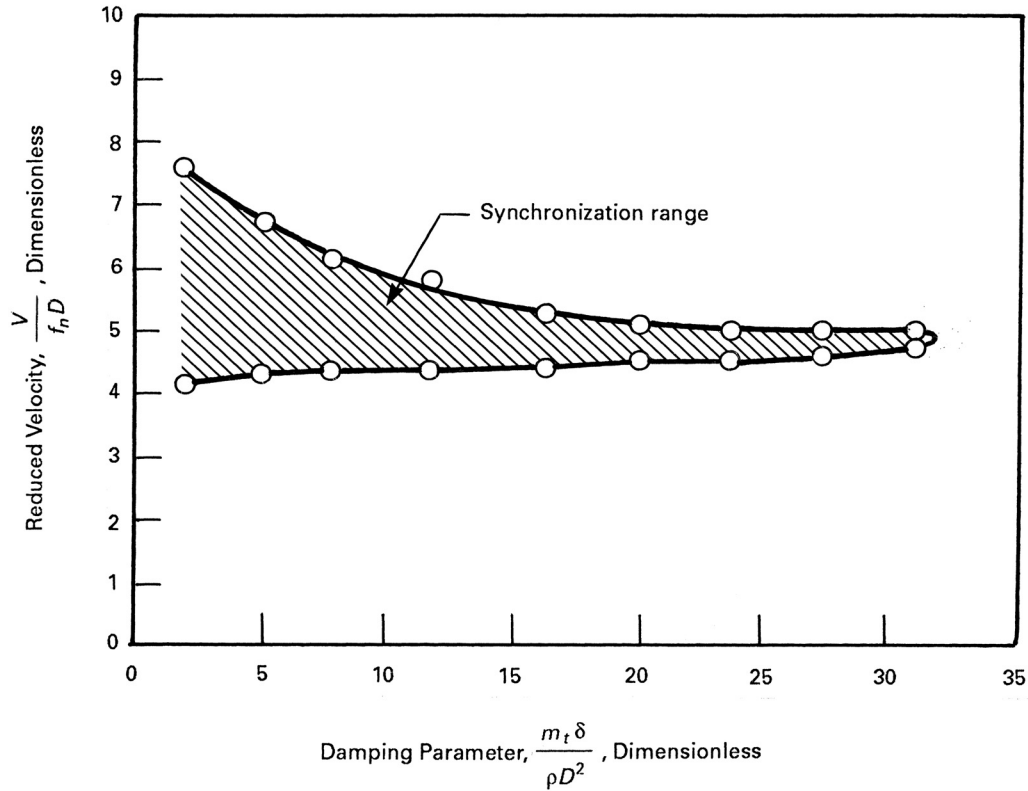
(d) The drag force on the structure increases.

(e) The strength of the shed vortices increases.

Within the synchronization band, substantial resonant vibration in lightly damped structures often occurs. Vibration amplitudes up to three diameters peak-to-peak have been observed in dense fluids, such as water, over cables and tubing. The vibrations are predominantly transverse to the flow and they are self-limiting (refs. [82], [87], and [94]).

Large-amplitude, synchronized vibrations in the drag direction have been observed for a single cylinder in water. These oscillations initiate at relatively low flow velocities corresponding to subharmonic frequencies of

Figure N-1323-1
Synchronization of the Vortex Shedding Frequency and the Tube Natural Frequency for a Single, Flexibly-Mounted Circular Cylinder



GENERAL NOTE: Synchronization occurs within the shaded region. (Ref. [106])

vortex shedding, i.e., at $1/4$, $1/3$, or $1/2$ the flow velocity required for synchronization according to eq. N-1321(68) (refs. [107] and [108]). However, the synchronization in the drag direction is not as strong as in the lift direction, and usually occurs only for lightly damped structures in dense fluids (refs. [87] and [110]). Lock-in has not been observed in two-phase flow or deep (more than a few rows) inside a closely spaced tube bundle.

N-1324 Design Procedures for a Circular Cylinder

Whenever possible, lock-in operating conditions should be avoided, but complex designs often make this impossible. Thus, criteria are given for which lock-in can be avoided, and off-resonance structural dynamic analysis can be employed, as well as design procedures to calculate the response during lock-in.

N-1324.1 Avoiding Lock-In Synchronization. Lock-in for a single cylinder can be avoided by one of the following four methods (refs. [82], [106], [108], and [109]). For tube arrays, only (a), (b), and (c) are applicable methods, and V must be the flow velocity in the minimum gap ($P - D$).

(a) If the reduced velocity for the fundamental vibration mode ($n = 1$) satisfies:

$$V/f_1 D < 1 \quad (72)$$

then both lift and drag direction lock-in are avoided.

(b) If for a given vibration mode the reduced damping is large enough

$$C_n > 64 \quad (73)$$

then lock-in will be suppressed in that vibration mode.

(c) If for a given vibration mode

$$V/f_n D < 3.3 \quad (74)$$

and

$$C_n < 1.2 \quad (75)$$

then lift direction lock-in is avoided and drag direction lock-in is suppressed.

(d) If the structural natural frequency falls in the ranges $f_n < 0.7f_s$ or $f_n > 1.3f_s$, then lock-in in the lift direction is avoided in the n th mode.

The reduced damping C_n is calculated according to

$$C_n = \frac{4\pi\xi_n M_n}{\rho D^2 \int_{L_e} \phi_n^2(x) dx} \quad (76)$$

where $\xi_n = \delta_n/2\pi$ is the fraction of critical damping measured in air and M_n is the generalized mass

$$M_n = \int_0^L m_t(x) \phi_n^2(x) dx \quad (77)$$

with ϕ_n the n th mode shape function and $m_t(x)$ is the cylinder mass per unit length. The range L_e in the denominator implies that the integration is over only the region of the cylinder length subject to lock-in cross flow. Note that m_t is calculated according to:

$$m_t(x) = m_s(x) + m_c(x) + m_A(x) \quad (78)$$

For an isolated cylinder, m_A is the displaced fluid mass. If sections of the cylinder are close to other bodies, then the possibility of increased added mass and fluid damping must be taken into account (refs. [81], [82], [90] to [93]).

N-1324.2 Vortex-Induced Response. Off resonance, the response can be calculated using standard methods (ref. [96]) of forced-vibration analysis and eq. N-1321(69) for the forcing function (refs. [82] and [89]). The resultant response is ordinarily small. If operating conditions are such that lock-in cannot be avoided or suppressed, then the resonant vortex induced response must be calculated. Three approaches for calculating the response are recommended for three classes of structures and flows: single uniform cylinder in uniform flow, tube arrays, and nonuniform cylinders in nonuniform flow.

(a) *Uniform Structure and Flow.* If a uniform cylinder is subject to uniform cross flow over its span, then both the vortex shedding frequency and the vortex force are constant over the span of the cylinder. The periodic vortex induced lift force is given by eq. N-1321(69). At lock-in, the vortex shedding frequency equals the natural frequency of the n th vibration mode, $f_s = f_n$, and the cylinder response is given by (refs. [82] and [89])

$$\frac{y_n^*}{D} = \frac{C_L J \phi_n^*}{16\pi^2 S^2 [m_t \xi_n / (\rho D^2)]} \quad (79)$$

This equation provides a conservative upper bound estimate to the amplitude of periodic vortex-induced vibration if the lift coefficient is taken as unity, $C_L = 1$, and the vortex shedding is fully correlated along the span of the cylinder, $J = 1$. Other values of C_L and J may be used in circumstances where experimental data are available. However, eq. (79) with $C_L = 1$ and $J = 1$ has been found to give overly conservative predictions owing to the

tendency of the actual lift coefficient to decrease at vibration amplitudes exceeding 0.5 diameter and the lack of perfect spanwise correlation at lower amplitudes. To obtain less conservative predictions, three semiempirical nonlinear methods are given in Table N-1324.2(a)-1. The mode shape factor γ generally varies between 1.0 and 1.3 (ref. [82]) and C_n is determined according to eq. N-1324.1(d)(76) using ξ_n determined in air.

(b) *Within Tube Arrays.* Coherent vortex shedding has been found to exist only in the first few rows in arrays of cylinders with center-to-center spacing less than 2 diameters, and the design procedures for a single cylinder are applicable using the velocity in the minimum gap ($P - D$). Within the array vortex shedding exists over a broad range of frequencies rather than at a single distinct frequency. The response within the array is generally less than that of a comparable single cylinder. The techniques that have been developed to predict vibration within the array are based on the theory of random vibration and are given in N-1340.

(c) *Nonuniform Structures and Flow.* Many cylindrical structures have nonuniform distribution of mass and stiffness, and they are exposed to flow velocities that vary over the span. In this case, only a part of the span of structure will resonate with vortex shedding and contribute to the excitation.

One method for treating nonuniform structures in non-uniform flows is

(1) determine the natural frequencies and mode shapes of the structure.

(2) determine the spanwise distribution of the flow.

(3) identify portions of the structure that can resonate with vortex shedding for each mode. This can be done by calculating the spanwise distribution of the vortex shedding frequency and estimating the potential for resonance by a band of plus or minus 30% from this frequency.

Table N-1324.2(a)-1
Semiempirical Correlations for Predicting
Resonant Vortex-Induced Vibration
Amplitude

Reference	Predicted Resonant Amplitude
111	$\frac{y_n^*}{D} = \frac{1.29\gamma}{[1 + 0.43(2\pi S^2 C_n)]^{3.35}}$
82	$\frac{y_n^*}{D} = \frac{0.07\gamma}{(C_n + 1.9)S^2} \left[0.3 + \frac{0.72}{(C_n + 1.9)S} \right]^{\frac{1}{2}}$
110	$\frac{y_n^*}{D} = \frac{0.32\gamma}{[0.06 + (2\pi S^2 C_n)^2]^{\frac{1}{2}}}$

(4) Apply a lift force given by eq. N-1321(69) with $f_n = f_s$ and $C_L = 1$ to those segments of the span that are resonant.

Procedures (1) through (4) are illustrated in refs. [89] and [112]. For a uniform cylinder in uniform cross flow, assumption (4) of complete correlation and $C_L = 1$ gives overly conservative predictions. Other values for C_L may be used where experimental data are available.

N-1330 FLUID-ELASTIC INSTABILITY

Many FIV mechanisms exist wherein as energy supplied to the system is increased, usually as increased flow velocity, a critical value is attained at which a large increase in response occurs. Continued increases in the supplied energy results in continued static or dynamic divergences (rapid increases) of the response. In general, fluid-elastic instability is a result of strong coupling between the structure and the fluid.

N-1331 Instability of Tube Arrays in Cross Flow

Fluid flow across an array of elastic tubes can induce a dynamic instability that can result in very large amplitude vibrations once a critical cross flow velocity is exceeded. Often, motion is limited only by tube-to-tube impacting. The flow of fluid over the tubes results in both fluid excitation and fluid-structure coupling forces on the tubes. The fluid-structure coupling excitation forces fall into several groups

(a) forces that vary approximately linearly with displacement of a tube from its equilibrium position (displacement mechanisms) (ref. [113]).

(b) fluctuations in the net drag forces induced by the oscillating tube's relative velocity with respect to the mean flow (fluid damping mechanism) (ref. [88]).

(c) combinations of the above forces that exhibit step changes as a certain amplitude is exceeded because of the abrupt shift in the point of flow separation (jet switch mechanism) (Ref. [114]). Instability may result from any or all of these fluid forces which are functions of the tube motion.

The general characteristics of tube vibration during instability are as follows.

(d) *Tube Vibration Amplitude.* Once a critical cross flow velocity is exceeded, vibration amplitude increases very rapidly with flow velocity V , usually as V^n where $n = 4$ or more, compared with an exponent in the range $1.5 < n < 2.5$ below the instability threshold. This can be seen in Figure N-1331-1, which shows the response of an array of metallic tubes to water flow. The initial hump is attributable to vortex shedding that tends to produce larger amplitudes in water flow than air flows.

(e) *Vibration Behavior With Time.* Often the large amplitude vibrations are not steady in time, but rather beat with amplitudes rising and falling about a mean value in a pseudorandom fashion (ref. [115]).

(f) *Synchronization Between Tubes.* Most often the tubes do not move as individuals, but rather move with neighboring tubes in somewhat synchronized orbits, as shown in Figure N-1331-2. This behavior has been observed in tests both in water and air (refs. [113], [115], [116], and [117]), with orbit shapes ranging from near circles to near straight lines. As the tubes whirl in their oval orbits they extract energy from the fluid. The stiffness mechanism requires motion of the adjacent tubes, but the damping mechanism does not.

(g) *Influence of Structural Variations.* Restricting the motion or introducing frequency differences between one or more tubes often increases the critical velocity for instability (refs. [115], [116], and [118]). Such increases are generally no greater than about 40%. Often the onset of instability is more gradual in a tube bank with tube-to-tube frequency differences than in a bank with identical tubes which are free to vibrate.

N-1331.1 Prediction of the Critical Velocity. Dimensional analysis considerations imply that the onset of instability is governed by the following dimensionless groups: the mass ratio $m_t/\rho D^2$; the reduced velocity $V/f_n D$; the damping ratio ξ_n , measured in the fluid; the pitch to diameter ratio P/D ; the array geometry (see Figure N-1331-3), and the Reynolds number VD/ν . In this section, V is the flow velocity in the gaps between the tubes, and is determined by the product of $P/(P - D)$ and the (approach) flow velocity that would occur if the tubes were not present. Note the added mass part of m_t may be much larger than the displaced fluid mass because of the confining effect of adjacent tubes (refs. [81], [90], and [92]). Also, for most cases, the flow is fully turbulent ($VD/\nu > 2000$) and the Reynolds number is not expected to play a major role in the instability. In such cases, the reduced critical velocity for the onset of instability can be expressed as a function of the remaining nondimensional parameters.

The relationship between the parameters can be investigated theoretically or experimentally. One general form that has been used to fit experimental data is

$$V_c/f_n D = C \left(m_t/\rho D^2 \right)^a \left(2\pi\xi_n \right)^b \quad (80)$$

where C and the indices a and b are functions of the tube array geometry. Experimental data suggest that a and b fall in the range $0.0 < a, b < 1.0$ (refs. [115], [116], [119], [138], and [139]).

N-1331.2 Recommended Formula. Mean values for the onset of instability can be established by fitting semiempirical correlations to experimental data. The correlation form chosen is

$$V_c/f_n D = C \left[m_t \left(2\pi\xi_n \right) / \rho D^2 \right]^a \quad (81)$$

Figure N-1331-1
Response of a Tube Bank to Cross Flow (Ref. [115])

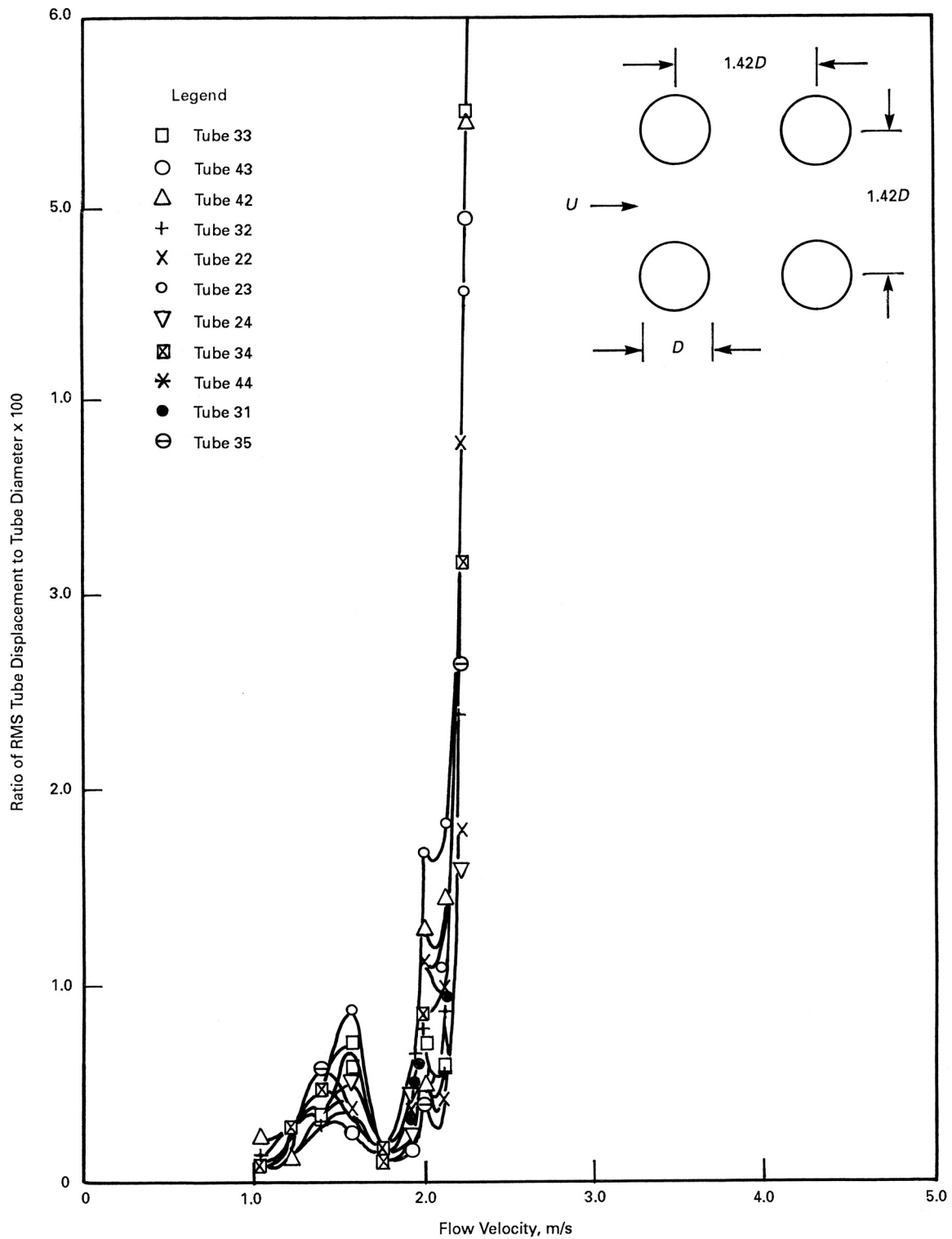
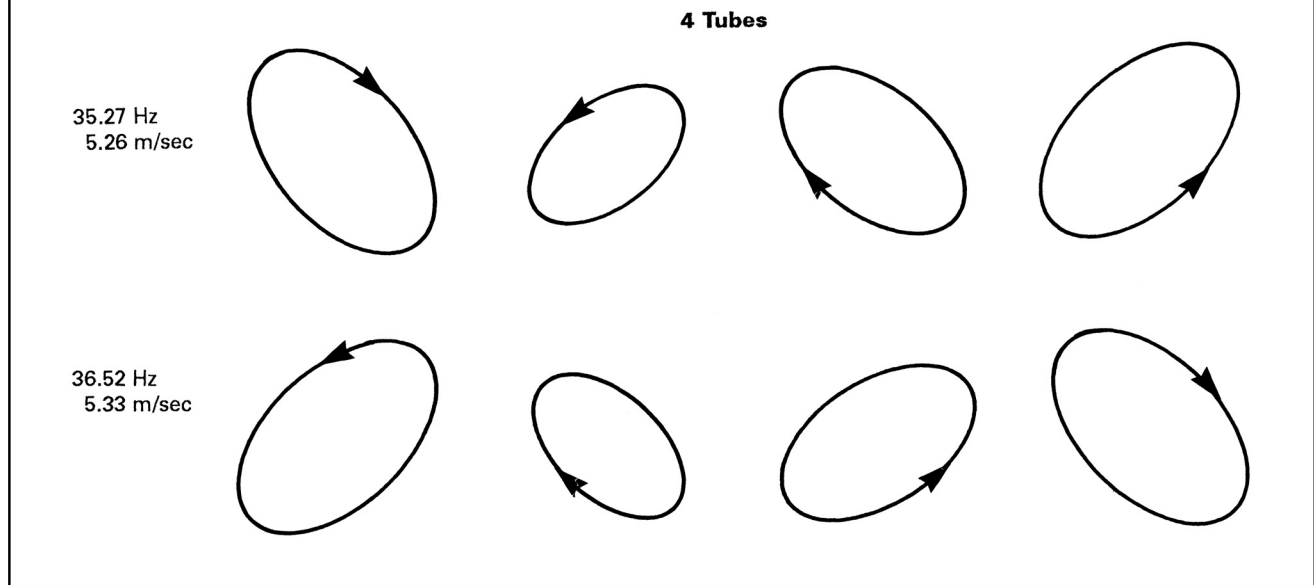


Figure N-1331-2
Tube Vibration Patterns at Fluidelastic Instability for a Four-Tube Row (Ref. [118])



where

V_c = critical cross flow velocity

f_n = natural frequencies of the immersed tube

For uniform cross flow, the tubes will be stable if the representative cross flow velocity V is less than the critical velocity V_c . If the flow is nonuniform over the tube lengths, an equivalent uniform cross flow gap velocity can be defined as either the maximum cross flow velocity, or the modal weighted velocity:

$$V_e^2 = \int_0^L V^2(x) \phi_n^2(x) dx / \int_0^L \phi_n^2(x) dx \quad (82)$$

where $V(x)$ is the cross flow velocity at each axial location of the tube. The tubes will be stable if $V_e < V_c$ for all modes.

The available 170 data points for onset of instability (ref. [120]) are shown in Figure N-1331-4. In the range $m(2\pi\xi_n)/\rho D^2 > 0.7$, there are sufficient data to permit fitting of eq. (81) to data for each array type. The mean values of C are

	Triangle	Rotated Triangle	Rotated Square	Square	All
C_{mean}	4.5	4.0	5.8	3.4	4.0

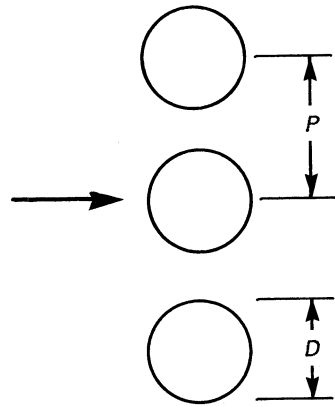
Based on theory for the displacement mechanism (Ref. [113]), which is active in this parameter range, $a = 0.5$ was chosen in these fits. For $m(2\pi\xi_n)/\rho D^2 < 0.7$, where the fluid damping mechanism is primarily active, neither the theory nor data are sufficient to establish values of C and a in eq. (81). Conservative estimates of the mean values of $V_c/f_n D$ for $m_t(2\pi\xi_n)/\rho D^2 < 0.7$ can be obtained

using eq. (81) with $a = 0.5$ and the mean C given in the table above. The use of eq. (81) with $a = 0.5$ and $C = 3.3$ has been recommended (refs. [80], [100]) for the entire mass damping parameter range of Figure N-1331-4.

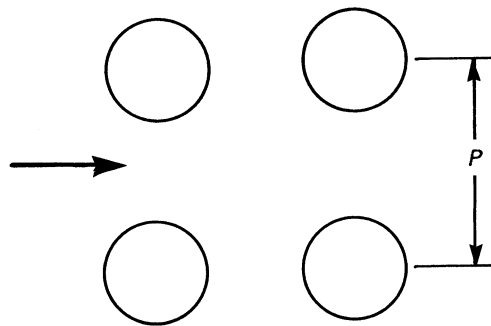
N-1331.3 Suggested Inputs. Accurately predicting the critical velocity requires scale model testing to determine the value of C and the damping ratio in each application, because practical flow and structural geometries contain features nonexistent in the simpler, controlled laboratory tests used to establish the data base of Figure N-1331-4 (ref. [120]). Usually, industrial tube arrays (bundles) involve multiple spans with intermediate supports provided by plates with holes slightly larger than the tube diameter. Also, flow may pass around the edge of the bundle and does not have the pure cross flow direction shown in Figure N-1331-3, even within the bundle. Furthermore, when the vibration amplitude is small, such as that experienced during subcritical vibration, not all support plates are active. Damping ratios in this vibration mode are typically small, from 0.1% in gas to about 1% in steam or water. When the vibration amplitude is large, as characterized by the onset of instability, support plate-to-tube interaction greatly increases the damping ratio which can reach 5% or more.

All the practical features discussed above tend to raise the critical flow velocity. Thus, the data base of Figure N-1331-4 can be used to determine a conservative criterion for avoiding fluid-elastic instabilities of tube arrays: if the design equivalent uniform cross flow gap velocity [eq. N-1331.2(82)] is less than the critical velocity [eq. N-1331.2(81)] computed with the suggested design values defined by the solid line ($C = 2.4$, $a = 0.5$) in

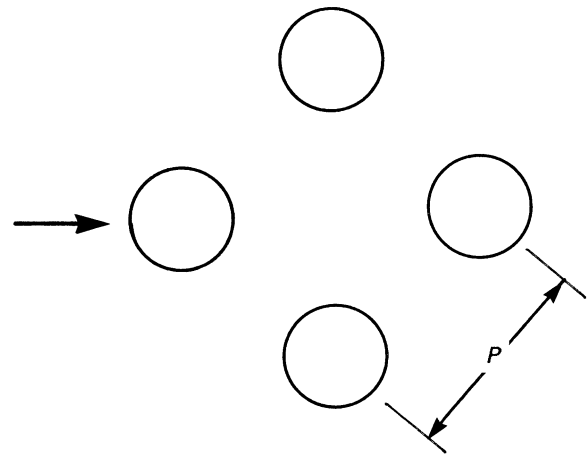
**Figure N-1331-3
Tube Arrangements**



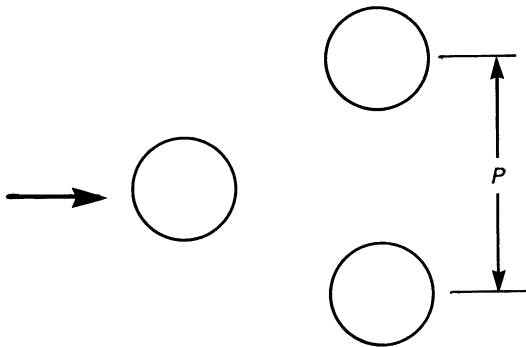
Tube Row



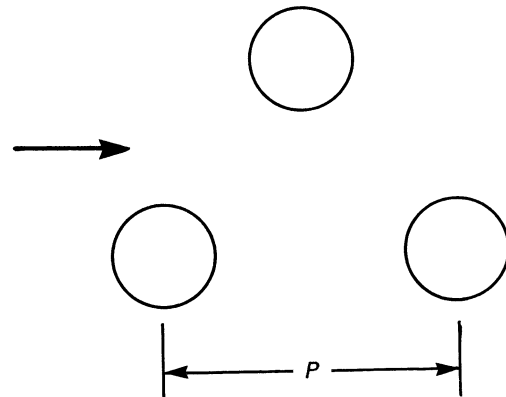
**Square Array
(90 Deg)**



**Rotated Square Array
(45 Deg)**



**Triangular Array
(30 Deg)**



**Rotated Triangular Array
(60 Deg)**

Figure N-1331-4
Stability Diagram

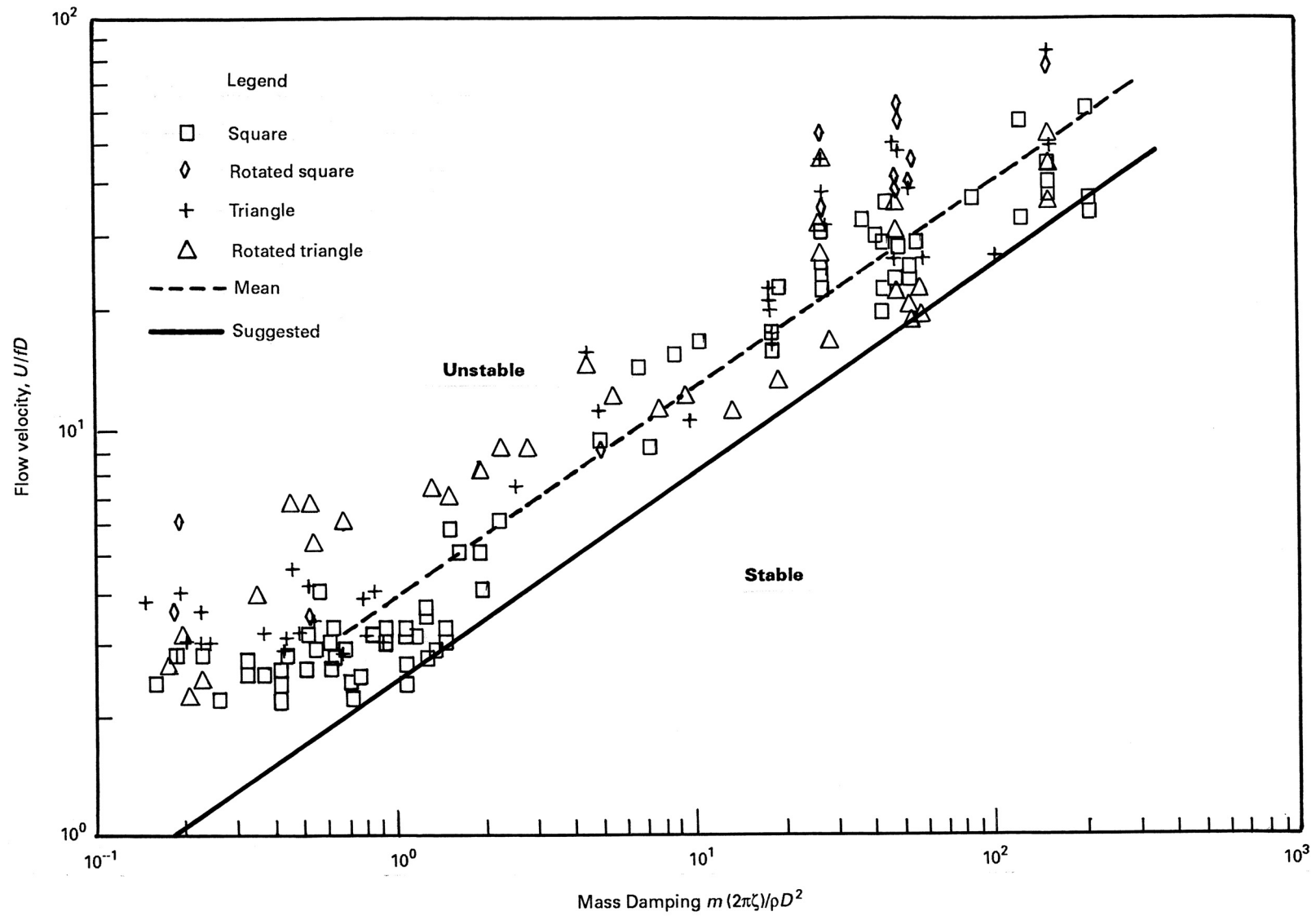


Figure N-1331-4 and with a damping ratio of 0.5% in gas, or 1.5% in “wet” steam or liquid, then instability is almost certainly not a problem, and scale model testing will not be necessary. Otherwise, more accurate values of C and the immersed tube’s damping ratios, or the critical velocity itself, must be determined by either model testing or from operational experience.

N-1340 TURBULENCE

In general, the coolant flow paths and flow rates promote and maintain turbulent flows that are optimal for purposes of heat transfer, but provide sources for structural excitation. Also, turbulence in the flow can affect the existence and strength of other excitation mechanisms associated with separating boundary layer flows (wakes), as discussed in N-1320 on vortex shedding. This section will concentrate on turbulence as a source of fluid excitation forces.

N-1341 Random Excitation

Where turbulent flow comes into contact with the surface of a structure some of the momentum in the flow is converted into fluctuating pressures. In addition to any forces produced by the mean flow component, random surface pressure fluctuations are produced by the turbulent velocity component. The time history of the surface pressure fluctuations, like the flow turbulence, is complex and amenable to description only on a statistical basis. However, the fluctuating pressure and the resulting flow-induced response usually can be regarded as ergodic and analyzed with a finite-time record not dependent upon the time origin.

For purposes of structural analysis and design, most useful information on the fluctuating pressures becomes available once the spatial spectral densities of the pressure field, $S_p(x_1, x_2, \Omega)$, are determined. The spectral density is the Fourier transform of the cross correlation of the pressure field

$$S_p(x_1, x_2, \Omega) = \frac{1}{2\pi} \int_{-\infty}^{\infty} \left\{ \lim_{t_0 \rightarrow \infty} \frac{1}{2t_0} \int_{t_0}^t p(x_1, t) \times p(x_2, t + \tau) d\tau \right\} e^{-i\Omega t} dt \quad (83)$$

and provides information about the average products of components of the pressure $p(x, t)$ as a function of the circular frequency Ω in radian/sec for every possible pairing of structural points, x_1 and x_2 , including the same point. S_p has units of (pressure)² (second). The frequency content of the spectra are band limited, from zero to a maximum frequency determined by the turbulence source. The magnitude of the power spectra increases when the energy of the turbulence at frequency Ω increases. The

size of a region of the structure over which the pressures at different points are coherent, or has some cause-effect relationship, is interpreted as a correlation length of the pressure field, or the size of the associated turbulent eddy (ref. [82]).

Only select parts of the surface pressures are effective in exciting dynamic structural response: those parts with frequency content in narrow bands centered on the structural natural frequencies and with correlation lengths similar in size to the spatial wavelength of the associated vibration mode (ref. [122]). The resulting structural response occurs in the narrow frequency bands with random amplitudes, and the widths of the frequency bands are determined by the system damping. Knowledge of the surface pressure statistics enables prediction of the associated structural response statistics utilizing the probabilistic theory of structural dynamics.

N-1342 Structural Response of Tubes and Beams

N-1342.1 Response to Homogeneous Turbulence Excitation. The assumption of a linear structure is justifiable for the small vibrations associated with the turbulence excitation of weakly coupled fluid-structure systems, and the linear structural dynamic analysis theory for arbitrary random loading of beams is highly developed. Since the energy dissipation mechanism of turbulent flow rapidly smooths disturbances caused by the structural boundaries of and in the flow channel, the statistical character of turbulent cross flow often varies gradually over the total length of select spans of single tubes and tube bundles, especially within the bundles. Thus, in many applications, the assumption of a uniform mean velocity and homogeneous turbulence is reasonable.

Assuming a homogeneous and ergodic pressure field, the equations of motion can be uncoupled to allow solution by modal analysis (see N-1222). The expression for the power spectral density of the cylinder response is (ref. [123])

$$S_y(x_1, x_2, \omega) = S_{f_0} L \sum_{j=1}^{\infty} \sum_{k=1}^{\infty} \phi_j \phi_k J_{jk}^2 H_j H_k^* \quad (84)$$

The mode shapes $\phi_j(x)$ satisfy the orthogonality relation

$$\int_0^L m_t(x) \phi_i(x) \phi_j(x) dx = M_j \delta_{ij} \quad (85)$$

where M_j is the generalized mass, defined here to have the same dimensions as m_t . Thus, if m_t is constant, $M_j = m_t$ and the orthogonality condition reduces to

$$\int_0^L \phi_i(x) \phi_j(x) dx = \delta_{ij} \quad (86)$$

The transfer function for the j th mode is

$$H_j(\omega) = [M_j(\omega_j^2 - \omega^2 + 2i\xi_j\omega\omega_j)]^{-1} \quad (87)$$

where ω_j and ξ_j are the modal natural frequency and damping, respectively. The acceptance integral is

$$J_{jk}^2(\omega) = L^{-1} \int_0^L \int_0^L \phi_j(x_1) \Gamma(x_1, x_2, \omega) \phi_k(x_2) dx_1 dx_2 \quad (88)$$

where the complex coherence function is

$$\Gamma(x_1, x_2, \omega) = S_f(x_1, x_2, \omega) / S_{fo}(\omega) \quad (89)$$

and S_f is the cross spectral density of the turbulent forcing function per unit length between two different points on the cylinder's length, $x = x_1$ and $x = x_2$. When $x_1 = x_2 = x$, $S_f(x, x, \omega) = S_{fo}(\omega)$ is the power spectral density (or auto-spectrum) that is independent of location for a homogeneous pressure field. The joint acceptance $J_{jj}(\omega)$ reflects the relative effectiveness of the forcing function to excite the j th vibration mode while the cross acceptance $J_{jk}(\omega)$, $j \neq k$, reflects contributions due to coupling between different modes. In general, the responses in two different modes are dependent upon each other.

The mean square response $\bar{y}^2(x)$ is the most useful measure for the amplitude or stress and strain design, and is found by integration of the power spectral density of the response, $S_y(x, \omega) = S_y(x, x, \omega)$ over the frequency band, or

$$\bar{y}^2(x) = \int_{-\infty}^{\infty} S_y(x, \omega) d\omega \quad (90)$$

The distribution of the positive and negative peaks in displacement has been found for the fundamental mode of a rod in parallel flow to be Gaussian (refs. [81], [82], and [124]). Assuming a Gaussian fluctuating pressure distribution, a Rayleigh distribution is expected for the absolute amplitude of response (refs. [82] and [125]).

Based on physical reasoning and experimental data, the complex coherence function of the homogeneous turbulence pressure field for the tubes in cross flow has been characterized as (refs. [122] and [126])

$$\Gamma(x_1, x_2, \omega) = \exp[-2|x_1 - x_2|/l_c] \times \exp[-i\omega(x_1 - x_2) \sin \theta / U_c] \quad (91)$$

where $l_c \ll L$ is the correlation length, which is a measure of the coherence range of the turbulent pressure field; U_c is the convection velocity, or the velocity at which the turbulent eddies move with the flow; and θ is the angle between the direction of the flow and the normal to the axis of the tube. Note that $U_c/\sin \theta$ is the phase velocity of the pressure signal along the tube.

In the case of lightly damped structures with well-separated modes, cross modal contribution to the response can be ignored, and eq. (90) can be analytically evaluated to be (refs. [112] and [125]):

$$\bar{y}^2(x) = \sum_j 2\pi\xi_j\omega_j S_y(\omega_j) = \sum_j \pi\xi_j f_j G_y(f_j) \quad (92)$$

In the second equality, the response is expressed in terms of the more commonly used engineering variables frequency f (in Hz) and the single-sided power spectral density G as a function of f , where

$$G(f) = 4\pi S(\omega) \text{ for } f = \omega / 2\pi > 0 \\ = 0 \text{ for } f < 0 \quad (93)$$

Under the same assumption of light structural damping

$$\bar{y}^2(x) = \sum_j \frac{LG_f(f_j)\phi_j^2(x)}{64\pi^3 M_j^2 f_j^3 \xi_j} J_{jj}^2 \quad (94)$$

where $G_f(f_j)$ is the single-sided power spectral density in (force/length)²/Hz generated by the turbulent pressure field at the natural frequency f_j of the j th vibration mode.

N-1343 Design Procedures for Tubes and Beams in Turbulent Cross Flow

In most situations, the component of turbulent flow normal to the axis of a cylinder is a more dominant excitation mechanism than the parallel component. The exception occurs when the flow direction is parallel or barely inclined to the cylinder axis. Thus, cross flow analysis using the component of flow normal to the cylinder should always be made, supplemented by the parallel flow analyses of N-1345 at small angles of inclination.

The theory in the subsections that follow can be applied to one-dimensional structures in general, but the specific information on the statistics of the pressure field must be limited to tubes (circular cylinders) until adequate information is available for other beam cross sections.

N-1343.1 Uniform Cross Flow. In the simplest case of uniform cross flow over the entire length of a lightly damped, rigid cylinder with an evenly distributed total mass and spring supports at both ends, $\phi = L^{-1/2}$, $\theta = 0$ in eq. N-1342.1(91), and the joint acceptance integral of eq. N-1342.1(88), with $i = j = 1$, reduces to (refs. [112] and [127])

$$J_{11}^2 \approx l_c/L, \quad l_c \ll L \quad (95)$$

The correlation length l_c in most cross flows over a tube (circular cylinder) is no more than three diameters (see N-1322). Also, although eq. (95) were derived for the fundamental mode of the transverse vibrations of a rigid, spring-supported cylinder, they can be used to estimate the joint acceptance of the fundamental mode of cylinders

which are simply supported or clamped at both ends. Of course, in determining the RMS response with eq. N-1342.1(94), the mode shapes corresponding to the actual boundary conditions and normalized according to eq. N-1342.1(86) are used. For other boundary conditions and higher modes, the joint acceptance integral will have to be evaluated either numerically (most cases) or in closed form from eq. N-1342.1(91). Since $J_{11}^2 \leq 1.0$, (ref. [127]), an upper bound response estimate can be found by setting all the $J_{jj} = 1.0$ in eq. N-1342.1(94).

The random characteristics of the forces exerted on the tubes by the turbulent flow must be obtained from tests. Two expressions for the power spectral density of the turbulent force per unit length on tubes in a tube array are:

$$G(f) = \left[C_R(f) \rho V_g^2 D / 2 \right]^2 \quad (96)$$

(ref. [100]) and

$$G(f) = \left[C_L(f) \rho V_g^2 D / 2 \right]^2 (D / V_g) \quad (97)$$

(ref. [128]) where the gap velocity V_g is related to the velocity upstream of the tubes, V_∞ , by

$$V_g / V_\infty = P / (P - D) \quad (98)$$

The coefficient $C_R(f)$ in eq. (96) has units of $\text{sec}^{-1/2}$ and is given in Figure N-1343-1 as a function of the frequency f . Therefore, the application of eq. (96) should be limited to the parameter range for which the data were taken, namely, high turbulent water flow (1 to 2 m/sec) entering closely spaced heat exchanger tubes of 12 to 19 mm in diameter. The decrease in $C_R(f)$ with penetration into the bundle is attributed to the highly turbulent inlet flow and possible vortex excitation observed in the first few tube rows. Data for the nondimensional lift coefficient $C_L(f)$ of eq. (97) have not been obtained for as many tube array configurations (refs. [141] and [142]), but use of this alternative expression may predict less conservative responses (ref. [142]).

For an isolated tube in cross flow, which is not subject to conditions of lock-in vortex shedding (see N-1323), the power spectral density $G_f(f)$ and the correlation length are strong functions of the turbulence created in the incident flow stream by the upstream structures. Relatively small amounts of turbulence can cause significant reductions in the effectiveness of vortex shedding as an excitation mechanism, and all periodicity can be eliminated with sufficiently strong incident turbulence. For given turbulence intensities and scale lengths of the incident flow, $G_f(f)$ is available (ref. [129]) and the correlation length l_c may be approximated by the scale length of the incident flow. In the absence of specific information about the incident flow, the random turbulence coefficient for the upstream tube in Figure N-1343-1(a) can be used to estimate $G_f(f)$ for most isolated tubes in cross flow

because of the wide variety of incident flow conditions contained in the data base. In the latter case, the velocity used in eqs. (96) and (97) should be the free stream velocity of the flow.

If upstream structures produce well defined vortices, strong excitation mechanisms may be created on isolated cylinders more than twenty diameters downstream (ref. [130]). Such configurations should be avoided.

N-1343.2 Multiple Spans of Uniform Cross Flow. In many applications, a cylinder is subject to one or more partial spans of uniform, but different, velocity and density cross flows that are uncorrelated with each other or with the flow over the remainder of the cylinder's length. These conditions often exist when baffles are used to channel different density flows in the interior of the pressure vessels (heat exchangers, reactors, etc.). The mean square response for such conditions can be determined by simple generalizations of the results given in N-1342.2 for homogeneous turbulence excitation.

Since the uniform cross flow over the span of length L_i is uncorrelated with the uniform cross flows over the other spans, G_{jj}^2 in eq. N-1342.1(94) can be calculated (ref. [127]) by summing the products of the locally defined spectra G_f^i and joint acceptances $(J_{jj}^i)^2$ over all the spans i over which there is significant cross flow. Thus, the mean square response becomes

$$\bar{y}^2(x) \approx \sum_j \sum_i \frac{L_i G_f^i(f_j) \phi_j^2(x)}{64\pi^3 M_{jj}^2 f_j^3 \xi_j} (J_{jj}^i)^2 \quad (99)$$

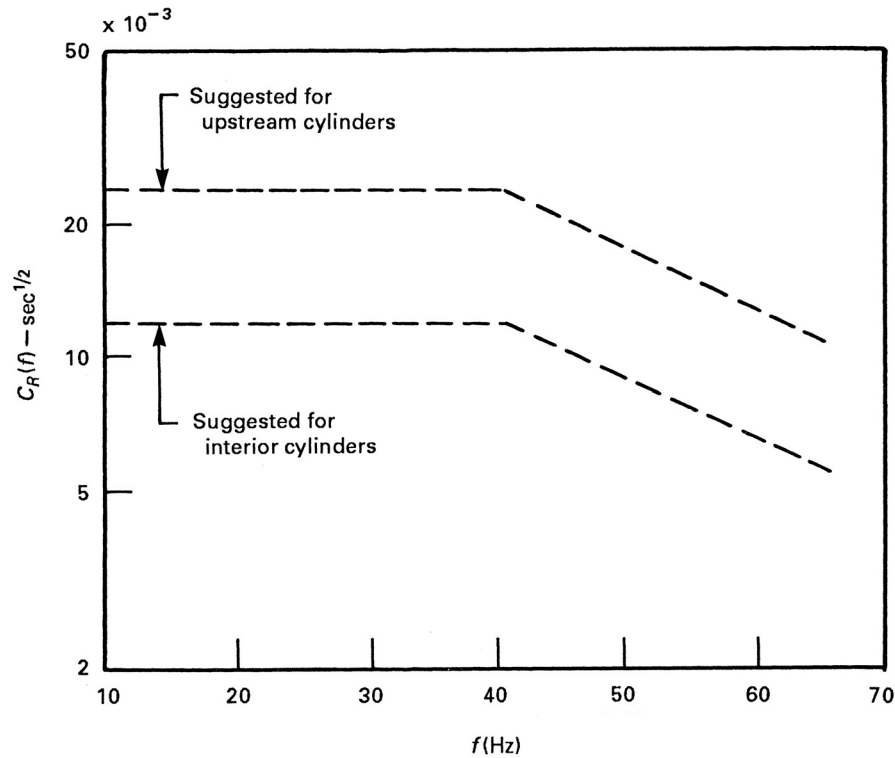
where

$$G_f^i(f_j) = G_f(f_j) \int_0^{L_i} \phi_j^2(x) dx \quad (100)$$

$(J_{jj}^i)^2$ are determined with eq. N-1342.1(88) using that part of ϕ_j , denoted by ϕ_j^i , that is active over the i th span with length L_i . As discussed in N-1343.1, if ϕ_j^i is similar to the fundamental mode shape of a one-span beam with simple or clamped supports at each end, then $(J_{jj}^i)^2 \approx l_c^i / L_i$. The correlation lengths inside a tube bundle are smaller than that for an isolated tube, being about 1–2 tube diameters. After specifying $G_f^i(f)$ and using eqs. N-1343.1(96) and N-1343.1(97) for instance, the mean square response can be determined.

N-1343.3 Nonuniform Cross Flow. In industrial heat exchangers, the cross flow velocities are seldom uniform over the entire length, or even one span of the tubes. While an average cross flow velocity can be used to estimate the

Figure N-1343-1
Random Excitation Coefficient for Arrays in Cross Flow (Ref. [100])



force spectra in eqs. N-1343.1(96), N-1343.1(97), and N-1343.1(98), when the velocity distribution is available, better estimates can be obtained by using mode shape-weighted power spectral densities similar to the generalized forces used in deterministic analysis (ref. [127]):

$$G_f(f_j) = \left(D/2 \right)^2 C_R^2(f_j) \int_0^L \left[\rho U^2(x) \right]^2 \phi_j^2(x) dx \quad (101)$$

for a single-span tube of uniform mass density and

$$G_f^i(f_j) = \left(D/2 \right)^2 C_R^2(f_j) \int_0^{L_i} \left[\rho_i U^2(x) \right]^2 \phi_j^2(x) dx \quad (102)$$

for a multi-span of spanwise uniform mass density. These estimates are not rigorously derivable, but they will lead to more accurate estimates of response, especially when the peaks in the velocity distributions are close to the antinodes of the vibration modes.

N-1344 Vortex-Induced Vibrations in a Tube Bundle

The existence of vortex shedding deep in a tube bundle is much less clearly defined than for a single cylinder. Experimental measurements involving tube bundles showed

that even if a resonance peak exists in the dynamic pressure power spectral density, it is much broader and not as well defined as in the case of a single tube. Furthermore, these peaks are bounded by the pressure power spectral density given by eq. N-1343.1(96). However, if lock-in vortex-induced vibration occurs in a particular span, the forcing function and the tube mode shape will be fully correlated and in-phase for that span. This means that the span joint acceptance, $J_{jj}^i = 1.0$. To be conservative, a lock-in vortex-induced vibration amplitude deep in a multi-span tube bundle can be calculated by substituting $J_{jj}^i = 1.0$ into eq. N-1343.2(99), for all spans and all modes where lock-in cannot be avoided or suppressed according to N-1324.1.

Classical vortex shedding does occur in the boundary tubes. For the first two to three rows of tubes in a tube bundle, vortex-induced vibration analysis following the procedure outlined in N-1320 is recommended.

N-1345 Cylinders in Axial Flow

Turbulence generally is a much weaker excitation mechanism in axial flow compared with cross flow, where the flow separates from the vibrating body. Also, axial flow is a source of flow damping which increases with flow rate

(refs. [81], [82], and [131]). As a result, RMS vibration amplitudes of tubes in axial flow are typically only a few percent of the tube diameter.

The surface pressure fluctuations that excite a tube in axial flow are due to many sources: local turbulence created by the shear flow in the developing boundary layer; free stream turbulence created by upstream disturbances (grid supports, abrupt changes in channel size, elbows, valves, etc.) that quickly attenuate downstream of the disturbance, localized acoustic noise (waves); and system acoustic noise that can propagate long distances (ref. [136]). For pipes and single rods in annuli subject only to fully developed flow, relatively general experimental characterizations of the homogeneous pressure fields are possible (ref. [131]), because they depend only on the local channel geometry and the flow rates. However, general characterizations have not been developed that account for upstream disturbances and adjacent bodies, although many specific systems have been studied (refs. [131] through [137]). Evidently, accurate predictions can be made when the pressure field is characterized in the same system as the response is measured, but the predictions from system to system may vary by an order of magnitude for the same axial flow velocity. Because response is usually much easier to measure than the pressure fluctuations necessary to characterize a pressure field, especially a nonhomogeneous one, empirical correlations of response have been developed for important component geometries (refs. [99] and [137]). The component and prototype tests upon which the correlations are based include all component geometries and excitation sources. Of course, the use of these correlations must be limited to the type of components and parameter variations for which they were developed.

N-1345.1 Recommended Design Procedures.

(a) When the characterization of the pressure field is available, then the response of the structure can be predicted by the general method outlined in N-1342. But, unlike cross flows, the convection velocity U_c is important and must be known in axial flow before the acceptance integral, eqs. N-1342.1(88) and N-1342.1(91), can be evaluated. In axial flows, it is not generally true that the larger the correlation length ℓ_c , the larger the response as in cross flows. Rather, the response is governed by the matching of the structural mode shape and the phase-coherence of the pressure field (refs. [112] and [122]), in addition to its power spectral density.

(b) Regardless of whether a pressure field characterization is available, the maximum amplitude of motion can be estimated to within an order of magnitude using the empirical correlation (ref. [99])

$$\frac{y^*}{D} = \left[\frac{5 \times 10^{-4} K_N}{\alpha^4} \right] \left[\frac{u^{1.6} \epsilon^{1.8} Re^{0.25}}{1 + u^2} \right] \times \left[\frac{D_h}{D} \right]^{-0.4} \left[\frac{\beta^{2/3}}{1 + 4\beta} \right] \quad (103)$$

if the cylinder parameters are within the ranges covered by the correlation:

$$2.1 \times 10^{-3} \leq u^2 = m_A v^2 L^2 / (EI) \leq 8 \times 10^{-1}$$

$$26.8 \leq \epsilon = L/D \leq 58.7$$

$$2.6 \times 10^4 \leq Re = VD/\nu \leq 7 \times 10^5$$

$$4.9 \times 10^{-4} \leq \beta = m_A / m_t \leq 6.2 \times 10^{-1}$$

$$2.10 \leq \alpha^2 = \Omega_1^2 [m_t L^4 / EI]^{1/2} \leq 20.8$$

$$1 \leq K_N \leq 5$$

where K_N is a noise factor representing a departure from quiet, steady axial flows of $K_N = 1$. Commercial systems are expected to be bounded by $K_N = 5$. E is the modulus of elasticity, I is the beam area moment of inertia, and ν is the fluid kinematic viscosity.

N-1400 DYNAMICS OF COUPLED FLUID-SHELLS

N-1410 INTRODUCTION

It is well known that the motion of a solid in a heavy liquid is different from that in a vacuum. In the case of a sphere or a cylinder moving in an infinite fluid medium, for example, the presence of the fluid can be accounted for by simply adding to the physical mass of the solid, the mass of fluid it displaces. Hence the term added mass, or hydrodynamic mass, has been used to describe this solid-fluid interaction phenomenon. When the fluid is viscous, then in addition to an added mass, there is an apparent added damping, or hydrodynamic damping, due to the viscosity of the fluid.

In the case of thin cylindrical shells vibrating with fluids entrapped in-between, the effect of fluid-structure interaction is far more complicated. Data from pre-operational tests of nuclear plants showed that the natural frequencies of the thermal shields of light water nuclear reactors were much lower than their corresponding values measured in-air. Since then, theoretical studies by Fritz and Kiss (ref. [143]), Horvay and Bowers (ref. [144]), Au-Yang (refs. [93] and [145]), Chen (refs. [90] and [146]), Krajcinovic (ref. [147]), and Levin and Milan (ref. [148]) conclusively showed that the coupling effect of narrow fluid gaps on thin cylindrical shells is much stronger than that of infinite fluid media. Their conclusions are supported by

laboratory tests (refs. [90] and [149]). A few review papers on this topic were written by Chen (ref. [90]), Brown (ref. [150]), and Au-Yang (ref. [151]).

This guide describes a simple method to account for the effect of fluid-structure interaction on the response of cylindrical shells coupled by fluid gaps, using the structural priority approach (ref. [155]). In this approach, the effect of fluid-structure interaction is completely accounted for by the “added mass” and the “added damping” terms. The principal advantage of this method is its simplicity. Once the added mass and the added damping terms are computed, the structural response analysis can be carried out by standard methods without any need to revise the computer programs either for calculating the structural response or for calculating fluid forcing function. On the other hand this method is based on linear dynamic theory in which the principal interest is to estimate low-frequency structural responses. The vibration amplitude of the structure must be small compared with the width of the fluid gaps in order that linear dynamic theory holds. Internal structures of commercial nuclear reactors such as the core support structure and the thermal shield are usually restricted by limiter blocks so that their motions are small compared with the fluid annular gap width. In flow-induced vibration, seismic and loss-of-coolant analyses of these components, usually only the responses of the lower few modes are of interest, and the methods described in this guide can be applied. Structural analyses involving large displacements or in which responses to the high frequency components of the forcing function cannot be ignored, or analyses in which the principal interest is to estimate the hydraulic forcing function rather than the structural responses, should not follow the methods described in this guide.

N-1420 NOMENCLATURE

- a = radius of inner cylinder or (when used as a subscript or superscript) pertaining to the inner cylinder
- b = radius of outer cylinder or (when used as a subscript or superscript) pertaining to the outer cylinder
- c = velocity of sound
- $C_{\alpha m}$ = Fourier coefficient of the m th cylinder axial mode onto the α th acoustic mode
- $[C]$ = equivalent viscous damping coefficient matrix
- f = frequency in Hz
- $\{f_o\}$ = externally applied force
- $\{f\}$ = reaction force due to fluid shell coupling
- $f_{m,n}$ = natural frequency of the (m,n) th mode
- h = Fourier component of the hydrodynamic mass matrix element
- H = hydrodynamic mass
- \bar{H} = generalized hydrodynamic mass
- I = modified Bessel function of first kind
- J = Bessel function of first kind

- K = modified Bessel function of second kind
- $[K]$ = stiffness matrix
- K_{ij} = element of the stiffness matrix
- ℓ = length of cylindrical shell
- m = axial modal index of the cylindrical shell
- n = circumferential modal index of the cylindrical shell
- p = pressure
- $q = (u, v, w)$, displacement vector of the shell
- r = radial coordinate or (used in Figure N-1451-1, = a/b)
- R = radial function of acoustic mode
- u = displacement of the shell in the axial direction
- v = displacement of the shell in the tangential direction
- w = displacement of the shell in the normal direction
- x = axial coordinate
- α = axial mode number for the acoustic mode
- β = circumferential mode number for the acoustic mode
- $\delta_{33} = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 1 \end{bmatrix}$ matrix in the (u, v, w) space
- $\epsilon = \alpha$ or $(2\alpha - 1)/2$ as defined in eq. N-1440(108)
- θ = azimuthal coordinate
- Θ = mode shape function
- λ = defined in eq. N-1451.2(115)
- μ = surface mass density of the shell
- ν = hysteretic damping factor; kinematic viscosity
- ρ = mass density of fluid
- ϕ = axial acoustic mode shape function
- ψ = axial cylinder mode shape function
- ω = angular frequency (rad/sec)

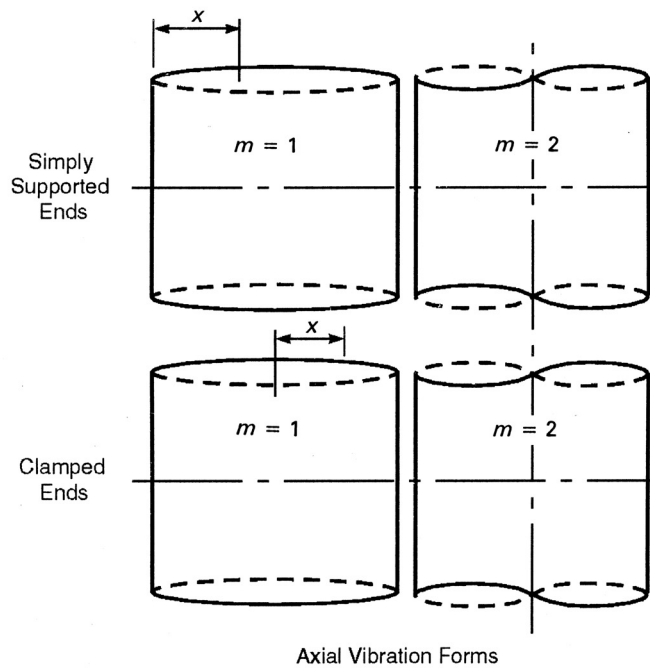
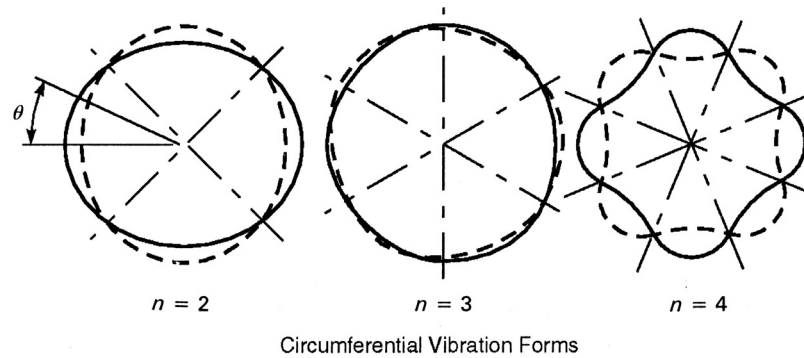
N-1430 FREE VIBRATION OF THIN CYLINDRICAL SHELL IN AIR

Figure N-1430-1 shows the vibration mode of a thin, finite cylindrical shell. Viewed from the ends, the vibration of the cylinder may consist of any number of waves distributed around the circumference. The number of circumferential waves is denoted by n , with $n = 1$ being the beam mode. Throughout this write-up, it is assumed that the circumferential mode shape is normalized to unity,

$$\int_0^{2\pi} \theta^2 \left(\theta \right) d\theta = 1 \quad (104)$$

When viewed from the side, the deformation of the cylinder consists of a number of waves distributed along the length of a generator. The number of half waves along a generator is denoted by m , $m = 1, 2, 3, \dots$. As shown in Figure N-1430-1, the axial wave forms depend on the end conditions of the cylinder.

Figure N-1430-1
Vibration Forms for Circular Cylindrical Shells



The equation of motion for the free vibration of the shell is,

$$[\mu]\{\ddot{q}\} + [K]\{q\} = 0 \quad (105)$$

Thus to determine the natural frequencies, one has first of all to calculate the modal stiffness matrix. In simple cases of cylinders with simply supported or clamped ends, simplified expressions for the stiffness matrix elements have been derived (refs. [92] and [152]), and the natural frequencies of the shell can be computed. There will be three roots for each set of K_{ij} , corresponding to three natural frequencies for each set of m, n . Two of these frequencies are usually much higher than the third one. These are

associated with vibrations predominantly in the axial and tangential directions. The lowest frequency, on the other hand, is associated with vibrations predominantly in the normal direction. This is the flexural mode and is usually the mode of interest in structural analysis.

N-1440 ACOUSTIC MODES OF A FLUID ANNULUS BONDED BY RIGID WALLS

The natural frequencies associated with the (α, β) th mode of the fluctuation pressure distribution

$$p_{\alpha\beta} = R_{\alpha\beta}(r)\phi(x)\cos\beta\theta \quad (106)$$

inside the fluid annulus are given by the roots of the equation (ref. [153]),

$$J'_\beta(x_{\alpha}^a)Y'_\beta(x_{\alpha}^b) = J'_\beta(x_{\alpha}^b)Y'_\beta(x_{\alpha}^a) \quad (107)$$

where J, Y are Bessel functions and,

$$x_{\alpha}^2 = (w/c)^2 - (\epsilon\pi/\ell)^2 > 0 \quad (108)$$

where

- $\epsilon = \alpha$ if the fluid annulus has both pressure released ($p = 0$) or both hard ($p = \max.$) ends;
- $= (2\alpha - 1)/2$ if one end is pressure released and the other end is hard.

Note that acoustic modes exist in the fluid annulus only if,

$$\omega/c > \epsilon\pi/\ell$$

The frequency at which,

$$\omega/c > \epsilon\pi/\ell$$

is known in acoustics as the coincidence frequency.

N-1441 Frequency Equation and Mode Shape for a Thin Fluid Annulus

For many applications, the fluid annulus is thin compared with its radius, so that the condition $(b - a)/a > 5$ is satisfied. Under this condition, the following approximate equation for the natural frequencies holds true for the plane wave mode (no radial node).

$$f_{\alpha\beta} = \left(c/2\pi \right) \left[\left(\epsilon\pi/\ell \right)^2 + \left[2\beta/(a+b) \right]^2 \right]^{1/2}$$

The second radial mode (with one radial node) usually has a lowest frequency much higher than that of the fundamental radial mode up to $\alpha = \beta = 5$. Thus, it can be ignored if only the lowest twenty or so acoustic modes are included in the analysis.

N-1450 FREE VIBRATION OF COUPLED FLUID-SHELL SYSTEMS

When the cylindrical shells bounding the fluid annulus are flexible, then not only are the motions of the shells coupled to the fluid, but they are coupled together by the fluid between them. It was shown that (ref. [145]) there is no cross circumferential model coupling between the fluid and a system of two coaxial, circular cylindrical

shells. However, except in the rare case when both the cylinder and the fluid can be represented by the same mode shape function, as is the case of a simply supported cylinder vibrating in a fluid annulus with open ends, there will be coupling between the axial structural and acoustic modes (refs. [93] and [145]).

N-1451 The Hydrodynamic Mass Matrix

It was shown that for small motions of the shell, the pressure induced by the structure in the fluid is proportional to the normal component of the acceleration \ddot{w} , of shell (refs. [93], [145], and [155]):

$$\{p\} = [H]\{\ddot{w}\} \quad (109)$$

The "constant" of proportionality is commonly called the hydrodynamic mass or added mass matrix. In general, it is a full matrix, the element of which is dependent on the frequency. For two coaxial cylindrical shells coupled by a fluid gap, the hydrodynamic mass matrix elements are (refs. [93] and [145]),

$$\begin{aligned} H_{mn}^{ab} &= \sum_{\alpha} C_{am}^a C_{an}^b h_{an}^{ab}, & H_{mn}^a &= \sum_{\alpha} (C_{am}^a)^2 h_{an}^a \\ H_{mn}^{ba} &= \sum_{\alpha} C_{am}^a C_{an}^b h_{an}^{ba}, & H_{mn}^b &= \sum_{\alpha} (C_{an}^b)^2 h_{an}^b \end{aligned} \quad (110)$$

where C_{am}^a , for example, is the projection of the m th axial mode shape, ψ_m of cylinder a (the inner cylinder) onto the α th axial acoustic mode shape ϕ_{α} of the fluid annulus:

$$C_{am}^a = \int_0^{\ell} \psi_m^a(x) \phi_{\alpha}(x) dx \quad (111)$$

C_{α}^b are similarly defined. h (and hence H) are in units of mass per unit area.

The general expressions for the hydrodynamic mass which are valid for any boundary condition and any D/ℓ ratio, are quite complicated, but still can easily be computed with the aid of a personal computer. In the following paragraphs, simplified expressions for computing h are given for several commonly encountered special cases.

N-1451.1 Slender Cylinder Approximation. When the conditions $|\epsilon\pi a/\ell - \omega a/c| \ll 1$ and $|\epsilon\pi b/\ell - \omega b/c| \ll 1$ are simultaneously satisfied, ref. [145] shows that:

$$\begin{aligned} h_{an}^a &= (\rho a/n)(b^{2n} + a^{2n})/(b^{2n} - a^{2n}) \\ h_{an}^{ab} &= -(2\rho b/n)(a^n b^n)/(b^{2n} - a^{2n}) \\ h_{an}^b &= (\rho b/n)(b^{2n} + a^{2n})/(b^{2n} - a^{2n}) \\ h_{an}^{ba} &= -(2\rho a/n)(a^n b^n)/(b^{2n} - a^{2n}) \end{aligned} \quad (112)$$

Note that the h s are independent of the axial acoustical wave number α in this case. From N-1452.1, the generalized hydrodynamic masses per unit length for the beam mode ($n = 1$) are,

$$\begin{aligned}\bar{H}^a &= \rho \pi a^2 (b^2 + a^2) / (b^2 - a^2) \\ \bar{H}^{ab} &= \bar{H}^{ba} = -2\rho \pi ab / (b^2 - a^2) \\ \bar{H}^b &= \rho \pi b^2 (b^2 + a^2) / (b^2 - a^2)\end{aligned}$$

N-1451.2 The Ripple Approximation. When the ratio $(b - a)/(b + a)$ is small, curvature effects become unimportant and the annular gap can be treated as a rectangular fluid region. In this case, the expressions for the hydrodynamic masses are greatly simplified to,

$$h_{an}^a = h_{an}^b = \rho [\lambda_{an} \tanh(\lambda_{an} d)]^{-1} \quad (113)$$

$$h_{an}^{ab} = h_{an}^{ba} = -\rho [\lambda_{an} \sinh(\lambda_{an} d)]^{-1} \quad (114)$$

where

$$d = b - a \text{ and,}$$

$$\lambda_{an}^2 = \left(\epsilon \pi / \ell \right)^2 + \left[2n / (a + b) \right]^2 - \left(\omega / c \right)^2 \quad (115)$$

N-1451.3 Incompressible Fluid Assumption. When $\omega/c \ll \epsilon \pi / \ell$, the term ω/c can be dropped from the argument of the Bessel function in eq. N-1440(107), the resulting hydrodynamic masses are independent of frequency. Since this condition is satisfied when $c \rightarrow \infty$, it is called the incompressible fluid assumption. However, the incompressible fluid assumption must be used with caution for applications to large structures. For a structure 300 in. (7.6 m) long, vibrating in water at 600°F (315°C) (e.g., nuclear reactor internal components), the incompressible fluid assumption generally introduces errors in the hydrodynamic matrix elements of more than 10% for frequencies above 20 Hz. Thus, it should not be used in analyses involving rapid transients in which high frequency contributions to the response are significant.

N-1451.4 Single Beam Mode ($n = 1$) in Narrow Annuli. When a cylinder vibrates inside a stationary, rigid cylinder of slightly larger diameter, its added mass due to the water between the two cylinders can be estimated by a very simple equation (ref. [143]):

$$h^a = \rho a^2 / [(b - a)(1 + 12a^2 / \ell^2)] \quad (116)$$

The above equation is valid only for the beam mode ($n = 1$) vibration. Figure N-1451-1 is a comparison between eq. (116) and the exact solution and shows

surprisingly good agreement between them as long as the fluid gap is small. From N-1452.1, the generalized, or effective hydrodynamic mass per unit length of the cylinder is:

$$\bar{H}^a = \frac{\rho \pi a^2}{(b - a)/a} \cdot \frac{1}{1 + 12a^2 / \ell^2} \quad (117)$$

N-1451.5 Single Cylinder Containing Fluid. The case when there is only one cylinder containing a fluid is given by (ref. [145]):

$$\begin{aligned}h_{an} &= \rho I_n(x_{\alpha} a) / [x_{\alpha} I'_n(x_{\alpha} a)] \quad \text{for } f < c\epsilon / 2\ell \\ &= \rho J_n(x_{\alpha} a) / [x_{\alpha} J'_n(x_{\alpha} a)] \quad \text{for } f > c\epsilon / 2\ell \\ &= \rho a / n \quad \text{for } f = c\epsilon / 2\ell\end{aligned} \quad (118)$$

where J_n , I_n are Bessel and modified Bessel functions of the first kind. From N-1452.1, for the beam mode ($n = 1$), the generalized hydrodynamic mass per unit length of the cylinder is,

$$\bar{H}_1 = \pi a h a \ell$$

N-1451.6 Single Cylinder in Infinite Fluid. The case where there is only one cylinder in an infinite fluid is given by

$$\begin{aligned}h_{an} &= -\rho K_n(x_{\alpha} a) / [x_{\alpha} K'_n(x_{\alpha} a)] \quad \text{for } f < c\epsilon / 2\ell \\ &= -\rho Y_n(x_{\alpha} a) / [x_{\alpha} Y'_n(x_{\alpha} a)] \quad \text{for } f > c\epsilon / 2\ell \\ &= \rho a / n \quad \text{for } f = c\epsilon / 2\ell\end{aligned} \quad (119)$$

From N-1452.1, for the beam ($n = 1$) mode, the generalized hydrodynamic mass per unit length is,

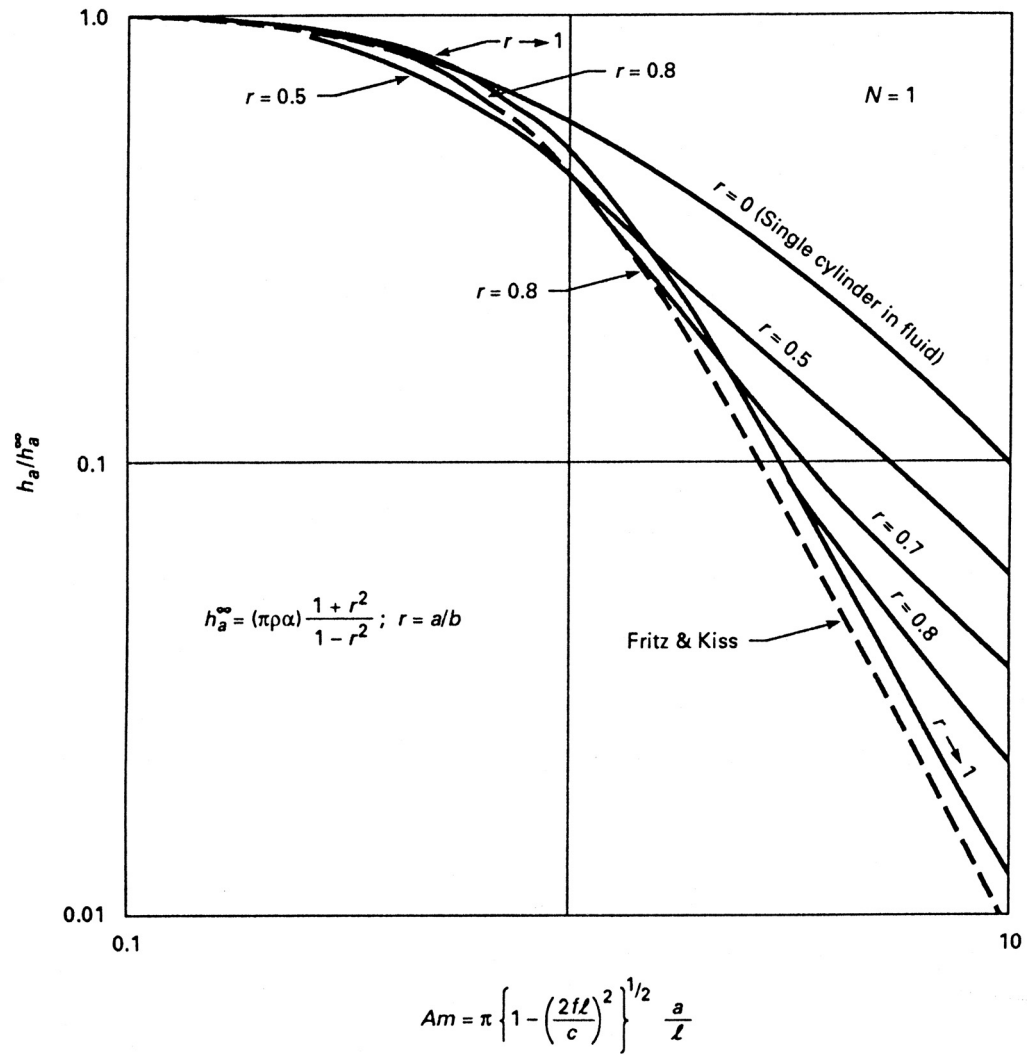
$$\bar{H}_1 = \pi a h a \ell$$

N-1452 Natural Frequencies of Coupled Fluid-Shells in Special Cases

Once the hydrodynamic mass matrix is computed, it can be input into finite element computer programs to calculate the natural frequencies of fluid coupled shells. First, several special cases in which simplified method of analysis can be used will be considered.

N-1452.1 One-Axial Mode Approximation. In many cases of practical importance, when the D/ℓ ratio of the cylinders are not very small, only the first axial mode is important in the dynamic analysis. Under these circumstances the frequencies of two coaxial cylindrical shells coupled by a fluid gap between them can be estimated by solving the following simplified coupled equations (ref. [154])

Figure N-1451-1
Comparison of Fritz and Kiss Solution With Exact Solution



GENERAL NOTE: h_a^∞ is the Fourier coefficient for the hydrodynamic mass surface mass density of an infinitely long cylinder.

$$\begin{aligned} \left(\mu^a + \bar{H}_{\text{ln}}^a \right) \left[\left(f_{\text{ln}}^a \right)^2 - f^2 \right] w^a - f^2 \bar{H}_{\text{ln}}^{ab} w^b &= 0 \\ -f^2 \bar{H}_{\text{ln}}^{ba} w^a + \left(\mu^b + \bar{H}_{\text{ln}}^b \right) \left[\left(f_{\text{ln}}^b \right)^2 - f^2 \right] w^b &= 0 \end{aligned} \quad (120)$$

where $[\bar{H}]$ is the generalized hydrodynamic mass matrix with elements:

$$\begin{aligned} \bar{H}_{\text{ln}}^a &= (q^a)^T [H^a \delta_{33}] (q^a) [(q^a)^T (q^a)]^{-1} = H_{\text{ln}}^a / (1 + n^{-2}) \\ \bar{H}_{\text{ln}}^b &= H_{\text{ln}}^b / (1 + n^{-2}) \\ \bar{H}_{\text{ln}}^{ab} &= H_{\text{ln}}^{ab} / (1 + n^{-2}) \end{aligned} \quad (121)$$

\bar{H}_{ln}^a , for example, is the generalized hydrodynamic mass of the inner cylindrical shell, expressed in terms of the hydrodynamic mass defined in eq. N-1451(110). From eq. (121) it can be seen that, because the hydrodynamic mass effects exist only in the normal direction, the effectiveness of H in reducing the natural frequencies of the shell is decreased by the factor $1 + n^{-2}$. For the beam mode ($n = 1$), its effectiveness is reduced by a factor of $1/2$. For high shell modes ($n \gg 1$), it is as effective as the physical mass.

For each n , two coupled frequencies and amplitude ratios for w_a/w_b can be obtained by solving the system of eq. (120). The torsional and longitudinal modes of vibration are not affected and are not coupled by the fluid gap in the inviscid fluid assumption. Note that $[H]$ is dependent on f , and eq. (120) has to be solved by iteration except in the incompressible fluid limit.

The above equations can be generalized to a system of several coaxial cylindrical shells coupled by fluid gaps between them (ref. [155]). However, since the hydrodynamic mass is frequency dependent, many iterations may be necessary to obtain all the coupled frequencies. This procedure may become prohibitive when there are more than three cylinders.

N-1452.2 Cases When Only One Cylinder is Flexible (ref. [156]). If only one cylinder is flexible, coupling between cylinders can be ignored. The only parameter of interest is the “in-water” frequency of the shell, which is also the natural frequency of the fluid-shell system. In this case, the natural frequencies can be obtained by a graphical method. If one plots the two functions

$$g_1 = f/f_{mn} \quad (122)$$

$$g_2 = \left[1 + \bar{H}_{mn}(f) / \mu \right]^{-1/2} \quad (123)$$

then where g_1 and g_2 intercept is a natural frequency of the coupled fluid-shell system. The lowest coupled frequency is usually referred to as the “in-water” frequency

of the shell while the higher coupled frequencies are usually referred to as acoustic frequencies. This distinction, however, has no technical basis.

In the incompressible fluid limit, only the “in-fluid” natural frequency of the cylinder is of interest. This is given by,

$$\tilde{f}_{mn} = f_{mn} \left[1 + \bar{H}_{mn} / \mu \right]^{-1/2} \quad (124)$$

N-1453 Use of Hydrodynamic Mass Matrix in Finite Element Structural Computer Programs

Closed form solutions are possible only in highly idealized cases of perfect circular cylindrical shells with classical boundary conditions coupled by perfectly uniform annular gaps. In many applications, even though the fluid gaps are annular and the uncoupled fluid dynamic problem is amenable to closed form solution, closed form solutions to the coupled fluid shell problem are not possible because the cylindrical shells are not uniform, and the boundary conditions are not classically simply supported or clamped. Examples of this sort are common in the nuclear industry. In these cases, the structural problem is usually solved with the help of finite element computer programs.

The effect of fluid-structure interaction can be readily incorporated into the finite element computer program using closed form solutions for the hydrodynamic mass matrix, provided that the finite element computer program is designed to accept a full mass matrix. Many commercially available finite element computer programs have this capability.

It was shown in ref. [155] that in finite element form, the hydrodynamic mass matrix can be written as:

$$\begin{aligned} \left[M_H \right] &= \begin{bmatrix} \frac{a^{-1} \sum_{\alpha} h_{\alpha}^a (\phi_{\alpha} \Delta A_a) (\phi_{\alpha} \Delta A_a)^T}{a^{-1} \sum_{\alpha} h_{\alpha}^{ba} (\phi_{\alpha} \Delta A_b) (\phi_{\alpha} \Delta A_a)^T} \\ \frac{b^{-1} \sum_{\alpha} h_{\alpha}^{ab} (\phi_{\alpha} \Delta A_a) (\phi_{\alpha} \Delta A_b)^T}{b^{-1} \sum_{\alpha} h_{\alpha}^b (\phi_{\alpha} \Delta A_b) (\phi_{\alpha} \Delta A_b)^T} \end{bmatrix} \end{aligned} \quad (125)$$

where ΔA_a , ΔA_b are element surface areas. These mass matrices can then be input into the finite element computer program just as the physical mass, with the exception that unlike the physical mass, the hydrodynamic mass, being a pressure, is effective only in the normal direction of the shell. Hence, the hydrodynamic mass should be associated only with the normal degree of freedom.

In the incompressible fluid approximation where the hydrodynamic mass matrix is independent of frequency, solution of the coupled fluid-shell dynamic problem is straightforward. In general, however, the hydrodynamic mass matrix is a full matrix with frequency dependent matrix elements. Solution of the coupled fluid-shell problem involves iteration between calculation of the mass matrix elements and the finite element computer program. When some of the coupled natural frequencies are near to the classical hard-walled acoustic modal frequencies, the hydrodynamic masses become extremely sensitive to the frequencies and solution to the coupled fluid-shell system can be difficult.

The method reported in ref. [149] greatly simplifies the above computational procedure by reducing the size of the matrix, while still giving reasonable estimates of the coupled frequencies of the fluid shell system. It consists of calculating the 2×2 "generalized" hydrodynamic mass matrix, \bar{H}_{mm}^a , \bar{H}_{mn}^b , \bar{H}_{mn}^{ab} , and \bar{H}_{mn}^{ba} of the cylindrical shells a and b , and uniformly distributing these hydrodynamic masses over the cylindrical surfaces. For the off diagonal elements, \bar{H}_{mn}^{ab} , \bar{H}_{mn}^{ba} , it is assumed that coupling exists only for elements of the cylinders that are directly opposite to each other. The resulting mass matrix is tri-diagonal instead of full, and greatly simplifies the computational procedure. The tri-diagonal hydrodynamic mass matrix approach yielded natural frequencies of a coupled fluid-shell system that agree quite well with those obtained from closed form solution.

N-1460 FORCED RESPONSE OF COUPLED-SHELL SYSTEM

When the motion of the shells is small, the equation of motion for forced response of a coupled fluid-shell system can be written as

$$[\mu]\{\ddot{q}\} + [C]\{\dot{q}\} + [K]\{q\} = \{f_0\} + \{f\} \quad (126)$$

where ν is the equivalent hysteretic damping factor, f_0 is the incident force acting on the shells, which is present irrespective of the motion of the shells. Examples of f_0 are random pressure caused by turbulent boundary layers, or transient pulses caused by shock waves impinging on the shells. f is the reaction pressure induced on the shell as a consequence of its own motion: a result of the interaction (or coupling) between the fluid and the structure. From eq. N-1451(109),

$$\{f\} = \{p \cdot \Delta A\} = [M_H][\delta_{33}]\{\ddot{q}\} \quad (127)$$

so that eq. (126) becomes,

$$([\mu] + [M_H][\delta_{33}])\{\ddot{q}\} + [C]\{\dot{q}\} + [K]\{q\} = \{f_0(t)\} \quad (128)$$

Equation (128) is exactly the same as the equation of motion in-air with the forcing function f_0 . The only difference is that added to the physical mass matrix $[\mu]$ is a

hydrodynamic mass matrix M_H , which is effective only in the normal direction of the shells. Thus, once the hydrodynamic mass matrix is computed, the dynamic problem can be solved routinely, by finite element structural analysis computer programs or otherwise, as if there were no coupling between the fluid and the structure.

N-1470 HYDRODYNAMIC DAMPING

Just as the fluid mass adds to the effective mass of cylindrical shells, the viscosity of the entrapped fluid adds to the effective damping of the coupled fluid-shell system. This hydrodynamic damping, or added damping, due to coupling of a cylindrical structure and a viscous fluid, was studied by Chen (refs. [90] and [146]), Yeh and Chen (Ref. [157]), and Mulcahy (ref. [158]). It was shown that the equivalent hydrodynamic modal damping ratio is dependent on a dimensionless parameter,

$$S = \omega a^2 / \nu, \quad (129)$$

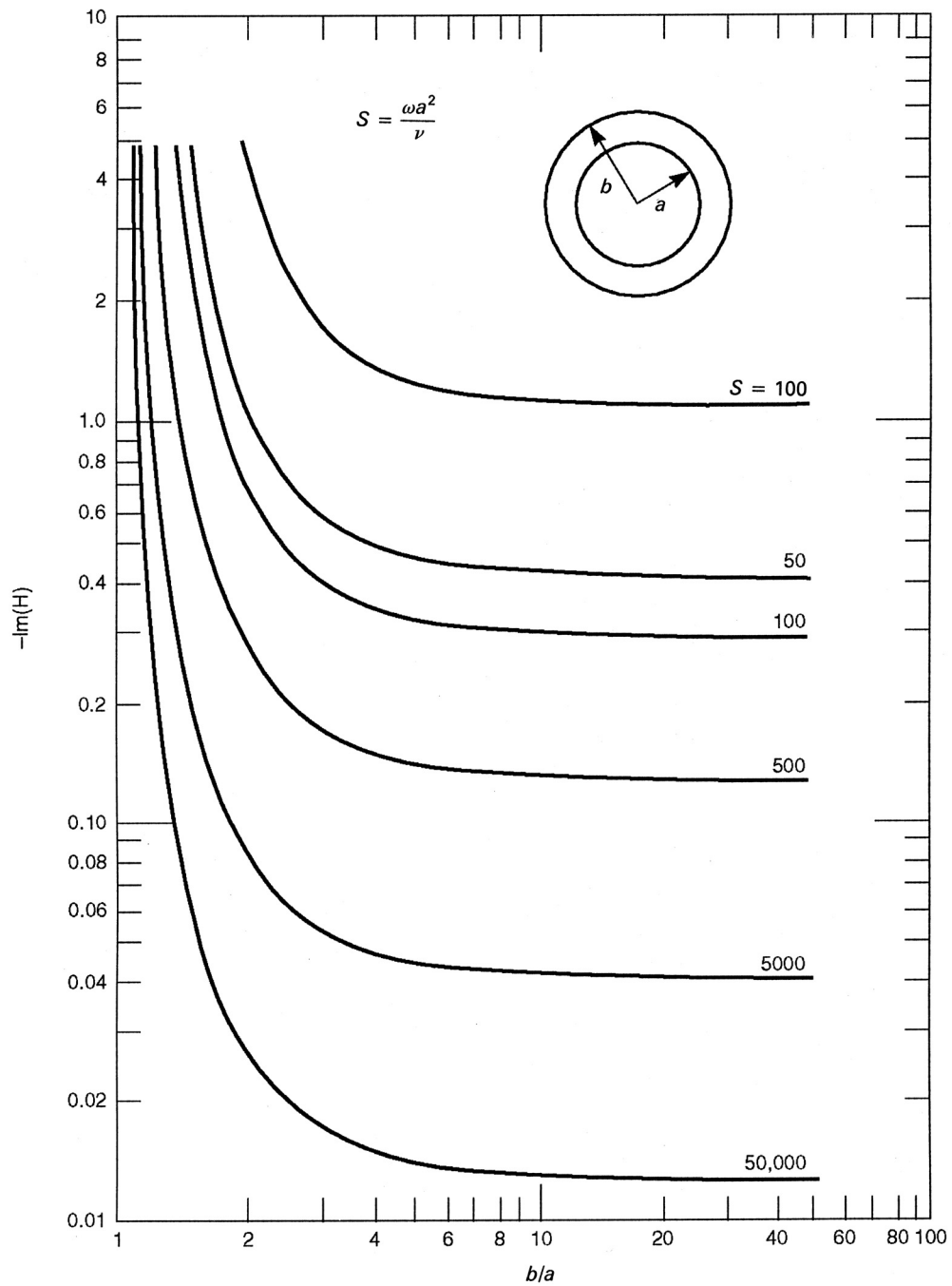
where ν is the kinematic viscosity of the entrapped fluid. The general expression for the hydrodynamic damping ratio, like that for the hydrodynamic mass, is quite complicated. Some insight, however, can be gained by studying the special case of the beam mode ($n = 1$) vibration of an infinitely long cylinder of radius a inside a stationary cylinder of radius b . In this case, the equivalent hydrodynamic modal damping ratio is given by

$$\zeta_1 = - \left[0.5\rho\pi a^2 / (M_H + \mu\pi a^2) \right] \text{Im}(Z) \quad (130)$$

where Z is a fairly complicated function of modified Bessel functions and S . Figure N-1470-1, reproduced from ref. [146], gives theoretical values of $-\text{Im}(Z)$ as a function of the radius ratio b/a , for different values of S , and shows that as $S \rightarrow \infty$, $-\text{Im}(Z)$ becomes infinitesimal even for small ratios of b/a , indicating that hydrodynamic damping is negligible for large values of S . As an example, for the vibration of the core support structure of a typical nuclear reactor inside the reactor vessel, $S \approx 10^9$. Thus, the hydrodynamic damping in this case is negligible. This conclusion, however, may not be true for the vibration of a thin rod inside a fluid-filled jacket of slightly larger diameter.

Although the above example is for the beam mode vibration of an infinitely long cylinder, the same conclusion holds true qualitatively for the shell mode vibration of finite cylindrical shells. This theoretical deduction is also supported by experimental observations. Damping ratios of thermal shields measured during pre-operational tests of pressurized water reactors show little difference from their corresponding values measured in-air. Thus, hydrodynamic damping in large cylindrical structures can be ignored.

Figure N-1470-1
Imaginary Part of Z as a Function of b/a for Selected Value of S (Ref. [146])



N-1500 FLUID TRANSIENT DYNAMICS

In the course of preparation.

N-1600 MISCELLANEOUS IMPULSIVE AND IMPACTIVE LOADS

In the course of preparation.

N-1700 COMBINED RESPONSES**N-1710 DYNAMIC RESPONSE COMBINATION**

In the design of nuclear power components it is necessary to consider the combined responses caused by two or more different sources of dynamic loading. There are two situations which arise: one in which the time phase relationship between the two or more responses is known (deterministic); and a second, where the time phase relationship is said to be random. In the first situation, the response may be determined by the algebraic summation method of N-1720. In the second case where the time phase relationship is random, one of the three methods N-1721, N-1722, or N-1723 may be used depending upon satisfying the qualifying conditions contained in N-1721, N-1722, or N-1723. The owner, either directly or through his designee, may prescribe whether the time phase relationship of individual load responses is to be considered deterministic or random. Consideration may be given to the probability of occurrence of the associated events which cause the combined loading in arriving at detailed procedures for combining randomly phased responses (see discussion in N-1725).

N-1720 ALGEBRAIC SUMMATION

When the time phase relationship between two or more time history responses is known and the dynamic system is linear, the individual collinear responses may be algebraically added to determine the combined maximum and minimum structural response. If the dynamic system is nonlinear, then the combination loading should be input into the dynamic system analysis in order to determine the combination response.

N-1721 Peak Combined Response

In those cases where the dynamic system is linear and where two or more responses are to be combined, the peaks of the individual collinear responses can always be conservatively added to obtain the combined response irrespective of whether the phase relationship of the responses is undefined, deterministic, or random.

N-1722 SRSS (Square Root of Sum of Squares) Method

N-1722.1 For Responses With Nearly Equal Dominant Frequencies. In those cases where the dynamic system is linear and where (a) the time phasing of the peak

individual collinear responses is random and (b) all of the individual responses have nearly an equal dominant³⁶ frequency with arbitrary amplitude, the SRSS method may be used to determine the combined design response (refs. [68], [69], and [70]).

N-1722.2 For Uncorrelated Dynamic Responses. The intent of this method for combination of transient dynamic responses is to achieve a conditional nonexceedance probability of at least 50% for the peak combined response of the system, component, or element considered. That is, there is a 50% probability that the magnitude of the summed responses will not exceed the SRSS value, given that the combined events have occurred. This goal is achieved by compliance with the following criteria provided that the amplitude of loads or accelerations for each input are conservatively represented (approximately at the level of the 84th percentile or the mean plus one standard deviation of the input amplitude).

Criterion. Dynamic or transient responses of linear structures, components, and equipment arising from combinations of dynamic loading or motions may be combined by SRSS provided that each of the dynamic inputs or responses has a *limited number of peaks* of force or acceleration and *approximately a zero mean*, and that the individual component inputs can be considered to be *relatively uncorrelated* (refs. [75] to [78]). Specific criteria for these three requirements are given below.

The requirement of a *limited number of peaks* is satisfied if any one of the following three criteria is met.

(a) For each response time history in the combination:

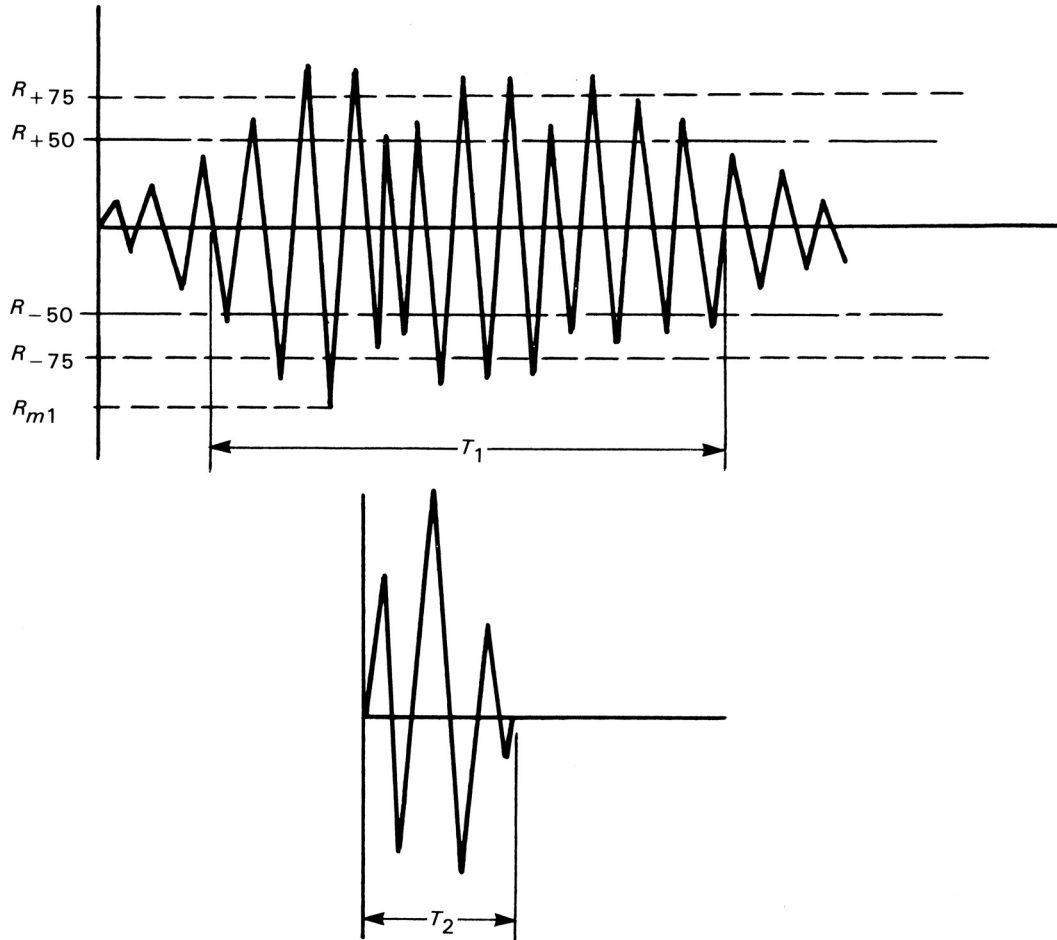
$$\frac{T_{75}}{\Delta T} \leq 0.02$$

and

$$\frac{T_{50}}{\Delta T} \leq 0.08$$

where T_{75} represents the summation of time intervals for all peaks over which the response exceeds 75% of either the maximum or minimum response (whichever is greater), T_{50} represents the time over which the response exceeds 50% of either the maximum or minimum response (whichever is greater), and ΔT represents the total time interval over which the maximum responses are expected to occur. For the case of random relative start times, ΔT represents the time interval during which the strong responses may overlap (generally taken as the time interval from the zero crossing time just preceding the first peak that exceeds 50% of the absolute maximum response to the zero crossing time just following the last peak that exceeds 50% of the absolute maximum response, of the longer time history). The strong motion of a seismic time history is discussed in N-1212.2(f). In determining the time T_{75} and T_{50} , only the durations of peaks with the same sign (positive or negative) are additive. These definitions are illustrated in Figure N-1722.2-1.

Figure N-1722.2-1
Definition of Notation



R_{m1} = absolute maximum response
 $R_{+75} = 0.75 |R_{m1}|$
 $R_{-75} = -0.75 |R_{m1}|$
 T_{+75} = summation of time intervals that response exceeds R_{+75}
 T_{-75} = summation of time intervals that response exceeds R_{-75}
 T_{75} = larger of T_{+75} or T_{-75}
 $\Delta T = T_1$, since $T_1 > T_2$

(b) For each input-time history in the combination:

$$\frac{T_{75}}{\Delta T} \leq 0.01$$

and

$$\frac{T_{50}}{\Delta T} \leq 0.04$$

where the definitions are the same as above, except for the substitution of input for response.

(c) If criteria (a) or (b) above are not met by each of the responses or inputs, respectively, in the combination, this requirement is still satisfied if the effective time ratios $(T_{75}/\Delta T)_e$ and $(T_{50}/\Delta T)_e$ for the response combination meet the requirements of (a) above. The effective time ratios for the response combination are given by:

$$\left(\frac{T_{75}}{\Delta T}\right)_e = \sqrt{\frac{\sum_{i=1}^N (R_m T_{75} / \Delta T)_i^2}{\sum_{i=1}^N (R_m)_i^2}}$$

where i represents each individual response in the combination and $(R_m)_i$ represents the maximum amplitude of that response component. The effective time ratio $(T_{50}/\Delta T)_e$ is obtained by substituting 50 for 75 in the above equation.

The requirement of *approximately a zero mean* is satisfied if, for the duration of strong input (or response), the ratio of the mean to maximum input (or response) is less than 0.20. The mean value is:

$$\mu = \frac{1}{N} \sum_{j=1}^n R_j$$

where μ is the mean value, R_j is the response at time increment j , and N is the total number of equally spaced time increments.

If this requirement is not met, responses can be divided into two parts (nonzero mean part and peak response relative to mean response) provided the response function is not skewed from the axis of the mean μ . The nonzero mean part of response can be combined algebraically (accounting for the sign of the mean response) and the peak responses relative to the mean responses can be combined SRSS if these relative responses meet the provisions of this section.

The requirement of *relatively uncorrelated* inputs is met if either of the following is satisfied:

(1) the individual dynamic inputs or responses are from events such that their relative start times are uncertain;

(2) if the relative start times are known, then the coefficient of correlation must be less than 0.3.

N-1723 SRSSE (Equivalent SRSS) Method

(a) In those cases where individual linear system time history responses have randomly defined time phase relationships and condition (b) of N-1722 is not satisfied, the SRSSE method may be used to determine the combined design response. As an illustration, N-1723.1 and N-1723.2 describe a detailed numeric procedure for determining the combinations for the case where two individual time history responses have a uniform random time phase relationship.

(b) For the case when three or more linear time history responses are to be combined, the procedures of N-1724 may be used to determine the combined SRSSE design responses (ref. [68]).

N-1723.1 SRSSE Method for Two Time History Responses. A method may be constructed to obtain the SRSSE response combinations as given in (a), (b), and (c).

(a) Determine one time history as a reference function. Bring the origin of the second function coincident with the origin of the reference function. Care should be taken to determine the reference function so that the time shift of the second function is consistent with the actual sequence of events expected. The origin of the second function could occur at, before, or after the origin of the reference function. The case where the origins are coincident will be discussed in (b) and (c). The procedure for the other two cases are similar.

(b) Shift the second function in small time steps relative to the reference function until the origin of the second function goes to the end of the reference function (see Figure N-1723.1-1). For each time phase difference τ , a scan is made from the time zero to $\tau + T_2$, if $(\tau + T_2) \geq T_1$, or from the time zero to $(\tau + T_2) < T_1$, to determine the largest amplitude (peak) of the sum of the two functions. T_1 and T_2 are the time duration of the reference function and second function, respectively. The algebraic maximum and minimum of these peaks, Maximum Combined Amplitude (FF), are plotted as functions of the time phase difference τ . The resulting function is called the Amplitude Phase Function (APF) which is shown in Figure N-1723.1-3.

(c) The functional relationship between the APF and the Probability Density Function (PDF) for the time shift (Figure N-1723.1-2) is then used to compute the Cumulative Distribution Functions (CDF) of the maximum and minimum combined amplitudes as detailed in N-1723.1.1 (Figure N-1723.1-4). The choice of the appropriate PDF for τ should be developed on a case-by-case basis. The uniform PDF is frequently a reasonable assumption when there is a weakly known cause and effect between the initiation of the causative events.

N-1723.1.1 Cumulative Distribution Function (CDF). Both the maximum and minimum Cumulative Distribution Functions need to be constructed.

Figure N-1723.1-1

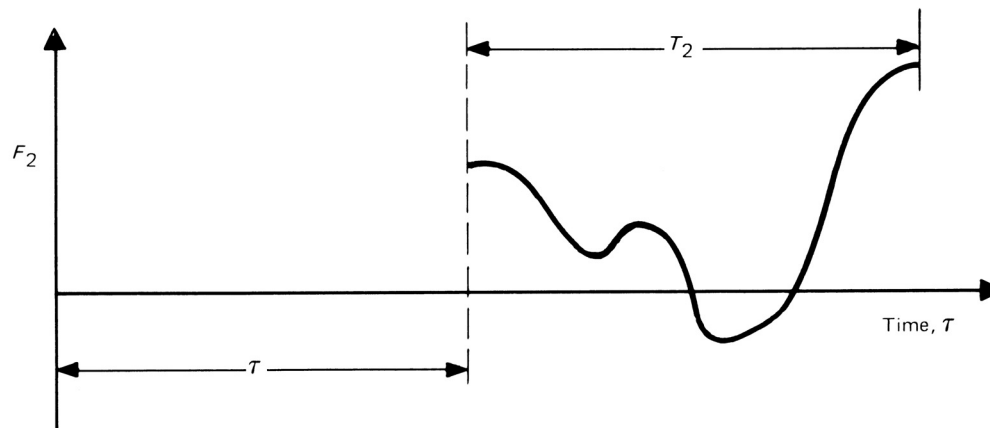
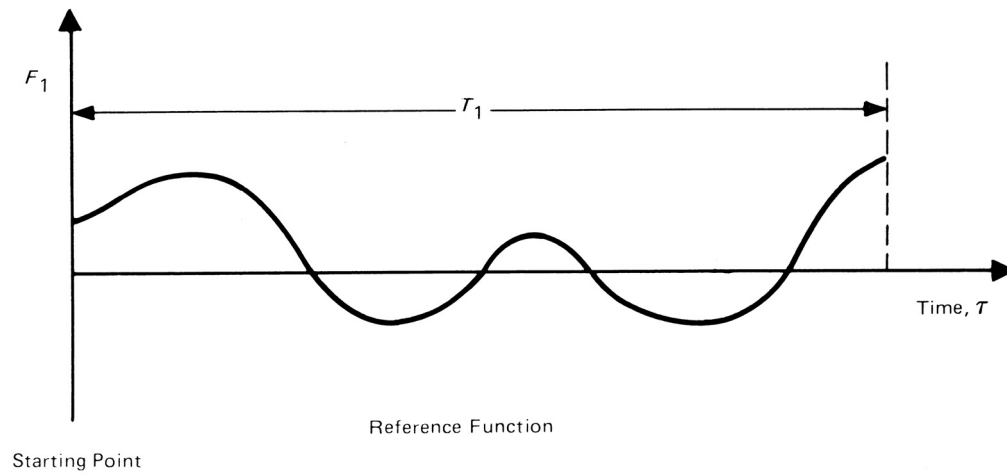


Figure N-1723.1-2

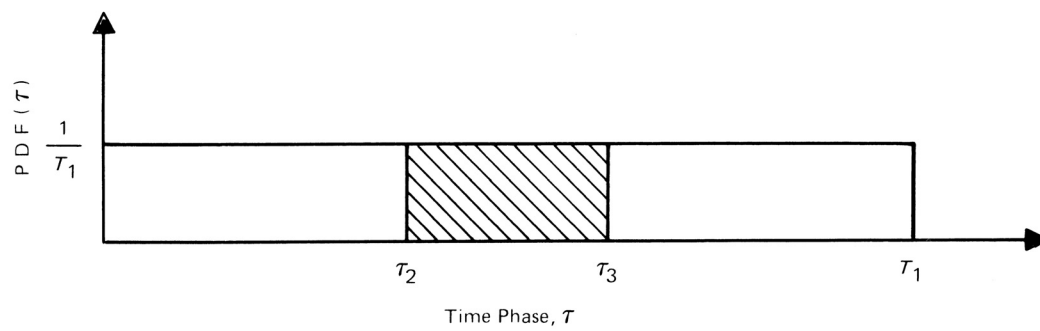


Figure N-1723.1-3

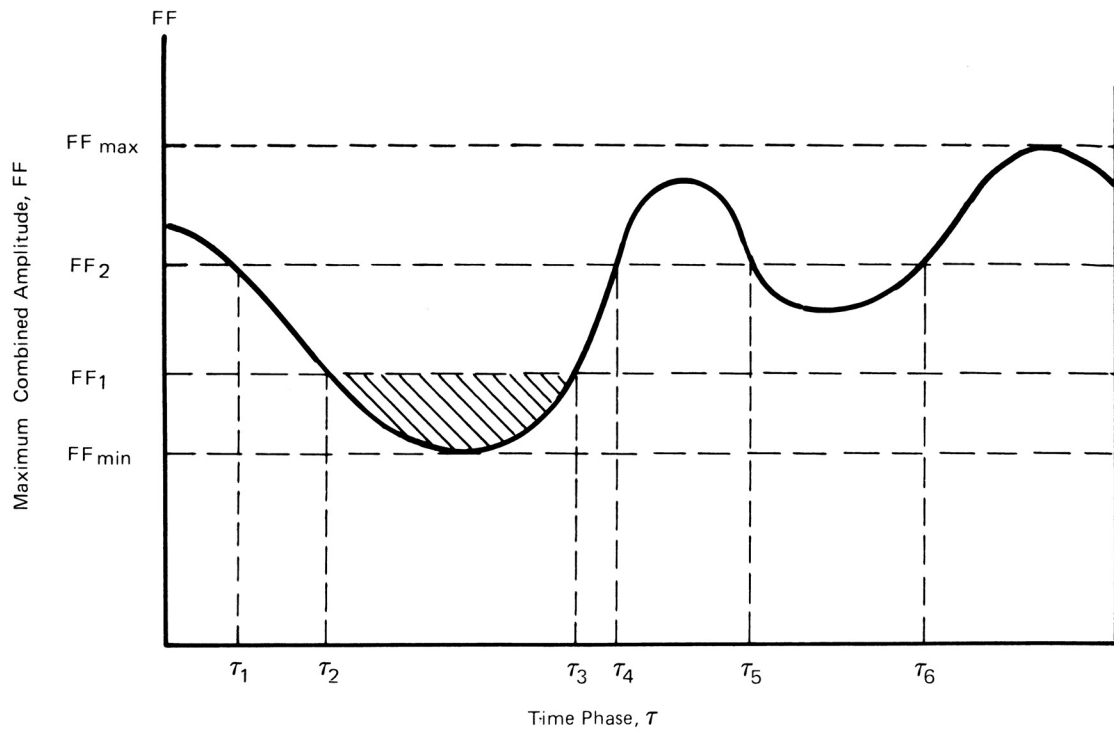
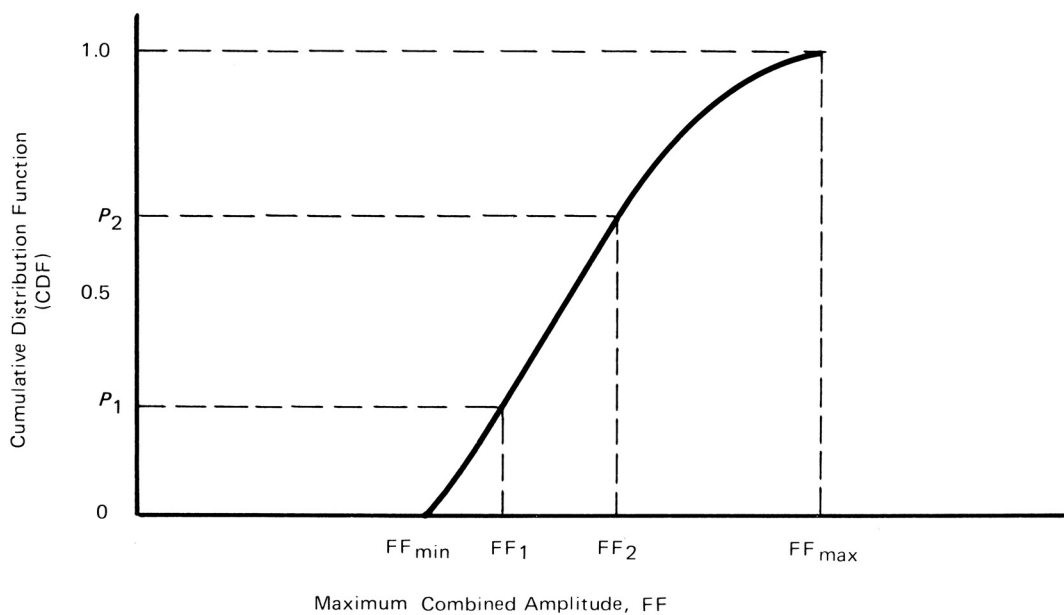


Figure N-1723.1-4



(a) From Figure N-1723.1-3, which illustrates the maximum APF function, it can be seen that any possible combined amplitude FF falls between two extreme limits FF_{\max} and FF_{\min} . It is clear that the probability that the value of FF is less than FF_{\max} is one, and the probability that the value of FF is less than FF_{\min} is zero. Hence, in Figure N-1723.1-4, we can immediately plot these two extreme points.

(b) The intermediate points on the CDF are found by integration of the distribution of the time shift τ between appropriate limits. Hence, at this point, the choice of the distribution for τ enters the calculations.

(c) Calculation of the intermediate points on the CDF requires the consideration of the functional relationship between the maximum combined amplitude FF and the time phase shift τ . For a given value of FF on Figure N-1723.1-3 (in this case FF_1) there is an interval on the τ axis (τ_2, τ_3) in Figure 1723.1-3. This same τ interval (τ_2, τ_3) can also be observed on the Probability Density Function for τ in Figure N-1723.1-2. The shaded area of Figure N-1723.1-2 represents the probability that τ falls in the interval (τ_2, τ_3). The event of τ falling in the interval (τ_2, τ_3) corresponds exactly to the event that FF is less than or equal to FF_1 . Therefore, the ordinate P of Figure N-1723.1-4 corresponding to FF_1 can be computed as

$$P_1 = \int_{\tau_2}^{\tau_3} \text{PDF}(\tau) d\tau \quad (131)$$

and for the case where

$$\text{PDF}(\tau) = 1/T_1 \quad (132)$$

then

$$P_1 = (\tau_3 - \tau_2)/T_1 \quad (133)$$

(d) The same procedure is followed for other levels of FF. Note that care must be exercised when integrating multiple valued functions to avoid a loss of information. If the level of FF selected contains more than one interval of τ , all intervals must be integrated to obtain the associated probability. For example, in the case of FF, the integration limits in eq. (c)(131) would be from τ_1 to τ_4 and from τ_5 to τ_6 . The result of the integration is designated P_2 and plotted in Figure N-1723.1-4.

(e) Selecting different levels of FF and repeating the above procedure, the CDF function as shown in Figure N-1723.1-4 can be constructed.

(f) This entire procedure is then repeated for the minimum APF function to find its corresponding CDF.

N-1723.1.2 Cumulative Distribution Function for the Absolute Maximum. An alternative and more conservative procedure than that given in N-1723.1.1 may also be used to generate the CDF of combined responses. In this alternative procedure, instead of summing the positive and negative peaks of the response separately as in

N-1723.1.1, the absolute value of the peak combined response is identified for each value of τ when generating the APF. In this alternative procedure, a single APF is generated which corresponds to a combination of both positive or negative combined responses, whichever is greater in absolute magnitude for each value of τ . The procedure to construct the absolute maximum CDF is the same as stated in N-1723.1.1, except that only one CDF is generated for each response combination instead of two.

N-1723.2 SRSSE Values. From the maximum and minimum CDF determined in N-1723.1.1, select the values for SRSSE maximum and SRSSE minimum combinations corresponding to the 50% probability levels. It should be noted that these 50% levels are conditional probability levels given that the causative events have occurred. In order to determine total probability levels associated with the SRSSE values, it is necessary to multiply by the corresponding simultaneous combined event occurrence as illustrated in ref. [70].

N-1724 SRSSE Method for Three or More Functions

A Monte Carlo simulation procedure (refs. [69] and [70]) will generally be required to generate the CDF if there are more than two time history response functions to be combined. The SRSSE maximum, minimum, or absolute maximum values are at the same 50% probability levels used in N-1723.2.

N-1725 Alternative CDF Values

The SRSSE values of N-1723.2 and N-1724 are at a conditional nonexceedance probability level of 50%. That is, there is a 50% probability that the magnitude of the summed responses will not exceed the SRSSE values, given that the combined events have occurred. When the peak combined response values of N-1721 are used, the 100% nonexceedance values result. Since the use of either of these levels (50%, 100%) must be specified through consideration of systems requirements and service levels, alternative nonexceedance probability levels between 50% and 100% may also be used.

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NONMANDATORY APPENDIX O

ARTICLE O-1000 RULES FOR DESIGN OF SAFETY VALVE INSTALLATIONS

O-1100 SCOPE AND DEFINITIONS

O-1110 SCOPE

(a) The scope of [Nonmandatory Appendix O](#) is confined to the design of the safety valve installations as defined in [O-1120](#). The loads acting at the safety valve station will affect the bending moments and stresses in the complete piping system, out to its anchors and/or extremities, and it is the designer's responsibility to consider these loads. This Appendix, however, deals primarily with the safety valve installation and not the complete piping system.

(b) The design of the safety valve installation requires that careful attention be paid to all loads acting on the system, the forces and bending moments in the piping and piping components resulting from the loads, the loading and stress criteria, and general design practices. All components in the safety valve installation must be given consideration, including the complete piping system, the connection to the main header, the safety valve, valve and pipe flanges, the downstream discharge or vent piping, and the system supports. The scope of this Appendix is intended to cover all loads on all components.

(c) This Appendix has application to either safety, relief, or safety relief valve installations. For convenience, however, the overpressure protection device is generally referred to as a safety valve. The loads associated with relief or safety relief valve operation may differ significantly from those of safety valve operation, but otherwise the rules contained herein are equally applicable to each type of valve installation.

(d) Pressure relief safety valve stations require detailed analysis. In performing its design function the station is subject to dynamic loading from the structural response to both thermal and hydraulic forces and, in some instances, significant impact loading from the valve mechanism. The resultant loading is a mechanical load.

(e) This Appendix provides guidance for design and analysis of the piping components of a safety valve station. The guidance is presented by discussing four areas of consideration

- (1) load computation — open discharge system
- (2) stress evaluation — open discharge system

(3) closed discharge system

(4) general design considerations

O-1120 DEFINITIONS

(a) *Safety Valve*. An automatic pressure relieving device actuated by the static pressure upstream of the valve and characterized by full opening pop action. It is used for gas or vapor service.

(b) *Relief Valve*. An automatic pressure relieving device actuated by the static pressure upstream of the valve which opens further with the increase in pressure over the opening pressure. It is used primarily for liquid service.

(c) *Safety Relief Valve*. An automatic pressure actuated relieving device suitable for use either as a safety valve or relief valve, depending on application.

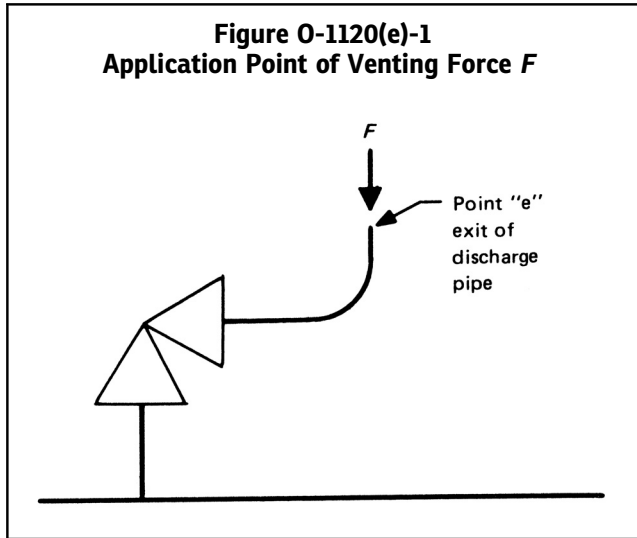
(d) *Power-Actuated Pressure Relieving Valve*. A relieving device whose movements to open or close are fully controlled by a source of power (electricity, air, steam, or hydraulic). The valve may discharge to atmosphere or to a container at lower pressure. The conditions, and such effects, shall be taken into account. If the power-actuated pressure relieving valves are also positioned in response to other control signals, the control impulse to prevent overpressure shall be responsive only to pressure and shall override any other control function.

(e) *Open Discharge Installation*. An installation where the fluid is discharged directly to atmosphere or to a vent pipe that is uncoupled from the safety valve. [Figure O-1120\(e\)-1](#) shows a typical open discharge installation with an elbow installed at the valve discharge to control direction of flow. [Figure O-1120\(e\)-2](#) shows a typical open discharge system discharging into a vent pipe. The values for l and m in [Figure O-1120\(e\)-2](#) are upper limits for which the rules for open discharge systems may be used.

(f) *Closed Discharge Installation*. An installation where the effluent is carried to a distant spot by a discharge pipe which is connected directly to the safety valve.

(g) *Safety Valve Installation*. The safety valve installation is defined as that portion of the system shown on [Figures O-1120\(e\)-1](#) and [O-1120\(e\)-2](#). It includes the run pipe, the branch connection, the inlet pipe, the valve, the discharge

Figure O-1120(e)-1
Application Point of Venting Force F



piping, and the vent pipe. Also included are the components used to support the system for all static and dynamic loads.

O-1200 METHOD OF AND PROCEDURE FOR LOAD COMPUTATION

O-1210 BASIC CONSIDERATIONS

(a) The load computation includes two thermodynamic computations: evaluation of momentum effects and evaluation of pressure effects. The response computation of the piping system includes consideration of transient, dynamic loading effects of the sudden opening and closing

valve action. The loads computation is combined with other specified loads and translated to a stress computation which is then compared to the Code allowable stress for acceptability.

(b) The basic principles of analysis of a pressure relief valve station are generally applicable to both open and closed discharge systems. The application of these basic principles may be quite different.

O-1220 OPEN SYSTEM — DISCHARGE THRUST

(a) The limiting dimensions for safety valve arrangements are shown by Figure O-1120(e)-2. The determination of the reaction force F value(s) is the responsibility of the piping system designer.

(b) The steady-state load due to steam reaction force from the opening and subsequent venting of the safety valve shall include consideration of both momentum and pressure effects and may be computed by the formula

$$F = (W/g)V_e + (P_e)A$$

where

A = exit flow area at point e

F = reaction force

g = gravitational constant, 32.2 lbf-ft/lbf-sec² (9.81 m/s)

P_e = static gage pressure at point e

V_e = exit velocity at point e

W = mass flow rate (relieving capacity stamped on the valve $\times 1.11$ — adjust for units to be compatible, if necessary)

The reaction force F is a design mechanical load that requires structural equilibrium for system stability and is applied as shown in Figure O-1120(e)-1.

(c) To ensure consideration of the effects of the suddenly applied load F , a dynamic load factor DLF, based on the relief/safety valve opening time and system dynamic characteristics, shall be applied to the forces and moments due to the reaction force F .

(d) Instead of a simplified dynamic analysis with the application of the DLF a dynamic hydraulic/structural system analysis may be performed.

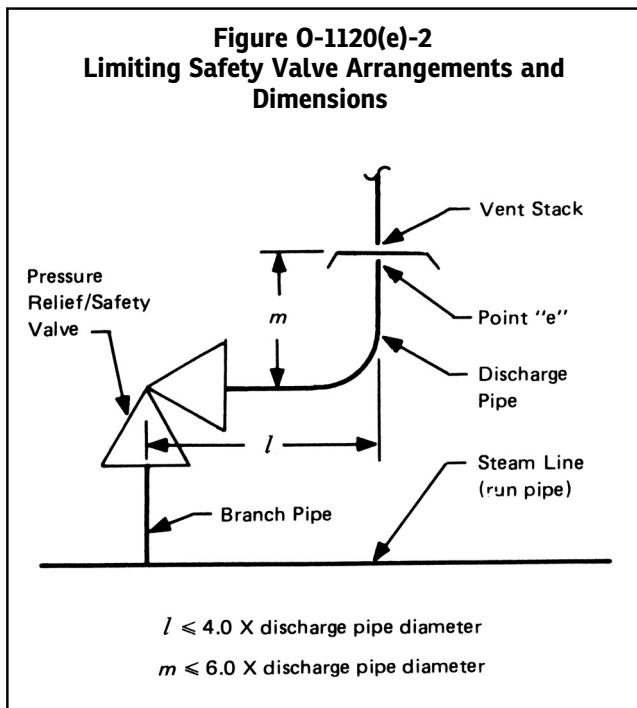
O-1230 OTHER MECHANICAL LOADS

Other mechanical loads to be considered should include, as a minimum

(a) interaction loads on the run pipe when more than one valve releases, and

(b) the transient impacting of the valve mechanism opening and closing, if applicable

Figure O-1120(e)-2
Limiting Safety Valve Arrangements and Dimensions



O-1300 STRESS EVALUATION OPEN SYSTEM

Evaluation of stresses due to the design mechanical loads may be made by using the rules for Class 1 piping, [O-1310](#) and for Class 2 or Class 3 piping, [O-1320](#).

O-1310 CLASS 1 PIPING

(a) Whenever any of the equations of NB-3650 are used in the analysis of Class 1 piping systems, the value of M_i shall include the reaction force moment. The contribution from the reaction force F to the branch moment M_b , as defined in Note (5) of Table NB-3682.2-1, shall be no less than the product $F \times$ nominal discharge pipe size \times DLF.

(b) Note that the use of the equations of NB-3650 for branch connections requires a nozzle spacing as defined by NB-3686.1(c).

(c) When eq. (9) of NB-3650 is used in the analysis, the value of M_i shall be defined as:

M_i = resultant moment due to a combination of primary loads. Loads to be considered include: weight; earthquake, considering only one-half the range of the earthquake and excluding the effects of anchor displacement due to earthquake; thrusts from relief and safety valve loads from pressure and flow transients; and other sustained mechanical loads.

(d) The combination of loads shall be specified in the Design Specification. In the combination of loads, all directional moment components in the same direction shall be combined before determining the resultant moment; i.e., resultant moments of loads shall not be combined.

(e) When eq. (10) of NB-3650 is used in the analysis, the value of M_i shall be defined as:

M_i = resultant range of moment due to a combination of primary plus secondary loads. Loads to be considered include: thermal expansion; anchor movement from any cause; earthquake effects; thrusts from relief and safety valve loads from pressure and flow transients; and other mechanical loads.

(f) The combination of loads shall be specified in the Design Specification. The earthquake loading shall be considered in conjunction with the operating conditions. Weight effects need not be considered in the range loading because they are noncyclic in character. In the combination of loads, all directional moment components in the same direction shall be combined before determining the resultant moment; i.e., resultant moments of loads shall not be combined. If a combination includes earthquake effects, M_i shall be the greater of the resultant range of moment due to the combination of all loads considering one-half the range of the earthquake or the resultant range of moment due to the full range of the earthquake.

O-1320 CLASS 2 OR CLASS 3 PIPING

(a) For Class 2 or Class 3 piping, eq. (9) of NC-3652 or ND-3652 is to be used with M_b to include the reaction force moment. The contribution from the reaction force F to the branch moment, as defined in NC-3652.4 or ND-3652.4, shall be no less than the product, $F \times$ nominal discharge pipe size \times DLF.

(b) Note that the use of eq. (9) of NC-3652 or ND-3652 for branch connections requires a nozzle spacing, as defined by NC-3643.3(c)(6) or ND-3643.3(c)(6).

O-1400 CLOSED DISCHARGE SYSTEMS — OPEN DISCHARGE SYSTEMS WITH LONG DISCHARGE PIPES — SYSTEMS WITH SLUG FLOW

(a) For closed discharge systems, open discharge systems with long discharge pipes, and systems with slug flow, the state of the art does not lend itself to a well defined method of load computation. For these systems the dynamic interaction forces of the total system including the attached discharge piping must be considered.

(b) When a safety valve discharge is connected to a relatively long run of pipe and is suddenly opened, there is a period of transient flow until the steady-state discharge condition is reached. During this transient period, the pressure and flow will not be uniform. When the safety valve is initially opened, the discharge pipe may be filled with air. If the safety valve is on a steam system, the steam discharge from the valve must purge the air from the pipe before steady state steam flow is established and, as the pressure builds up at the valve outlet flange and waves start to travel down the discharge pipe, the pressure wave initially emanating from the valve will steepen as it propagates, and it may steepen into a shock wave before it reaches the exit.

(c) Relief valves discharging into an enclosed piping system create momentary unbalanced forces which act on the piping system during the first few milliseconds following relief valve lift. The pressure waves traveling through the piping system following the rapid opening of the safety valve will cause bending moments in the safety valve discharge piping and throughout the remainder of the piping system. In such a case, the designer must compute the magnitude of the loads and perform appropriate evaluation of their effects.

(d) Particular attention should be given to the large forcing functions acting on the pipe if it contains water seals, two phase flow, or if there is a water column in the discharge piping.

(e) The reaction force effects are dynamic in nature. A time history dynamic solution, incorporating a multi-degree-of-freedom model solved for the transient hydraulic forces is considered to be a preferred method of analysis.

O-1500 DESIGN CONSIDERATIONS

Reference should be made to NB/NC/ND-7000.

It is recommended that the following be included as part of the total design consideration.

(a) Where not required by the Code, it is recommended that the header penetrations for relief valves be in accordance with the nozzle spacing recommendation of NB-3686.1(c).

(b) No more than one penetration should be made around the circumference of the run pipe (i.e., no two penetrations in the same transverse plane), the spacing to be in accordance with the preceding (a).

(c) The stress analysis of the pipe could require additional thickness for membrane protection above that required by the thickness equation for pressure load only.

(d) Detail design should preclude sharp notches that may be generated by the use of saddles, gussets, ribs, etc.

(e) Contoured outlets are often advantageous.

(f) The direction of discharge of several pressure relief valves on the same run pipe should be such as to tend to balance one another for all modes of operation specified in the piping design specification.

(g) Supports may require a detailed analysis to determine their role in restraint as well as support. Considerations should be given to the possibility that, under load, snubbing devices may permit significant deflections.

(h) The reaction force moment arm on the outlet piping should be minimized in accordance with the valve manufacturer's recommendation.

(i) The relief valve outlet piping stack clearance should be checked for interference from thermal expansion, earthquake displacements, etc. The vent stack and valve discharge piping system should be arranged such that pull out of the valve discharge pipe does not occur.

(j) Thermal expansion effects are to be considered as they presently are defined in the Code.

(k) The force due to venting should be included in the evaluation of the stack forces. The effects of back pressure in the discharge stack can be significant.

(l) The station should be arranged such that the discharge piping is void of collected water. The discharge piping from each valve or device should be at least of the same size as the valve outlet.

(m) Drains shall be provided so that condensed leakage, rain, or other water sources will not collect on the discharge side of the valve and adversely affect the reaction force. Safety valves are generally provided with drain plugs that can be used for a drain connection. Discharge piping shall be sloped and provided with adequate drains if low points are unavoidable in the layout.

(n) Where water seals are used ahead of the safety valve, the total water volume in the seals should be minimized. To minimize forces due to slug flow or water seal excursion, the number of changes of direction and the lengths of straight runs of piping should be limited.

(o) Often safety valves are full lift, pop-type valves and are essentially full flow devices with no capability for flow modulation. In actual pressure transients, the steam flow required to prevent overpressure is a varying quantity, from zero to the full rated capacity of the safety valves. As a result, the valves may be required to open and close a number of times during the transient. Since each opening and closing produces a reaction force, consideration should be given to the effects of multiple valve operations on the piping system, including supports.

NONMANDATORY APPENDIX P

ARTICLE P-1000 CERTIFIED MATERIAL TEST REPORTS

P-1100 INTRODUCTION

The requirements for a Certified Material Test Report (CMTR) are stated in NCA-3860. However, the material requirements vary with the class of construction, the product form of material, the requirements of the material specification, and the manufacturer's procedures. Since changes in any of these requirements may be made by Addenda or new editions of the Code, it is important for the purchaser and the supplier to know the requirements of the applicable edition and Addenda, so that the reported results can be compared with the requirements to determine whether or not the material is in compliance with the Code.

P-1200 GENERAL REQUIRED INFORMATION

The items of information given in (a) through (i) below are required in the CMTRs for metallic material (as defined by NCA-1220 and NB/NC/ND/NE/NF/NG-2100) used in Section III, Division 1, construction.

- (a) Name of certifying organization [P-1200(j)].
- (b) Number and expiration date of the organization's Certificate of Authorization or Quality System Certificate (Materials). Alternatively, if the organization was qualified by a party other than the Society, the revision and date of the written program under which the material is being certified.
- (c) Purchaser's order or contract number.
- (d) Description of the material, including specification number, grade, class, type, and nominal size, as applicable. For pipe made to specifications which include both seamless and welded pipe, the report shall designate which type it is.
- (e) Description of material identification marking.
- (f) Actual results of chemical analyses, tests, and examinations required by the Material Specification and this Section.
- (g) Reports of weld repairs performed, if any, as required by NB/NC/ND/NE/NF/NG-2500, including radiographic films, when radiography is required.

(h) Charpy V-notch and drop-weight test results required by NB/NC/ND/NE/NF/NG-2320, when this testing is required by NB/NC/ND/NE/NF/NG-2311. When Charpy V-notch impact tests are required, the report shall include the test temperature; the absorbed energy when required; the lateral expansion; and the location and orientation of the specimens used. When drop-weight tests are required, the report shall include the test temperature; the type, location, and orientation of the specimens used; and the results of the tests (break or no break).

(i) Nondestructive examinations performed and accepted as required by NB/NC/ND/NE/NF/NG-2500.

P-1300 INFORMATION REQUIRED UNDER SPECIFIC CIRCUMSTANCES

The information given in (a) through (f) is required under specific circumstances.

- (a) Heat treatment data, as follows:
 - (1) temperatures (or temperature ranges) and times at temperature used when the material specification requires specific temperatures and times
 - (2) heat treatment conditions when no specific temperatures (or temperature ranges) or times are required by the material specification
 - (3) the minimum solution annealing temperature used for austenitic stainless steels and high nickel alloys
 - (4) recorded temperature ranges and actual times at temperature, and heating and cooling rates, for postweld heat treatment of materials that are repaired by welding, when postweld heat treatment is required by NB/NC/ND/NE/NF/NG-2500
 - (5) recorded temperatures and actual times at temperature for test coupons when required by NB/NC/ND/NE/NF/NG-2210, when those paragraphs are applicable
- (b) Hydrostatic test pressure, when a hydrostatic test is required by the material specification, or notation that the hydrostatic test has not been performed, if it is deferred.
- (c) Ferrite Number for all A-No. 8 welding material except type 16-8-2, as required by NB/NC/ND/NE/NF/NG-2433.

(d) The grain size, reported in accordance with ASTM E112, when a fine grain size is specified for metallic materials.

(e) For welding materials, in addition to applicable paragraphs above, the process, the preheat and interpass temperature, the type of chemical analysis (filler metal/undiluted deposit), the shielding gas composition, and the Ferrite Number shall be reported as required by NB/NC/ND/NE/NF/NG-2400 and as applicable.

(f) A list of chemical analyses, tests, examinations, and heat treatments required by the Material Specification that were not performed.

P-1400 EXECUTION

(a) All requirements of the material specification and this Section need not be performed by the same organization. In that case the certifying organization shall ensure that each organization, providing material or services, is identified on the CMTR along with the activities for which it is responsible. Alternatively, the CMTR's furnished by the other organizations may be referenced on and attached to the CMTR of the organization which supplies the material.

(b) The CMTR shall include a dated statement affirming that the contents of the report are correct and accurate.

(c) Signing or notarization of the CMTR is not required.

NONMANDATORY APPENDIX Q

ARTICLE Q-1000 DESIGN RULES FOR CLAMP CONNECTIONS

Q-1100 INTRODUCTION

Q-1110 SCOPE

(a) The rules in [Nonmandatory Appendix Q](#) apply specifically to the design of clamp connections for pressure vessels and vessel parts and may be used in conjunction with the applicable requirements in Subsections NC, ND, and NE. These rules are not to be used for the determination of the thickness of supported or unsupported tube-sheets integral with a hub nor for the thickness of covers. These rules provide only for hydrostatic end loads and gasket seating and do not consider external loads or thermal effects.

(b) The design of a clamp connection involves the selection of the gasket, bolting, hub, and clamp geometry (see [Figure Q-1130-1](#)). Bolting shall be selected to satisfy the requirements of [Q-1140](#). Connection dimensions shall be such that the stresses in the clamp and hub, calculated in accordance with this Appendix, do not exceed the allowable stresses specified in [Q-1180](#). All calculations shall be made on dimensions in the corroded condition. Calculations for both assembly and operating conditions are required.

(c) It is recommended that either a pressure energized or a low seating load gasket, or both, be used to compensate for possible nonuniformity in the gasket seating force distribution. Hub faces shall be designed so as to have metal-to-metal contact outside the gasket seal diameter. This may be provided by recessing the hub faces or by use of a metal spacer (see [Figure Q-1130-1](#)). The contact area shall be of sufficient cross-sectional area to prevent yielding of either the hub face or spacer under both operating and assembly axial loads.

(d) It is recognized that there are clamp designs which utilize no wedging action during assembly since clamping surfaces are parallel to the hub faces. These designs should satisfy the bolting and corresponding clamp and hub requirements of a clamp connection design with a total included clamping angle of 10 deg. This will provide some safety against loads imposed by angular deflections of the connection faces during operation and also provide some compensation against mechanical and thermal ratcheting.

(e) The rules of this Appendix should not be construed to prohibit the use of other types of clamp connections provided that they are designed in accordance with good engineering practice and the method of design is acceptable to the Inspector. These rules shall apply only to new construction.

(f) Clamps designed to the rules of this Section shall be provided with a bolt retainer. The retainer shall be designed to hold the clamps together independently in case of failure of the primary bolting. An appropriate external yoke or multiple bolting is considered satisfactory for this requirement. Clamp-hub friction shall not be considered as a retainer method.

Q-1120 MATERIALS

(a) Materials used in the construction of clamp connections shall comply with the requirements given in NC, ND, or NE-2000, as applicable.

(b) Hubs and clamps shall not be machined from plate.

Q-1130 NOTATION

The symbols defined below are used in the equations for the design of clamp-type connections (see also [Figures Q-1130-1](#) and [Q-1130-2](#)).

A = outside diameter of hub

A_b = total cross-sectional area of bolts per clamp lug using the root diameter of the thread or least diameter of unthreaded portion, whichever is less. Cross-sectional area of bolt retainer shall not be included in calculation of this area.

A_c = total effective clamp cross-sectional area
 $= A_1 + A_2 + A_3$

A_1 = partial clamp area
 $= (C_w - 2C_t)(C_t)$

A_2 = partial clamp area
 $= 1.571C_t^2$

A_3 = partial clamp area
 $= (C_w - C_g)l_c$

A_{m1} = total cross-sectional area of bolts per clamp lug at root of thread or section of least diameter under stress, required for the operating conditions
 $= W_{m1}/2S_b$

- A_{m2} = total cross-sectional area of bolts per clamp lug at root of thread or section of least diameter under stress, required for gasket seating
 $= W_{m2}/2S_a$
- A_{m3} = total cross-sectional area of bolts per clamp lug at root of thread or section of least diameter under stress, required for assembly conditions
 $= W_{m3}/2S_a$
- A_m = total required cross-sectional area of bolts per clamp lug taken as the greater of A_{m1} , A_{m2} , or A_{m3}
- b = effective gasket or joint-contact surface seating width (see Table XI-3221.1-2)
- b_o = basic gasket or joint-contact surface seating width (see Table XI-3221.1-2)
- B = inside diameter of hub
- B_c = radial distance from connection center line to effective center of bolts
- C = diameter of effective clamp-hub reaction load
 $= (A + C_i)/2$
- C_i = inside diameter of clamp
- C_g = effective clamp gap taken at clamp-hub contact center (C)
- C_t = effective clamp thickness
- C_w = effective clamp width
- e_b = radial distance from effective center of the bolts to the centroid of the clamp body
 $= B_c - (C_i/2) - l_c - X$
- f = hub stress correction factor from Figure XI-3240-6. (This is the ratio of the stress in the small end of the hub to the stress in the large end.) (For values below limit of the Figure, use $f = 1.0$.)
- g_o = thickness of hub neck at small end
- g_1 = thickness of hub neck at intersection with hub shoulder
- g_2 = height of hub shoulder (g_2 shall not be larger than T)
- \bar{g} = radial distance from the hub inside diameter to the hub shoulder ring centroid
 $= \frac{Tg_1^2 + h_2g_2(2g_1 + g_2)}{2(Tg_1 + h_2g_2)}$
- G = diameter at location of gasket load reaction. Except as noted in Figure Q-1130-1, G is defined as follows (see Table XI-3221.1-2):
 - (a) when $b_o \leq 1/4$ in. (6 mm), G = mean diameter of gasket or joint contact face
 - (b) when $b_o > 1/4$ in. (6 mm), G = outside diameter of gasket contact face less $2b$
- h = hub taper length
- h_D = radial distance from effective clamp-hub reaction load to the circle on which H_D acts
 $= [C - (B + g_1)]/2$
- h_G = radial distance from effective clamp-hub reaction load to the circle on which H_G acts (for full face contact geometries $h_G = 0$)
- h_n = hub neck length (minimum length of h_n is $0.5g_1$ or $1/4$ in. (6 mm), whichever is larger)
- $h_o = \sqrt{Bg_o}$
- h_T = radial distance from effective clamp-hub reaction load to the circle on which H_T acts
 $= [C - (B + G)]/2$
- h_2 = average thickness of hub shoulder
 $= T - (g_2 \tan \phi)/2$
- \bar{h} = axial distance from the hub face to the hub shoulder ring centroid
 $= \frac{T^2g_1 + h_2^2g_2}{2(Tg_1 + h_2g_2)}$
- H = total hydrostatic end force
 $= 0.785 G^2P$
- H_D = hydrostatic end force on bore area
 $= 0.785 B^2P$
- H_G = difference between total effective axial clamping preload and the sum of total hydrostatic end force and total joint contact surface compression
 $= [1.571W/\tan(\phi + \mu)] - (H + H_p)$
- H_m = total axial gasket seating requirements for make-up ($3.14bGy$ or the axial seating load for self-energizing gaskets, if significant)
- H_p = total joint-contact surface compression load
 $= 2b \times 3.14GmP$ (for self-energized gaskets, use $H_p = 0$ or actual retaining load if significant)
- H_T = difference between total hydrostatic end force and hydrostatic end force on bore area
 $= H - H_D$
- I_c = effective moment of inertia of clamp relative to axis of entire section
 $= \left(\frac{A_1}{3} + \frac{A_2}{4}\right)C_t^2 + \frac{A_3l_c^2}{3} - A_cX^2$
- I_h = effective moment of inertia of hub shoulder ring relative to its neutral axis
 $= \left(T^2g_1 + h_2^2g_2\right)\frac{\bar{h}}{6} + \frac{Th_2g_1g_2(T - h_2)^2}{3(Tg_1 + h_2g_2)}$
 $= \left[(\text{in.}^3) + (\text{in.}^3)\right]\text{in.} + \frac{(\text{in.}^4)(\text{in.})^2}{[(\text{in.}^2) + (\text{in.}^2)]}$
 $= \text{in.}^4 + \text{in.}^4 = \text{in.}^4$
 $= (\text{mm}^3 + \text{mm}^3)\text{mm} + \frac{(\text{mm}^4)(\text{mm})^2}{[(\text{mm}^2) + (\text{mm}^2)]}$
 $= \text{mm}^4 + \text{mm}^4 = \text{mm}^4$
- L_a = distance from W to the point where the clamp lug joins the clamp body
- L_h = clamp lug height
- L_w = clamp lug width
- l_c = effective clamp lip length
- l_m = effective clamp lip moment arm
 $= l_c - (C - C_i)/2$

m = gasket factor from Table XI-3221.1-1
 M_D = component of moment due to H_D
 $= H_D h_D$
 M_F = offset moment
 $= H_D(g_1 - g_o)/2$
 M_G = component of moment due to H_G
 $= H_G h_G$
 M_H = reaction moment at hub neck
 $= M_O \left\{ 1 + \frac{.818}{\sqrt{Bg_1}} \left[T - \bar{h} + \frac{3.305I_h}{g_1^2(B/2 + \bar{g})} \right] \right\}$
 M_O = total rotational moment on hub (see Q-1150)
 M_P = pressure moment
 $= 3.14 \times PBT(T/2 - \bar{h})$
 M_R = radial clamp equilibrating moment
 $= 1.571 W \left\{ \bar{h} - T + [(C - N)\tan\phi]/2 \right\}$
 M_T = component of moment due to H_T
 $= H_T h_T$
 N = outside diameter of hub neck
 P = Design Pressure
 Q = reaction shear force at hub neck
 $= 1.818 \frac{M_H}{\sqrt{Bg_1}}$
 r = clamp body radius (shall be less than or equal to C_t)
 S_a = allowable bolt stress at atmospheric temperature (see Section II, Part D, Subpart 1, Table 3)
 S_b = allowable bolt stress at Design Temperature (see Section II, Part D, Subpart 1, Table 3)
 S_{OH} = allowable design stress for hub material at (service condition) Design Temperature (see Section II, Part D, Subpart 1, Tables 1A and 1B)
 S_{AH} = allowable design stress for hub material at (assembly condition) atmospheric temperature (see Section II, Part D, Subpart 1, Tables 1A and 1B)
 S_{OC} = allowable design stress for clamp material at (service condition) Design Temperature (see Section II, Part D, Subpart 1, Tables 1A and 1B)
 S_{AC} = allowable design stress for clamp material at (assembly condition) atmospheric temperature (see Section II, Part D, Subpart 1, Tables 1A and 1B)
 S_1 = hub longitudinal stress on outside at hub neck
 S_2 = maximum Lamé hoop stress at bore for hub neck section
 S_3 = hub axial shear stress (maximum) across the hub shoulder
 S_4 = hub radial shear stress (maximum) across the hub neck
 S_5 = clamp longitudinal stress at clamp body inner diameter
 S_6 = clamp tangential stress at clamp body outer diameter
 S_7 = shear stress (maximum) across clamp lips
 S_8 = clamp lug bending stress

S_9 = effective bearing stress between clamp and hub
 T = thickness of hub shoulder for design purposes. The hub shoulder ring is the ring with cross-sectional dimensions T by $(A - B)/2$.
 W = total design bolt load required for service or assembly, as may apply
 W_e = total effective axial clamping preload on one clamp lip and hub shoulder (gasket seating or assembly)
 $= 1.571 W / \tan(\phi + \mu)$
 W_{m1} = minimum required total bolt load for the service conditions [see Q-1140(b)(1)]
 W_{m2} = minimum required total bolt load for gasket seating [see Q-1140(b)(2)]
 W_{m3} = minimum required total bolt load for assembly [see Q-1140(b)(3)]
 X = radial distance from inside surface of clamp body to the centroid of the clamp body
 $= \left[\left(\frac{C_w}{2} - \frac{C_t}{3} \right) C_t^2 - \frac{(C_w - C_g) I_c^2}{2} \right] / A_c$
 y = gasket or joint-contact surface unit seating load (see Table XI-3221.1-1)
 Z = effective clamp-hub taper angle, deg (for gasket seating and preload, $Z = \phi + \mu$; for operating, $Z = \phi - \mu$) [see Q-1140(b)(4)]
 α = hub-neck pipe transition taper, deg (α shall not be greater than 45 deg)
 μ = effective friction angle, deg
 ϕ = clamp-hub taper angle, deg (ϕ shall not exceed 35 deg)

Q-1140 BOLT LOADS

(a) *General.* During assembly of the clamp connection, the design bolt load W is transferred via the clamp-hub taper angle to an axial load (effective clamp preload W_e). In addition, the effect of friction will cause W_e to be reduced for a given W . Friction effects can be reduced by lubrication or by jarring the clamps during assembly. An appropriate friction angle shall be established for both assembly and operating conditions.

(b) *Calculations.* In the design of the bolting for a clamp connection, complete calculations shall be made for three separate and independent sets of conditions which are defined as follows.

(1) The required bolt load for the service conditions W_{m1} shall be sufficient to:

(-a) resist the hydrostatic end force H exerted by the maximum allowable working pressure on the area bounded by the diameter of gasket reaction; and

(-b) maintain on the gasket or joint-contact surface, a compression load H_p which experience has shown to be sufficient to ensure a tight joint.

The minimum operating bolt load W_{m1} shall be determined in accordance with eq. (1).

Figure Q-1130-1
Typical Hub and Clamp

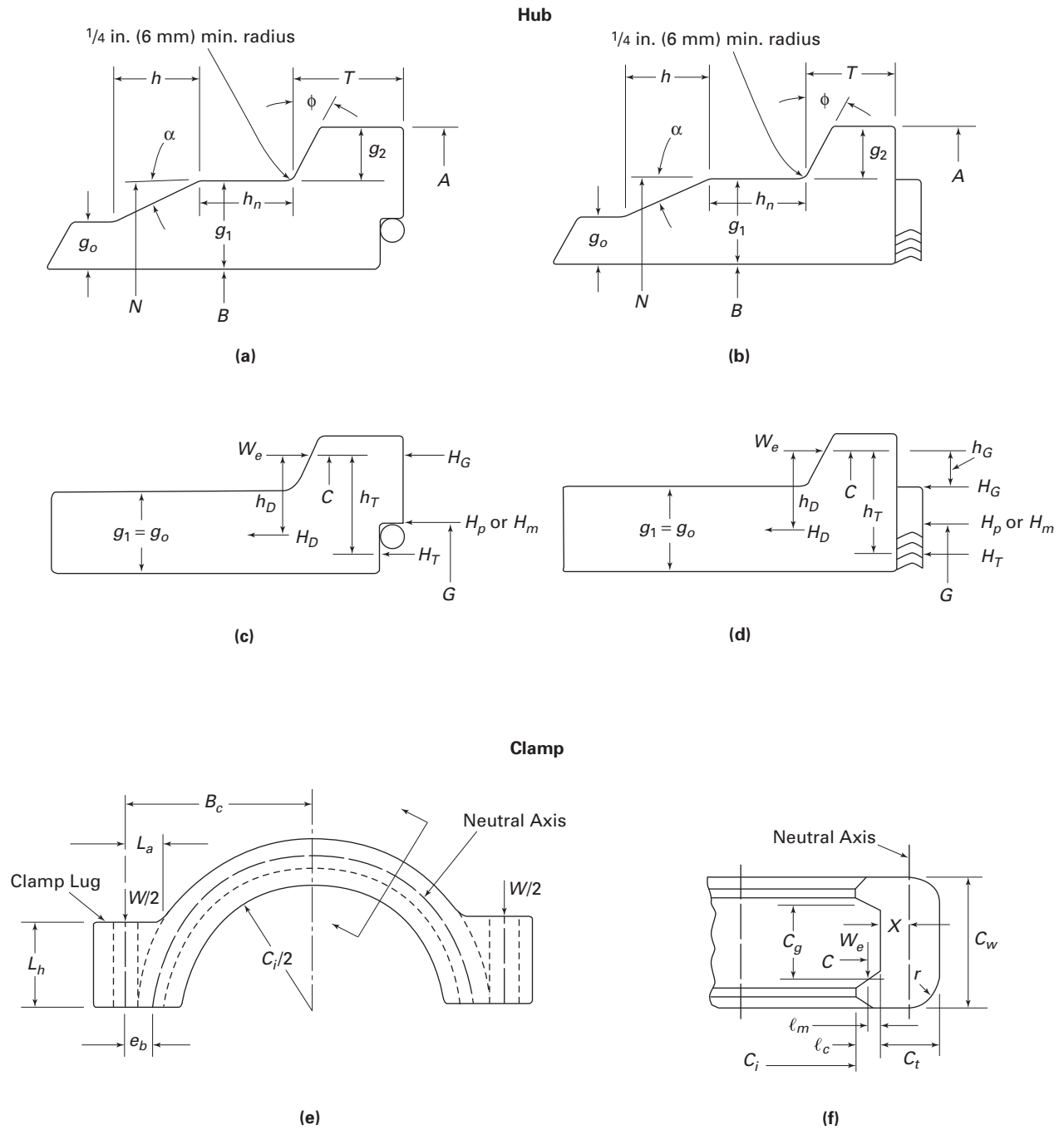
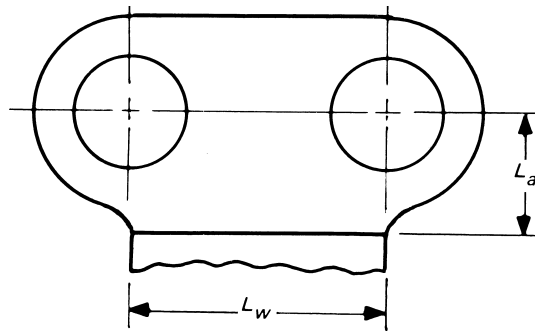
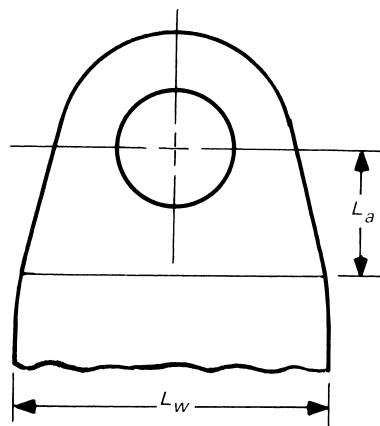


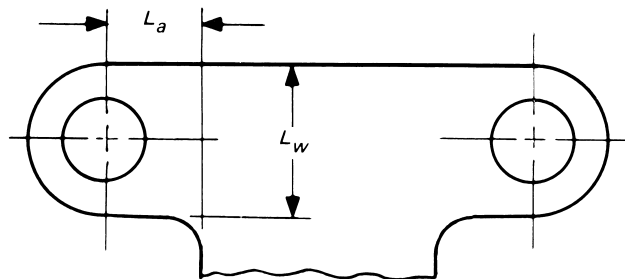
Figure Q-1130-2
Typical Clamp Lug Configurations



(a)



(b)



(c)

$$W_{m1} = 0.637 (H + H_p) \tan(\phi - \mu) \quad (1)$$

(2) Before a tight joint can be obtained, it is necessary to seat the gasket or joint-contact surface properly by applying a minimum initial load (under atmospheric temperature conditions without the presence of internal pressure), which is a function of the gasket material and the effective gasket area to be seated. The minimum initial bolt load required for gasket seating W_{m2} shall be determined in accordance with eq. (2).

$$W_{m2} = 0.637 H_m \tan(\phi + \mu) \quad (2)$$

(3) To ensure proper preloading of the clamp connection against service conditions, an assembly bolt load W_{m3} shall be determined in accordance with eq. (3).

$$W_{m3} = 0.637 (H + H_p) \tan(\phi + \mu) \quad (3)$$

(4) In eq. (1)(1) credit for friction is allowed based on clamp connection geometry and experience, but shall be limited to a value in which $(\phi - \mu)$ is equal to or greater than 5 deg. In eqs. (2)(2) and (3)(3), friction shall be considered and be such that μ is equal to or greater than 5 deg. This will then satisfy the requirements of Q-1110(d).

(5) The need for providing sufficient bolt load for either gasket seating in accordance with eq. (2)(2) or assembly in accordance with eq. (3)(3) will prevail on many low pressure designs and with facings and materials that require a high seating load where the service bolt load computed by eq. (1)(1) is insufficient to properly preload the connection.

(c) *Required Bolt Area.* The total cross-sectional area of bolting A_m required shall be the greater of the values for service conditions A_{m1} , gasket seating conditions A_{m2} , or assembly condition A_{m3} . Bending of the bolting due to nonparallel nut bearing surfaces shall be compensated for by use of a stress correction factor in bolt area calculations or by use of spherically seated nuts and/or washers.

(d) *Clamp Connection Design Bolt Load W .* The bolt load used in the design of the clamp connection shall be the values obtained from eqs. (4) and (5).

For service conditions:

$$W = W_{m1} \quad (4)$$

For assembly conditions:

$$W = (A_m + A_b) S_a \quad (5)$$

Q-1150 HUB MOMENTS

In the calculation of hub stresses, the moment of a load acting on the hub is the product of the load and its moment arm. The moment arm is determined by the relative position of the effective hub-clamp reaction diameter with respect to that of the load producing the moment [see

Figure Q-1130-1, sketches (c) and (d)]. In addition to the load moments, additional reaction moments with relation to the hub shoulder ring centroid are considered to compensate for hub-to-pipe transition, radial pressure, and clamp radial equilibrating effects.

For the service conditions, the rotational hub moment M_O is the sum of six individual moments M_D , M_G , M_T , M_F , M_P , and M_R based on the design bolt load of eq. Q-1140(d)(4) with moment arms as given in Figure Q-1130-1.

For assembly, the rotational hub moment M_O is based on the design bolt load of eq. Q-1140(d)(5) in which case

$$M_O = \frac{0.785 W(C - G)}{\tan(\phi + \mu)} \quad (6)$$

Q-1160 CALCULATION OF HUB STRESSES

The stresses in the hub shall be determined for both the service and the assembly conditions.

(a) The reaction moment M_H and reaction shear Q as defined in Q-1130 shall be calculated at the hub neck for rotational moment M_O .

(b) Hub stresses are to be calculated from the following equations:

Hub longitudinal stress

$$S_1 = f \left[\frac{PB^2}{4g_1(B + g_1)} + \frac{1.91 M_H}{g_1^2(B + g_1)} \right] \quad (7)$$

Hub hoop stress

$$S_2 = P \left(\frac{N^2 + B^2}{N^2 - B^2} \right) \quad (8)$$

Hub axial shear stress

$$S_3 = \frac{0.75 W}{T(B + 2g_1) \tan Z} \quad (9)$$

Hub radial shear stress

$$S_4 = \frac{0.477 Q}{g_1(B + g_1)} \quad (10)$$

Q-1170 CALCULATION OF CLAMP STRESSES

The stresses in the clamp shall be determined for both the service and the assembly conditions. Clamp stresses are to be calculated from the following equations:

Clamp longitudinal stress

$$S_5 = \frac{W}{2C(\tan Z)} \left[\frac{1}{C_t} + \frac{3(C_t + 2l_m)}{C_t^2} \right] \quad (11)$$

Clamp tangential stress

$$S_6 = \frac{W}{2} \left[\frac{1}{A_c} + \frac{|e_b| (C_t - X)}{I_c} \right] \quad (12)$$

Clamp lip shear stress

$$S_7 = \frac{1.5 W}{(C_w - C_g) C \tan Z} \quad (13)$$

Clamp lug bending stress

$$S_8 = 3 W \frac{L_a}{L_w L_h^2} \quad (14)$$

In addition, a bearing stress calculation is to be made from eq. (15) for either the clamp or hub.

$$S_9 = \frac{W}{(A - C_i) C \tan Z} \quad (15)$$

Table Q-1180-1
Allowable Design Stress for Clamp Connections

Stress Category	Allowable Stress
S_1	$1.5S_{OH}$ or $1.5S_{AH}$
S_2	S_{OH}
S_3	$0.8S_{OH}$ or $0.8S_{AH}$
S_4	$0.8S_{OH}$ or $0.8S_{AH}$
S_5	$1.5S_{OC}$ or $1.5S_{AC}$
S_6	$1.5S_{OC}$ or $1.5S_{AC}$
S_7	$0.8S_{OC}$ or $0.8S_{AC}$
S_8	S_{OC} or S_{AC}
S_9	[Note (1)]

NOTE:

- (1) 1.6 times the lower of the allowable stresses for hub material (S_{OH} , S_{AH}) and clamp material (S_{OC} , S_{AC}).

Q-1180 ALLOWABLE DESIGN STRESS FOR CLAMP CONNECTIONS

Table Q-1180-1 gives the allowable stresses that are to be used with the equations of Q-1160 and Q-1170.

NONMANDATORY APPENDIX R

ARTICLE R-1000 DETERMINATION OF PERMISSIBLE LOWEST SERVICE METAL TEMPERATURE FROM T_{NDT} FOR CLASSES 2 AND MC CONSTRUCTION

R-1100 INTRODUCTION

R-1110 SCOPE

These rules provide the method for determining the permissible lowest service metal temperatures for materials having nil-ductility transition temperatures (T_{NDT}) determined in accordance with NC- or NE-2311(a)(8) or NC- or NE-2331(a)(2).

R-1200 DETERMINATION OF PERMISSIBLE LOWEST SERVICE METAL TEMPERATURE

The permissible lowest service metal temperature is defined as:

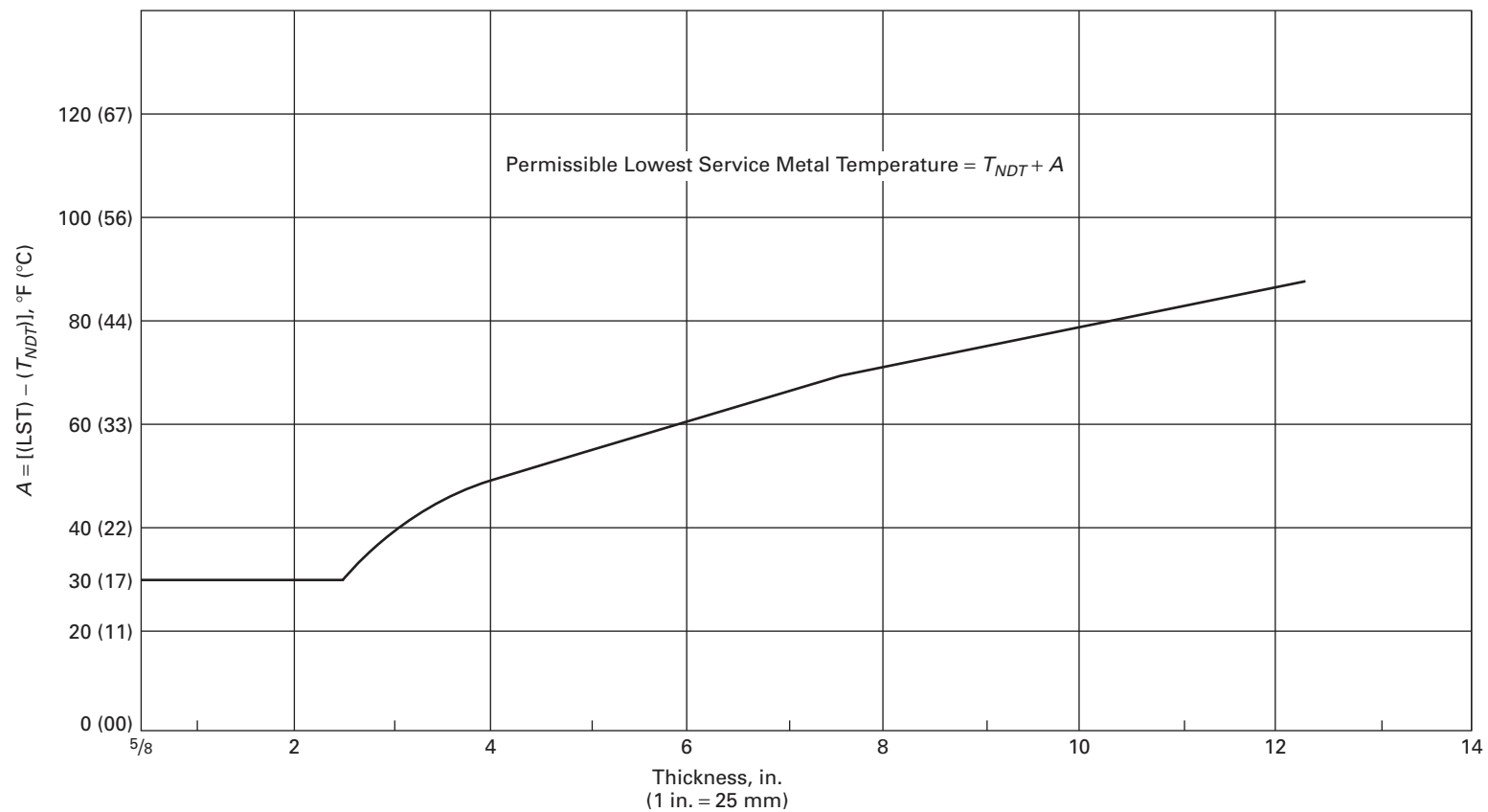
$$T_{NDT} + A$$

where T_{NDT} is determined in accordance with NC- or NE-2311(a)(8) or NC- or NE-2331(a)(2), and A is determined from [Figure R-1200-1](#) for the thickness of the material.

R-1210 MATERIAL ACCEPTABILITY

For the material to be acceptable, the permissible lowest service metal temperature shall not be higher than the specified lowest service metal temperature.

Figure R-1200-1
Determination of Permissible Lowest Service Metal Temperature



NONMANDATORY APPENDIX S

ARTICLE S-1000 PUMP SHAFT DESIGN METHODS

S-1100 INTRODUCTION

(a) This Appendix provides guidelines for the design and evaluation of shafts for Section III nuclear pumps. They include suggestions for the loads to be considered and the method for arriving at a design that will sustain these loads for the life of the pump.

(b) The method presented is not intended to exclude other methods which may be equally satisfactory. Experience and expertise will be factors in determining the best method. In the final analysis the skill and judgement of the designer determine the quality of the design.

S-1200 SCOPE

The guidelines presented are intended to cover the design of pump shafts, including rigid couplings, up to, but not including, separable parts which attach the shaft to the driver.

S-1300 DESIGN REQUIREMENTS

S-1310 DESIGN METHODS

The design rules presented in this section are provided as guidance to the designer of pump shafts. Alternative design methods may be used based on the pump manufacturer's experience with pumps in similar service.

S-1320 OPERATING CONDITIONS

Pump shaft assemblies are subject to combinations of steady state and variable or transient loads. These loads include torsional, lateral, bending, axial, and thermal components. They may occur as a result of power input, hydraulically imposed forces, static or dynamic unbalance, rotating element runout, internal misalignment, thermal distortion, system and component vibration, and resonance. External system applied loads, such as seismic and other loads associated with A, B, C, and D Service Level Loadings must also be considered.

S-1400 RESPONSIBILITY

(a) *Owner's Responsibility.* It shall be the responsibility of the Owner or his designee to include in the Design Specification all external forces and operating conditions that may have an effect on the operability of the pump shaft.

(b) *Pump Designer Responsibility.* It is the responsibility of the pump Designer to include in the evaluation the specified pump loads and identify the type and magnitude of the internal operating loads on the shaft assembly.

S-1500 OPERATING LOADS

S-1501 MAXIMUM SHAFT OPERATING LOADS

In establishing maximum shaft operating loads, the design should take into account plant service that the pump will experience and external loads associated with these conditions. Other forms of off-normal operating loadings may include inservice testing, inadvertent starting and stopping, loss of coolant accident, etc.

S-1502 THERMAL LOADS

Thermal loads shall be considered as one of the internal operating loads.

S-1503 OFF-NORMAL OPERATING LOADS

The maximum steady state and transient loads will usually occur in the pump when it is operating away from its best efficiency point. Typically it will be the combination of loads producing stresses at regions of high stress concentration that may result in high cycle fatigue failures of shafts. The very nature of these loads makes it difficult to quantify them and in some cases a bounding estimate must be made. Examples of load sources include low flow recirculation, flow separation, and other hydraulic instabilities which cause radial and axial alternating or transient loads on the shaft. Lacking a thorough understanding of these loads, conservative design practices based on years of operating experience must be used to insure successful design of pump shafts.

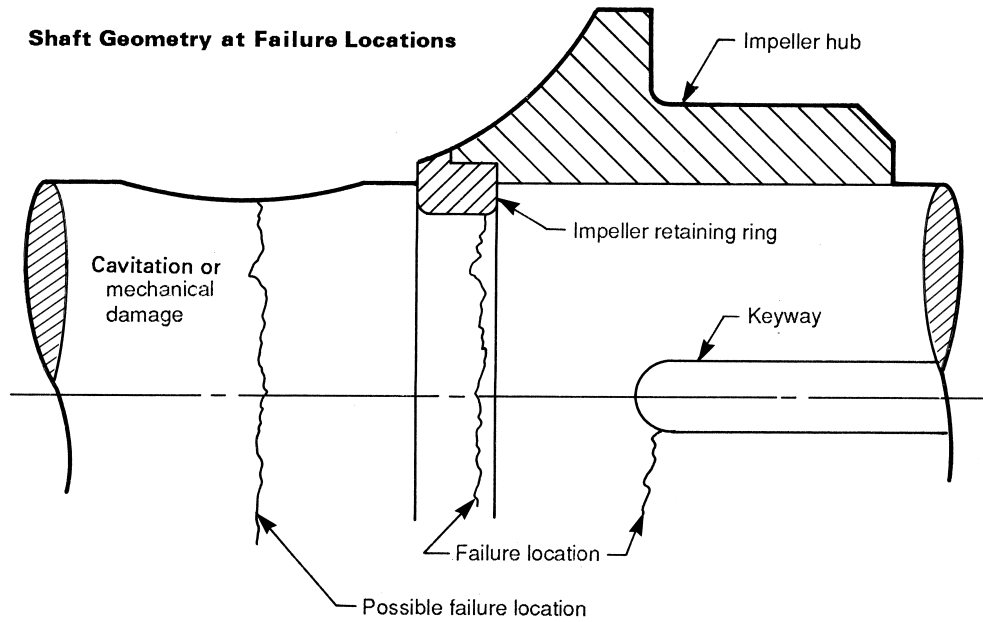
S-1600 SHAFT FAILURE MODES

Shaft failures usually occur at points of high stress concentration or structural discontinuities. The most common locations of shaft failures are threaded regions, shaft grooves, shoulders, keyways, couplings, and collars (see

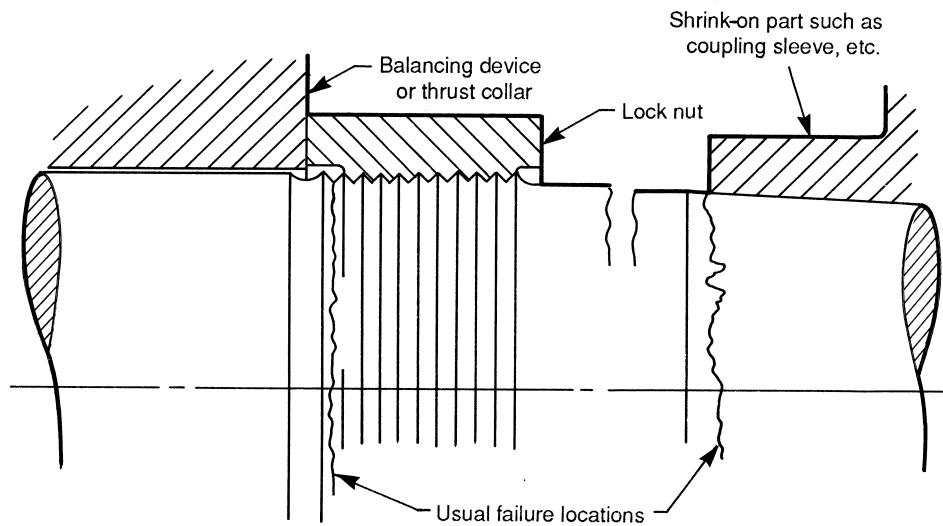
[Figure S-1600-1](#)). Areas susceptible to erosion/corrosion, stress corrosion cracking, thermal transients, and steep temperature gradients are also possible locations for shaft failure.

Figure S-1600-1

Figure S-1600-1
Typical Centrifugal Pump Shaft Failure Locations



(a) Shaft Groove or Interstage External Damages



(b) Threaded or Shrunk-on Parts

ARTICLE S-2000 DESIGN PROCEDURE

Any method of evaluation (analytical or experimental) which can be substantiated by data from pumps in service, experiencing conditions similar to the specified operating limits, may be used.

S-2100 CRITICAL SPEEDS

The evaluation shall address both torsional and lateral, and where applicable, axial critical speeds, shaft deflections, and stresses. Critical speeds and shaft deflections shall be such as to avoid any difficulties for the specified range of the design and operating conditions. The actual percentage difference between critical and operating speeds shall take account of the method of determination of critical speed. The percentage difference between stress allowed and calculated shall also take account of the accuracy of the design and the analysis method.

S-2200 MAXIMUM TORSIONAL LOAD

The maximum torsional load shall be defined. The maximum torsional shear stress for this load (stress resulting from this load without application of concentration factors) shall be based on design experience or experimental evidence for the particular class of pump involved. The maximum driver horsepower may determine the maximum torsional loading for units with short shafts (typical of pump types A, B, and C). The motor startup torque may determine the maximum torsional load for pumps with relatively long shafts (typical of type L pumps). Torsional alternating or transient loads shall be considered if applicable.

S-2300 SHAFT EVALUATION

The flow chart, [Figure S-2300-1](#), outlines a procedure for evaluation of pump shafts to meet load requirements. This procedure establishes a basic sizing criterion as well as a detailed fatigue evaluation method.

The basic shaft sizing criterion is based on maximum shear stress and conservative cyclic loading factors. These fatigue factors include an evaluation of the endurance limit of the unnotched and polished test specimen reverse bending test data in air (S_e), which in the absence of specified data can be approximated as:

$$S_e = 0.5S_u$$

This polished specimen test endurance limit is then factored for the product of reduction factors that account for such items as environment, reliability, size, finish, duty cycles, etc., which is conservatively estimated as one-third. Consequently, this corrected material endurance limit stress can be represented as:

$$S'_e = \frac{1}{K_t} \cdot \frac{S_e}{3}$$

where the terms used on the chart, [Figure S-2300-1](#), are as follows:

K_t = stress concentration factors. An initial value of 6.0 is suggested where reasonable stress riser control is exercised. Higher values may be required for designs with severe discontinuities (e.g., small fillet radii relative to shaft diameter). Lower values may be used if justified by design methods and/or testing that accounts for the specific shaft discontinuities under consideration. Notch sensitivity values, when available, may be used. For additional information, see ANSI/ASME B106.1M-1985.

S_a = alternating axial stress

S_b = alternating component of the shaft bending stress

S_e = material endurance limit

S'_e = design endurance limit

S_s = maximum shear stress

S_{ss} = material allowable shear stress

= $S_y/3$ unless higher values are justified through specified shear test data

S_u = material ultimate strength at design temperature

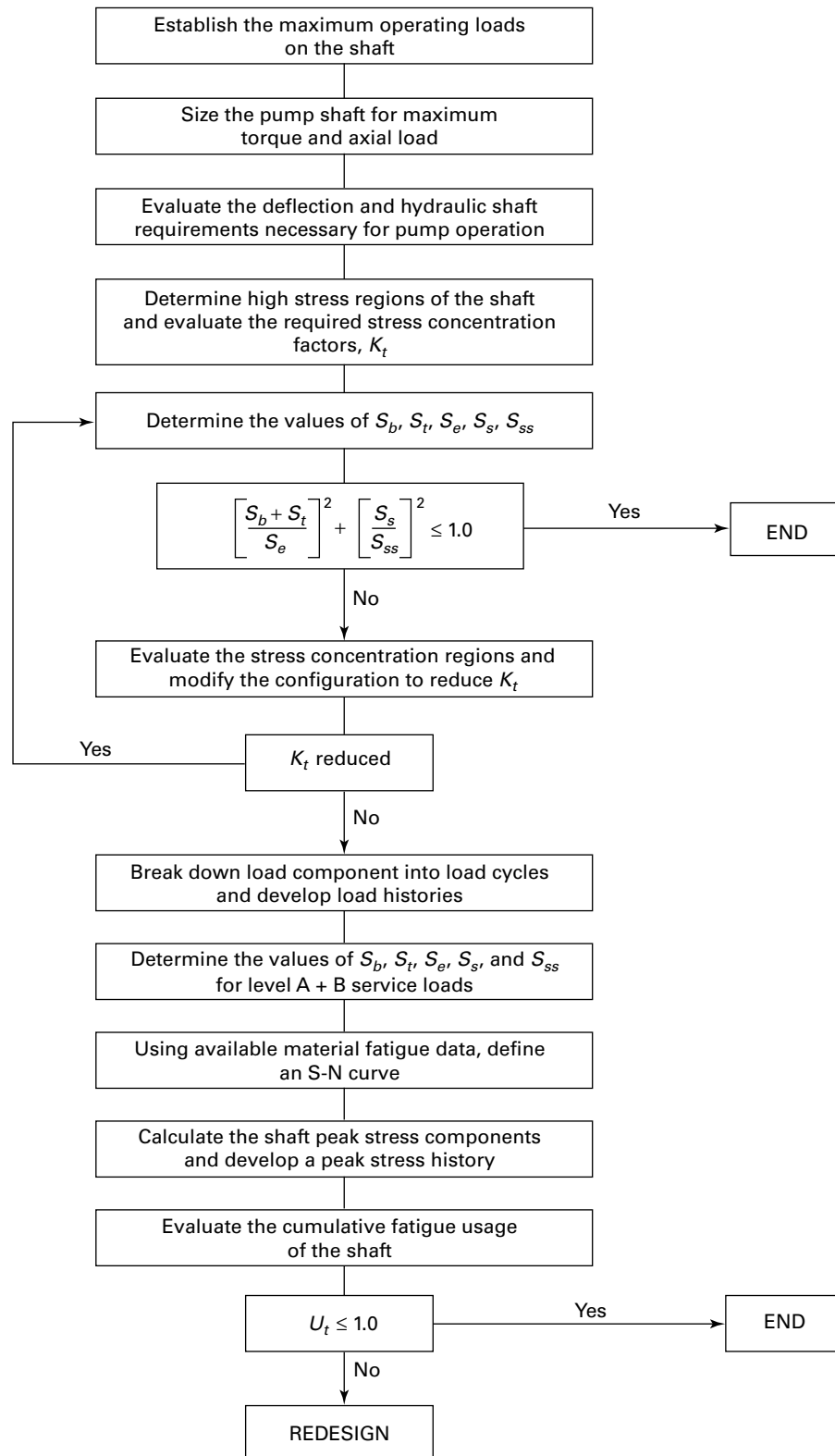
S_y = material yield stress at design temperature

U_t = summation of usage cycles. Each usage cycle shall be determined as the ratio of the maximum alternating stress in the shaft divided by the design fatigue stress of the material for that number of cycles. U_t will be determined by summing all of the above ratios.

S-2400 OTHER CONSIDERATIONS

The fatigue life of a shaft is not always the limiting factor in its design. The effect of misalignment and deflection of a shaft on the performance of support bearings, seals, and couplings as well as on other key power transmission components must also be taken into account. Shafts can be strong enough to meet fatigue life requirements, yet not stiff enough to satisfy natural frequency and operational requirements.

Figure S-2300-1
Steps in the Design of a Pump Shaft



NONMANDATORY APPENDIX T

ARTICLE T-1000 RECOMMENDED TOLERANCES FOR RECONCILIATION OF PIPING SYSTEMS

T-1100 INTRODUCTION

The building structure and major components of a power plant are constructed according to rules that permit varied tolerances. Since piping system installation follows construction of the building and installation of the major components, the piping systems must be permitted to vary within the space allotted to them. In addition, a large number of systems are often installed in a limited space. Interferences often occur and changes within Installation Tolerances may be used to eliminate the interference. The tolerances provided in this Appendix bridge the gap between the exactness associated with a design by analysis, and a practical and acceptable installation.

The basis for the tolerances and guidance in this Appendix was developed by the PVRC Technical Committee on Piping Systems.³⁷ Additional guidance on implementation of these tolerances has been published by EPRI.³⁸

(i.e., seismic time history analysis methods), the Designer shall review the applicability of these tolerances and establish more stringent guidelines if necessary. Further, this Appendix shall be restricted to piping systems analyzed using linear elastic methods.

This Appendix does not relieve the Designer of responsibility for consideration of other unique situations where more restrictive tolerances may be required to satisfy the intent of the design bases or the Code.

Installation Tolerances more restrictive than the Total Tolerances recommended in this Appendix may be specified. Less restrictive tolerances may be specified when engineering justification is provided to demonstrate that the design requirements have been satisfied.

Tolerances for complete, installed piping systems are addressed. Tolerances provided for manufacturing or fabricating the individual items or subassemblies that make up piping systems are not addressed, but the effect of these tolerances on the as-installed condition shall be within the Total Tolerance. Other design and construction areas which may be included in reconciliation such as design or operating conditions, support details, and gaps are not addressed.

Tolerances for support erection, including length and orientation of individual members and pipe location on the support, are specified in Subsection NF, Appendix NF-D, Tolerances.

T-1120 TERMS RELATED TO RECONCILIATION

Definitions of terms used in this Appendix are given in the following paragraphs.

T-1121 Nominal Dimension

This is the dimension which provides configuration and/or spatial information on piping drawings within specified tolerances.

T-1122 As-Analyzed Configuration

This is the configuration of piping components and supports, defined by nominal sizes, weights, cross section properties, and dimensions, which forms the basis for

(13) T-1110 SCOPE

This Appendix provides recommended tolerances and methods for satisfying the requirements of NCA-3554, "Modification of Documents and Reconciliation With Design Report" for piping systems designed to the rules of NB-, NC-, or ND-3600. This Appendix provides:

- (a) identification of dimensions and weights significant to the piping stress analysis; and
- (b) acceptable tolerances for these dimensions and weights such that if piping is installed within these tolerances, the reconciliation is accomplished.

These tolerances have been established such that their effect on the accuracy of analysis results is minimal and is consistent with accepted practices and the use of tolerances in the Code. These tolerances are applicable to most situations; however, specific situations where more restrictive tolerances may be needed are identified.

The tolerances in this Appendix are applicable to piping systems where conventional seismic analysis methods were used for the original design, i.e., modal response spectrum analysis methods. For piping systems where seismic analysis methods that are significantly more sensitive to the tolerances were used in the original design

the piping stress analysis. In the case of piping systems which are qualified by simplified rules, the design drawings for the systems are considered as the As-Analyzed Configuration.

T-1123 Critical-to-Design (CTD) Dimension

A dimension that must be satisfied, within a specified tolerance, in order for the piping stress analysis to remain valid. These dimensions define the relative configuration of the piping. They may also include dimensions which define the global or spatial position of the piping.

Examples of CTD dimensions are: the location of a pipe support relative to in-line pipe components such as valves, anchors, and other supports; the orientation of the pipe support center line relative to the pipe center line; length of pipe runs; and spacing between supports.

T-1124 Fit/Clearance (F/C) Dimension

This is a dimension that is specified to provide reasonable assurance that the piping fits into the allocated building space. These dimensions define the global position of the piping in three-dimensional space and deviations from these dimensions do not have a significant effect on the validity of the piping stress analysis. Examples of F/C dimensions are the locating dimensions of pipe center lines from column lines, walls, and floors. However, dimensions defining clearance required to allow thermal expansion or to protect critical components from adverse interaction due to dynamic (earthquake) loads are CTD dimensions.

T-1125 As-Built Documents

These are the drawings, sketches, or other documents which define the as-installed piping configuration (nominal dimensions and tolerances) and which have been reconciled with the stress analysis.

T-1126 Installation Tolerance

This is the specified acceptable departure from nominal piping dimensions, support locations and orientations, and component weights to be used during installation to provide for practical installation limitations. Installation Tolerances for CTD dimensions shall be less than or equal to the corresponding Total Tolerances.

T-1127 Total Tolerance

This is the maximum allowable departure between corresponding as-built and as-analyzed CTD piping dimensions, support locations and orientations, and component weights for the piping stress analysis to be applicable. The Total Tolerance is applicable only to CTD dimensions since F/C dimensions are not critical to the stress analysis. Acceptable Total Tolerances are provided in [T-1200](#).

T-1130 MEASUREMENT ACCURACY

Measurements of the as-built configuration for use in reconciliation shall be made using methods capable of producing accuracy to the nearest inch for linear dimensions and to the nearest 2 deg. for angular dimensions ([Figure T-1213-1](#) shows examples of angular dimensions).

T-1140 EVALUATION OF OUT-OF-TOTAL TOLERANCE CONDITIONS

CTD values that exceed Total Tolerances shall be recorded and evaluated to assure the design bases, including the applicable design code, have been satisfied. This evaluation may be made using engineering judgment, simplified models, or by reanalyzing the complete original model. The objective should be to determine if the condition being evaluated had any significant effect on the response of the piping systems to the design loadings. The use of engineering judgment shall be documented with the technical reasoning described so that a technically qualified third party reviewer will understand how the evaluation was justified. The documentation requirements for evaluation based on engineering judgment shall be consistent with other engineering calculations.

If it is determined that there is no significant effect, then the original analysis remains valid and the results (stress summaries, support design loads, etc.) need not be revised. If changes are found to be significant, then the affected results shall be revised.

T-1200 TOTAL TOLERANCES

T-1210 GENERAL

The Total Tolerances are given for the reconciliation of the piping stress analysis in terms of the maximum allowable plus or minus departure between the as-analyzed (nominal) value and the as-built value. These tolerances should not be used to circumvent specific requirements established by the Design or Installation Specification nor other applicable material or fabrication standards.

As-built piping systems that conform with the piping stress analysis within the Total Tolerances are acceptable and evaluation of the as-built condition is not required.

T-1211 Altering Relative Position

Tolerances shall not be applied in such a manner that would result in the alteration of the relative position of piping components and supports. For example, supports shall not be moved across valves, tees, elbows, etc. In no case shall the permissible location change the directions of a support or its function.

T-1212 Independence

The tolerances given are independent of each other and are not interrelated to position of adjacent items except as noted in [T-1211](#). Each tolerance may be applied independently of any other.

T-1213 Methods of Dimensioning

Examples of angular dimensions are shown in [Figure T-1213-1](#). Two common methods of dimensioning piping installation are shown in [Figure T-1213-2](#). Either method or some combination of these methods are usually used on the Design Drawings. The chain method references a dimension to the adjacent dimensions. The common point method references a number of dimensions to a common base. All measurements should be rounded off to the nearest inch for linear dimensions and the nearest 2 deg for angular dimensions.

T-1220 PIPING CONFIGURATION TOLERANCES**(13) T-1221 Center Line Lengths**

Center line length to fittings, flanges, valves, and piping specialties including branch line piping shall be within the tolerance specified below. (Locations of branch connections on the run pipe are not included. see [T-1222](#).)

Specified Nominal Dimension, ft (m)	± Total Tolerance, in. (mm)
0 to < 5 (0 to < 1.5)	3 (75)
5 to < 10 (1.5 to < 3.0)	6 (150)
10 to < 15 (3.0 to < 4.5)	9 (225)
15 to < 20 (4.5 to < 6.0)	12 (300)
20 to < 25 (6.0 to < 7.5)	15 (375)
25 to < 30 (7.5 to < 9.0)	18 (450)
30 to < 35 (9.0 to < 10.5)	21 (525)
35 and over (10.5 and over)	24 (600)

NOTE: The 3 in. (75 mm) tolerance for nominal dimensions of 0 to less than 5 ft (1.5 m) may not be applicable in all cases. For example, cantilevered vents and drains, relief valve inlet and outlet piping, and instrument piping near the connection to the process piping are cases where more restrictive tolerances may be needed. If so, they shall be specified on the design documents.

When common point dimensions (see [Figure T-1213-2](#)) are used, the tolerances along the pipe center line are to be applied to each pipe leg, i.e., piping between changes in direction. There may be cases where the nominal dimensions must be determined from the difference in common point dimensions for the ends of the pipe leg.

T-1222 Center Line Location of Branch Connections (Relative to Run)

(a) This section applies to branch connections independent of the connection detail or type of fitting used.

(b) The location of branches for branch/run size combinations indicated by an asterisk (*) in [Table T-1222-1](#) have no design related tolerance restrictions except when the stress intensification factor (SIF) on the run pipe exceeds 1.0, in which case the maximum tolerance is ±2 ft 0 in. (600 mm) and the requirements of (d) shall be satisfied.

(c) The location of branches for the branch/run size combinations without an asterisk (*) in [Table T-1222-1](#) (including sizes not listed in the table) have the same tolerance as in [T-1221](#).

(d) For all branch/run size combinations, changes in location of the branch connection that exceed 2 ft (600 mm) shall be approved by the Designer and should not

(1) change the flow path defined on the piping system diagram.

(2) affect system functional flow and pressure characteristics, such as branch connections used for differential pressure and flow measurement, for safety and relief valves, and for core safety injection.

(3) be allowed without considering the effect of run pipe movement on the branch pipe. (Not significant if the tolerances of [T-1221](#) are satisfied.)

T-1223 Angular Deviation of Pipe Center Line

Angular deviation of pipe center line from theoretical center line is limited to ±10 deg. see [Figure T-1213-1](#), Rolled Elbows ("B°") and Bends; Nonstandard Elbows ("C°").

T-1224 Angular Tolerance on Power Operated Valves

Angular tolerance on power operated valves over NPS 2 (DN 50) (Nominal Pipe Size) is limited to ±15 deg. See [Figure T-1213-1](#), Valves ("D°"). For power operated valves NPS 2 (DN 50) and smaller, the angular tolerance of ±15 deg applies only when the weight of the valve operator equals or is less than the valve analyzed weight. When the weight of the valve operator exceeds the valve's analyzed weight for an NPS 2 (DN 50) or smaller valve, the angular tolerance is not provided in this Appendix. There are no CTD restrictions on angular orientation of manually operated valves.

T-1230 PIPING SUPPORT LOCATION/ ORIENTATION TOLERANCES

The tolerance provided as piping support tolerances assume the support is the specified type and rating. The support type shall not be changed without approval of the Designer.

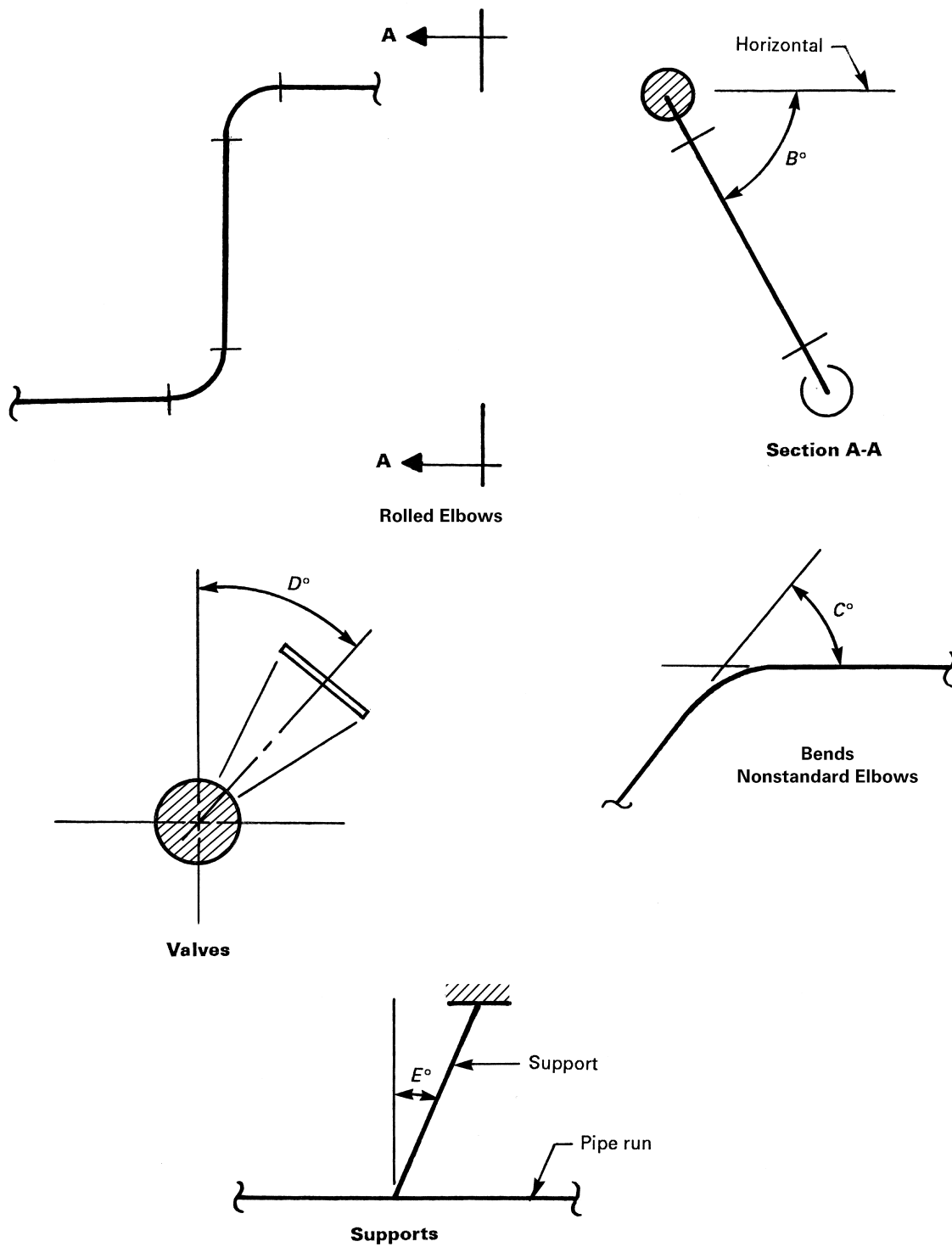
These tolerances are applicable to the reconciliation of the piping stress analysis. They are not applicable for the reconciliation of the pipe support design.

T-1231 Location of Supports

Tolerances on the location of supports, anchors, and restraints along the pipe center line of the horizontal or vertical straight runs that do not contain significant concentrated weight are

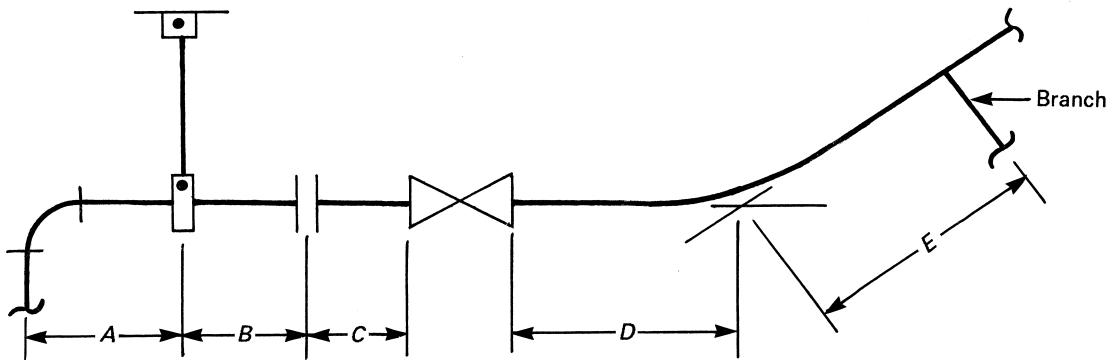
Pipe Size	Tolerance
NPS 2 (DN 50) and smaller	6 in. (150 mm)
NPS 2½ (DN 65) and larger	One pipe diameter or 12 in. (300 mm), whichever is greater

Figure T-1213-1
Illustrations of Angular Dimensions — Pipe Legs, Valves, Supports, Bends

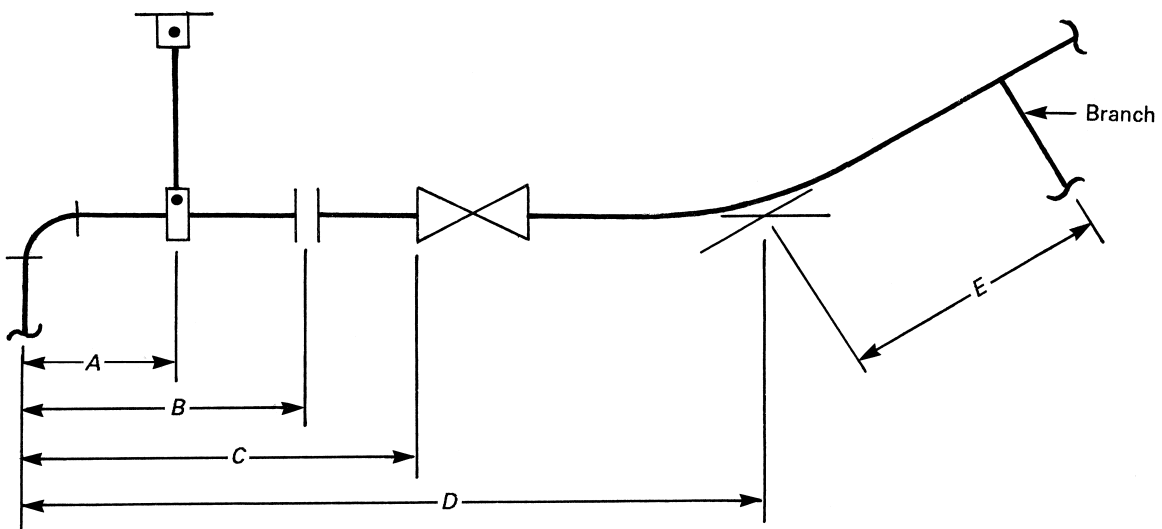


GENERAL NOTE:
 B° , C° , D° , and E° indicate angular dimensions.

Figure T-1213-2
Illustrations of Linear Dimensions



Method 1 — Chain Dimensions



Method 2 — Common Point Dimensions

Table T-1222-1
Branch/Run Size Combinations

Run Size — NPS (DN)	$\frac{3}{4}$ (20)						
	1 (25)						
	1½ (40)						
	2 (50)						
	2½ (65)	*					
	3 (80)	*	*				
	4 (100)	*	*				
	6 (150)	*	*	*	*		
	8 (200)	*	*	*	*	*	
	10 (250)	*	*	*	*	*	*
	12 (300)	*	*	*	*	*	*
	14 (350)	*	*	*	*	*	*
	16 (400)	*	*	*	*	*	*
	18 (450)	*	*	*	*	*	*
	20 (500)	*	*	*	*	*	*
	24 (600)	*	*	*	*	*	*
	$\frac{3}{4}$ (20)	1 (25)	1½ (40)	2 (50)	2½ (65)	3 (80)	4 (100)
	Branch Size — NPS (DN)						

T-1232 Location of the First Support on Either Side of Spans That Contain Concentrated Weights

Tolerances on the location of the first support or restraint on either side of spans that contain concentrated weights such as valves, flanges, risers, fittings, bends, or other concentrated loads are

Pipe Size	Tolerance
NPS 12 (DN 300) and smaller	Smaller of 3 pipe diameters or 12 in. (300 mm)
Larger than NPS 12 (DN 300)	One pipe diameter

These tolerances may not be applicable to the first restraint adjacent to a bend, which is oriented in the plane of the bend, when the piping is subjected to waterhammer, steamhammer, or relief valve discharge loading. In these cases, if more restrictive tolerances are needed they must be specified on the design documents.

T-1233 Location of the First Support From Rotating Equipment Nozzles

Tolerances for the location of the first support or restraint in each direction from rotating equipment nozzles are

Pipe Size	Tolerance
NPS 2 (DN 50) and smaller	3 in. (75 mm)
NPS 2½ (DN 65) and larger	One-half pipe diameter or 6 in. (150 mm), whichever is greater

T-1234 Snubbers

The location of snubbers with axis coincident to the pipe leg center line is not a critical dimension. However, snubbers with tolerances greater than those of T-1231 should be approved by the Designer who must assure they are not affected by the anticipated thermal movements.

T-1235 Angular Orientation of Supports

The tolerance for the angular orientation of supports is ± 5 deg [see Figure T-1213-1, Supports ("E")]. The angular tolerance may be increased to ± 10 deg if this value plus the angularity change from thermal expansion of the system does not exceed functional limitations established by the support manufacturer and the Designer assures that any increased loadings on the connection are compatible with the design.

T-1240 WEIGHT TOLERANCES**T-1241 General**

Tolerances on both uniformly distributed weights and concentrated weights are included in this Appendix. These tolerances do not imply that components should be weighed, but provide for reconciliation of weight changes identified in the design process. As a practical matter, the concentrated weight of items is a concern primarily for valves and other in-line components of significant weight, but particularly power operated valves. Changes to the distributed weight of insulation systems and other piping items should also be considered.

T-1242 Uniformly Distributed Weight

The uniformly distributed weight for the piping system may vary by $\pm 20\%$ from the as-analyzed weight.

This weight tolerance may need to be reduced for piping primarily supported by constant force supports or springs.

T-1243 Concentrated Weight

Concentrated weights of in-line items such as valves may vary by the greater of $\pm 20\%$ of the analyzed weight or 20 lb (9 kg).

NONMANDATORY APPENDIX U

ARTICLE U-1000 RULES FOR PUMP INTERNALS

U-1100 INTRODUCTION

U-1110 SCOPE

These rules apply to materials, fabrication and examination of internal items for Classes 1, 2 and 3 pumps. Pump internal items are those parts other than pump cases, inlets and outlets, covers clamping rings, seal housings, related bolting and other items as covered in Subsections NB, NC, and ND.

U-1120 CATEGORIES

Category as set forth in [Table U-1600-1](#) is the grouping of various pump internal items for the purpose of applying the rules of this Appendix. Categories for typical pump types are shown in [Figures U-1500-1](#) through [U-1500-7](#). The figures are not to scale, and are not intended to convey any preference for pump type or design, but are provided as a guide to the manufacturer to identify the various internal items of a pump for categorization. In determining categories for items of pump types not specifically illustrated, a pump or pump detail which is most nearly representative shall apply. Categories 1 and 2 are pressure retaining items presently covered by Subsections NB, NC and ND. Categories 3 through 6 are pump internal items which may be constructed in accordance with this Appendix except that Material Manufacturers and/or Material Suppliers for Category 3 through 6 items are not required to comply with NCA-3800. Material which forms an integral welded extension to Category 1 material in [Figure U-1500-1](#), items 1–5, may be classified as Category 4 or 5 material when it serves a nonpressure-retaining function and its structural effect is considered in the design of the Category 1 material. For use in Class 1, 2, and 3 pumps the extension material shall satisfy the requirements of NB-2300 NC-2300, or ND-2300.

U-1200 GENERAL REQUIREMENTS

U-1210 RESPONSIBILITIES AND DUTIES

It is the responsibility of the Certificate Holder manufacturing pumps to assign each item of a pump to the proper category and to indicate the categories in a report or on a drawing.

U-1220 CODE STAMPING

Certification Mark is not required for pump internal items.

U-1300 MATERIALS

U-1310 GENERAL REQUIREMENTS FOR MATERIAL

U-1311 Scope of Principal Terms Employed

The term *materials* as used in this Appendix applies to those items produced to material specifications permitted by Section III, Division 1, and/or other material permitted by this Appendix.

U-1312 Permitted Material Specifications

(a) Materials used for Category 3 and 4 items shall conform to the requirements of one of the specifications for materials given in [Table U-1610-1](#) of this Appendix for Class 1, 2 and 3 pumps; materials listed in Section II, Part D, Table 2A for Class 1 and 2 pumps; materials listed in Table 2A for Class 2 pumps; or materials listed in Table 1A for Class 3 pumps; and to the special requirements of this Appendix which apply to the item for which the material is used.

(b) Category 5 and 6 items may be made from any material suitable for the intended service.

(c) The Certificate Holder manufacturing pumps shall provide a list which identifies the material used for each Category 3, 4, 5 or 6 item. This list may be a bill of materials or a separate list.

(d) Where the tensile strength, yield strength, tempering temperature or aging temperature listed in [Table U-1610-1](#) differ from the requirements of the material specification, the minimum requirements listed in [Table U-1610-1](#) shall apply.

U-1313 Special Requirements Conflicting With Permitted Material Specifications

(a) Special requirements stipulated in this Appendix shall apply in lieu of the requirements of the material specifications wherever the special requirements conflict with the material specification requirements. Where the

**Table U-1600-1
Summary of Requirements**

Typical Items	Category No. & Pump Class	Stress Report	Certified Material Test Report	Nondestructive Examination	Impact Testing	Material Identif.
Pressure Retaining Items	Category 1: Class (All)	X	Subsection NB, NC, ND			
Pressure Retaining Bolting	Category 2: Class (All)	X	Subsection NB, NC, ND			
Shafting Line Shaft Couplings	Category 3: Class 1	—	X	[Note (5)] [Note (7)]	X	X
Impeller Nuts or Impeller Locking Screw	Class 2 Class 3	— —	X X	— —	[Note (4)] [Note (4)]	X [Note (1)]
Impellers Bearing Support	Category 4: Class 1	—	[Note (2)] [Note (3)]	[Note (6)] see U-1120	— see U-1120	X
Bearings Journals	Class 2 Class 3	— —	[Note (2)] [Note (2)]	— see U-1120 — see U-1120	— see U-1120 — see U-1120	X [Note (1)]
Im./Case Rings Keys	Category 5: Class 1	—	[Note (2)]	— see U-1120	— see U-1120	[Note (1)]
Mech. Seal (Met.)	Class 2	—	—	— see U-1120	— see U-1120	[Note (1)]
Parts	Class 3	—	—	— see U-1120	— see U-1120	[Note (1)]
Bolting (Internal)						
Bearings						
Journals						
Packing	Category 6: Class 1	—	—	—	—	—
Gaskets	Class 2	—	—	—	—	—
"O" Rings	Class 3	—	—	—	—	—
Carbon						

NOTES:

- (1) Quality control system.
- (2) Materials Manufacturer's Certificate of Compliance.
- (3) Certified Material Test Reports required for impellers.
- (4) When required for the pump per Design Specification.
- (5) Ultrasonic Examination required for Class 1 shafting.
- (6) Magnetic Particle or Liquid Penetrant Examination required for Class 1 impellers.
- (7) Magnetic Particle or Liquid Penetrant Examination required for all Class 1, Category 3 items.

special requirements include an examination, test, or treatment which is also required by the material specification, the examination, test, or treatment need be performed only once. Any required nondestructive examinations shall be performed as specified in [U-1430](#). Any examination, repair, test, or treatment required by the material specification or this Appendix may be performed by the Material Manufacturer, Material Supplier, or the Certificate Holder manufacturing pumps. The Material Manufacturer or Material Supplier shall obtain approval from the Certificate Holder manufacturing pumps for the weld repair of materials (see [U-1440](#)).

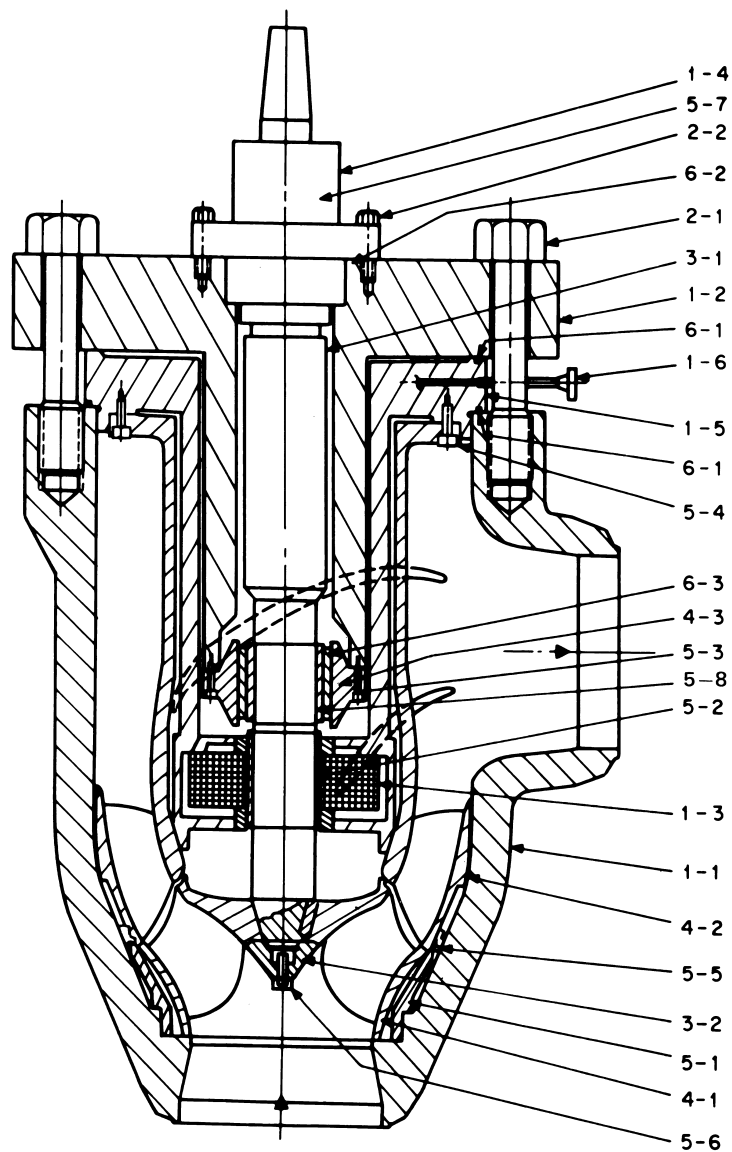
(b) For materials listed in [Table U-1610-1](#) for Category 3 and 4 items, the tensile test requirements of the material specification may be performed on representative samples

of each heat of material used, for each specified heat treatment. The tensile strength and yield strength results shall satisfy the specified values and be below the maximum specified values listed in [Table U-1610-1](#). Where the material will be used to fabricate various item sizes in different heat treated thicknesses, the manufacturer shall assure himself that the heat treatment specified will be effective for the entire size range.

U-1314 Certification of Materials

(a) *Certification by Material Manufacturer.* The Material Manufacturer shall provide a Certified Material Test Report for Category 3 items and Category 4, Class 1 impellers, including all welding and brazing materials used on these items. The Material Manufacturer shall certify that

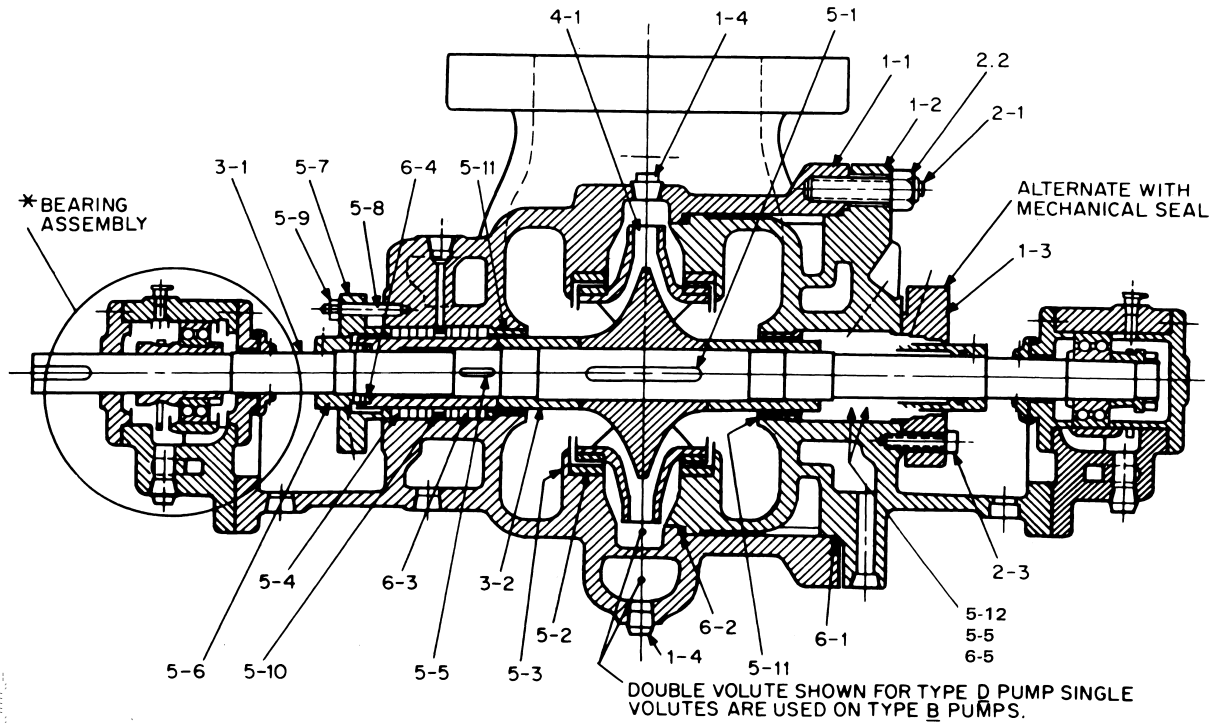
Figure U-1500-1
Typical for Type A, C, E, F, and/or Some J (NB-3400) Pumps



Key to Fig. U-1500-1

Cat.	Pump Item	Name (Typical)
1	1-1	Casing
	1-2	Main Flange
	1-3	Heat Exchanger
	1-4	Seal Housing
	1-5	Thermal Barrier
	1-6	TB Connection Piping (Include Flange)
2	2-1	Main Flange Bolts
	2-2	Seal Housing Bolts
3	3-1	Shaft
	3-2	Impeller Nut
4	4-1	Impeller
	4-2	Diffuser
	4-3	Bearing Support
5	5-1	Diffuser Adapter (Casing Wear Ring)
	5-2	Thermal Sleeve
	5-3	Radial Bearing Bolts
	5-4	Turning Vane/Thermal Barrier Bolts
	5-5	Diffuser/Diffuser Adapter Bolts
	5-6	Impeller Nut Locking Bolt
	5-7	Seals—Mechanical Seal (Metallic)
	5-8	Bearings/Journals and Shaft Sleeves
6	6-1	Flange Gaskets
	6-2	Seal Housing O-Ring
	6-3	Radial Bearing
	6-4	Mechanical Seal (Nonmetallic)

Figure U-1500-2
Typical for Type B and D Pumps (NC-3400 and ND-3400)

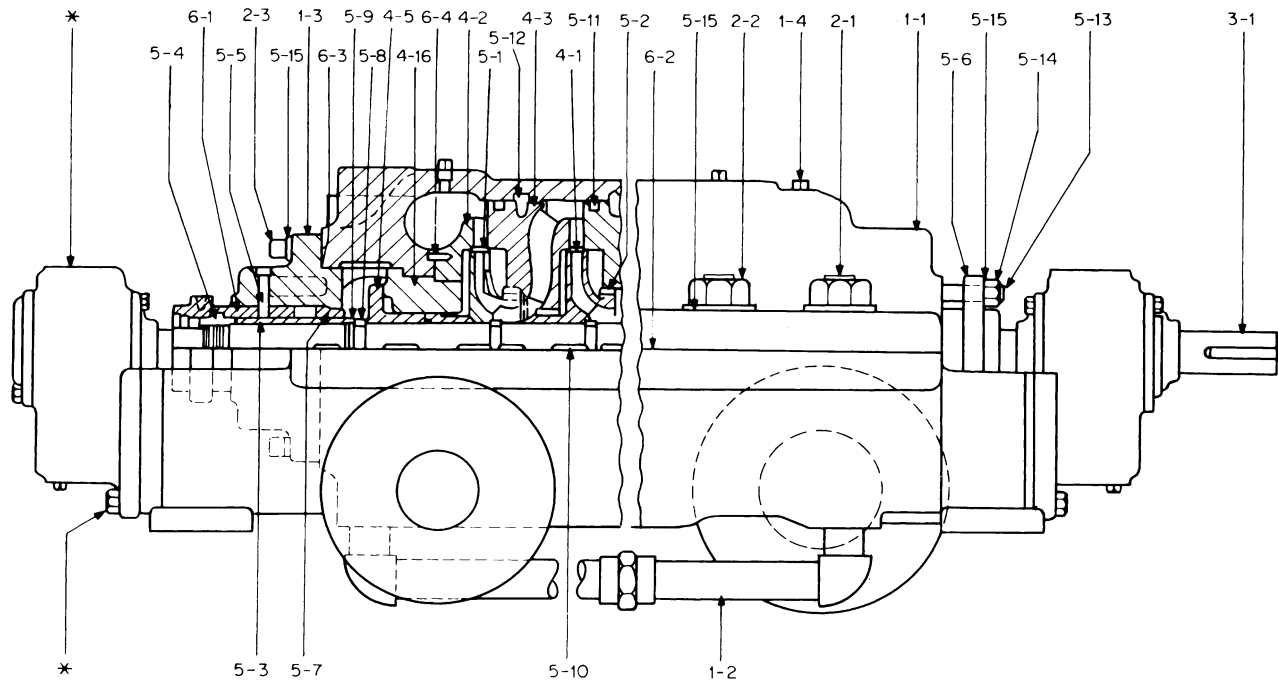


Key to Fig. U-1500-2

Cat.	Pump Item	Name (Typical)	Cat.	Pump Item	Name (Typical)
1	1-1	Case	5	5-5	Shaft Sleeve Key
	1-2	Head		5-6	Shaft Sleeve Nut
	1-3	Seal Plate		5-7	Packing Gland
	1-4	Case Plug		5-8	Packing Gland Stud
2	2-1	Case Stud		5-9	Packing Gland Nut
	2-2	Case Stud Nut		5-10	Lantern Ring
	2-3	Seal Plate Bolt		5-11	Packing Box Bushing
3	3-1	Pump Shaft		5-12	Mech. Seal Metallic Parts
	3-2	Impeller Locknut	6	6-1	Head Gasket
4	4-1	Impeller		6-2	Inner Head Gasket
				6-3	Packing
5	5-1	Impeller Key		6-4	Shaft Sleeve Seal Ring
	5-2	Case Wearing Ring		6-5	Mech. Seal Nonmetallic Parts
	5-3	Impeller Wearing Ring			
	5-4	Shaft Sleeve			
			* Bearing Assembly		

*Part not covered by this Appendix

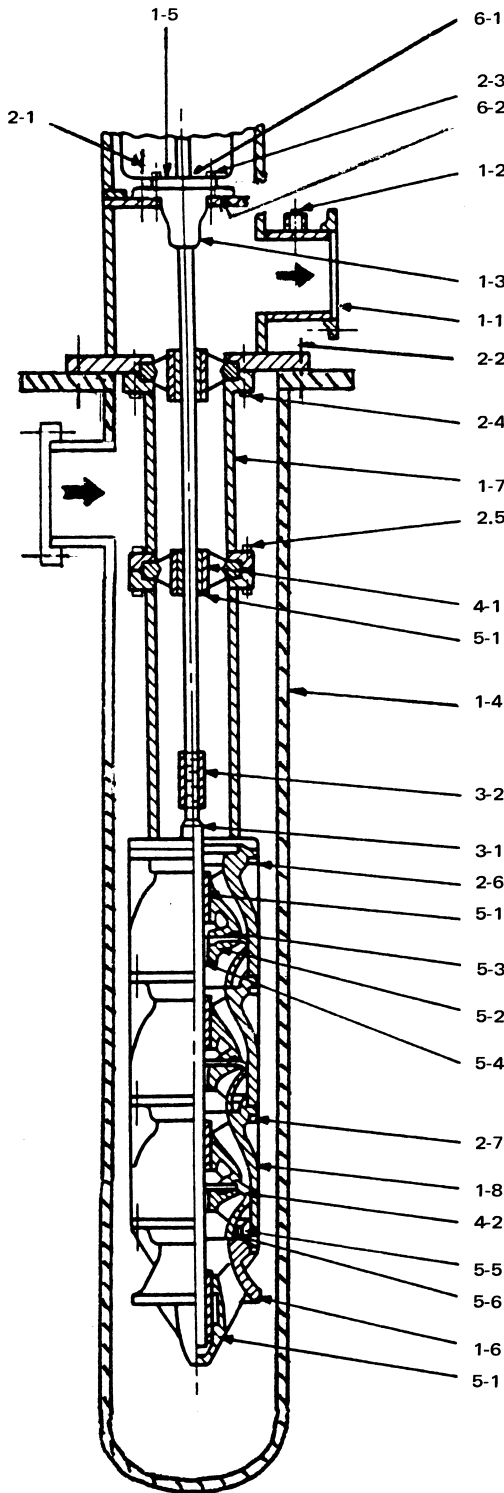
Figure U-1500-3
Typical for Type G and H Pumps (NC-3400 and ND-3400)



Key to Fig. U-1500-3

Cat.	Pump Item	Name (Typical)	Cat.	Pump Item	Name (Typical)
1	1-1	Casing	5	5-3	Shaft Sleeve
	1-2	Piping, Balance Line (Including Fittings)		5-4	Nut, Shaft Sleeve
	1-3	Stuffing Box		5-5	Seal Cage (Metallic)
	1-4	Plugs, Casing		5-6	Gland Halves
2	2-1	Studs, Main Flange		5-7	Bushing, Stuffing Box
	2-2	Nuts, Main Flange		5-8	Split Ring
	2-3	Cap Screws, Stuffing Box		5-9	Retaining Ring
3	3-1	Shaft		5-10	Key
				5-11	Sealing Ring, Channel
4	4-1	Impeller		5-12	Pin
	4-2	Diffuser		5-13	Studs, Gland
	4-3	Channel Ring		5-14	Nut, Gland
	4-4	Balance Sleeve		5-15	Washer
	4-5	Balance Drum	6	6-1	Packing, Stuffing Box
5	5-1	Impeller Ring		6-2	Gasket, Main Flange
	5-2	Casing Ring		6-3	Gasket, Metallic-Asbestos
				6-4	O-Ring

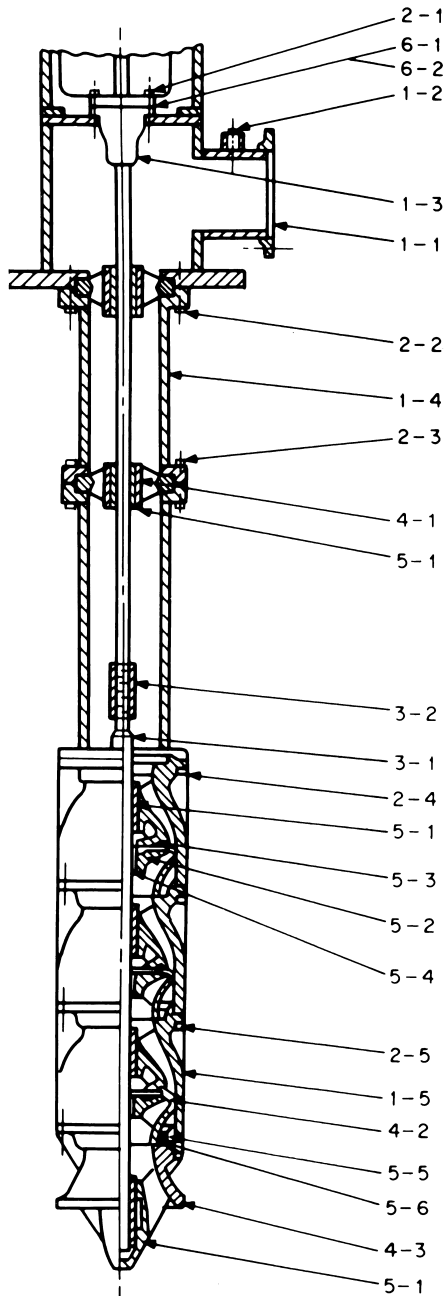
Figure U-1500-4
Typical for Type K Pumps (NC-3400 and ND-3400)



Key to Fig. U-1500-4

Cat.	Pump Item	Name (Typical)
1	1-1	Head
	1-2	Pipe Plug—Discharge
	1-3	Stuffing Box
	1-4	Barrel
	1-5	Packing Gland
	1-6	Suction Ball
	1-7	Column
	1-8	Bowl
2	2-1	Bolting—Stuffing Box to Head
	2-2	Bolting—Head to Barrel
	2-3	Bolting—Packing Gland to Stuffing Box
	2-4	Bolting—Column to Head
	2-5	Bolting—Column to Column
	2-6	Bolting—Bowl to Column
	2-7	Bolting—Bowl to Bowl
3	3-1	Shaft
	3-2	Coupling—Shaft
4	4-1	Spider
	4-2	Impeller
5	5-1	Bearing
	5-2	Split Ring—Thrust
	5-3	Capscrew—Split Ring
	5-4	Key—Impeller
	5-5	Wear Ring—Bowl
	5-6	Wear Ring—Impeller
6	6-1	Packing
	6-2	Gasket—Stuffing Box to Head

Figure U-1500-5
Typical for Type L Pumps (NC-3400 and ND-3400)



Key to Fig. U-1500-5

Cat.	Pump Item	Name (Typical)
1	1-1	Head
	1-2	Pipe Plug—Discharge
	1-3	Stuffing Box
	1-4	Column
	1-5	Bowl
2	2-1	Bolting—Stuffing Box to Head
	2-2	Bolting—Column to Head
	2-3	Bolting—Column to Column
	2-4	Bolting—Bowl to Column
	2-5	Bolting—Bowl to Bowl
3	3-1	Shaft
	3-2	Coupling—Shaft
4	4-1	Spider
	4-2	Impeller
	4-3	Suction Bell
5	5-1	Bearing
	5-2	Split Ring—Thrust
	5-3	Capscrew—Split Ring
	5-4	Key—Impeller
	5-5	Wear Ring—Bowl
	5-6	Wear Ring—Impeller
6	6-1	Packing
	6-2	Gasket—Stuffing to Head

Figure U-1500-6
Reciprocating Plunger Pump (NC-3400 and ND-3400)

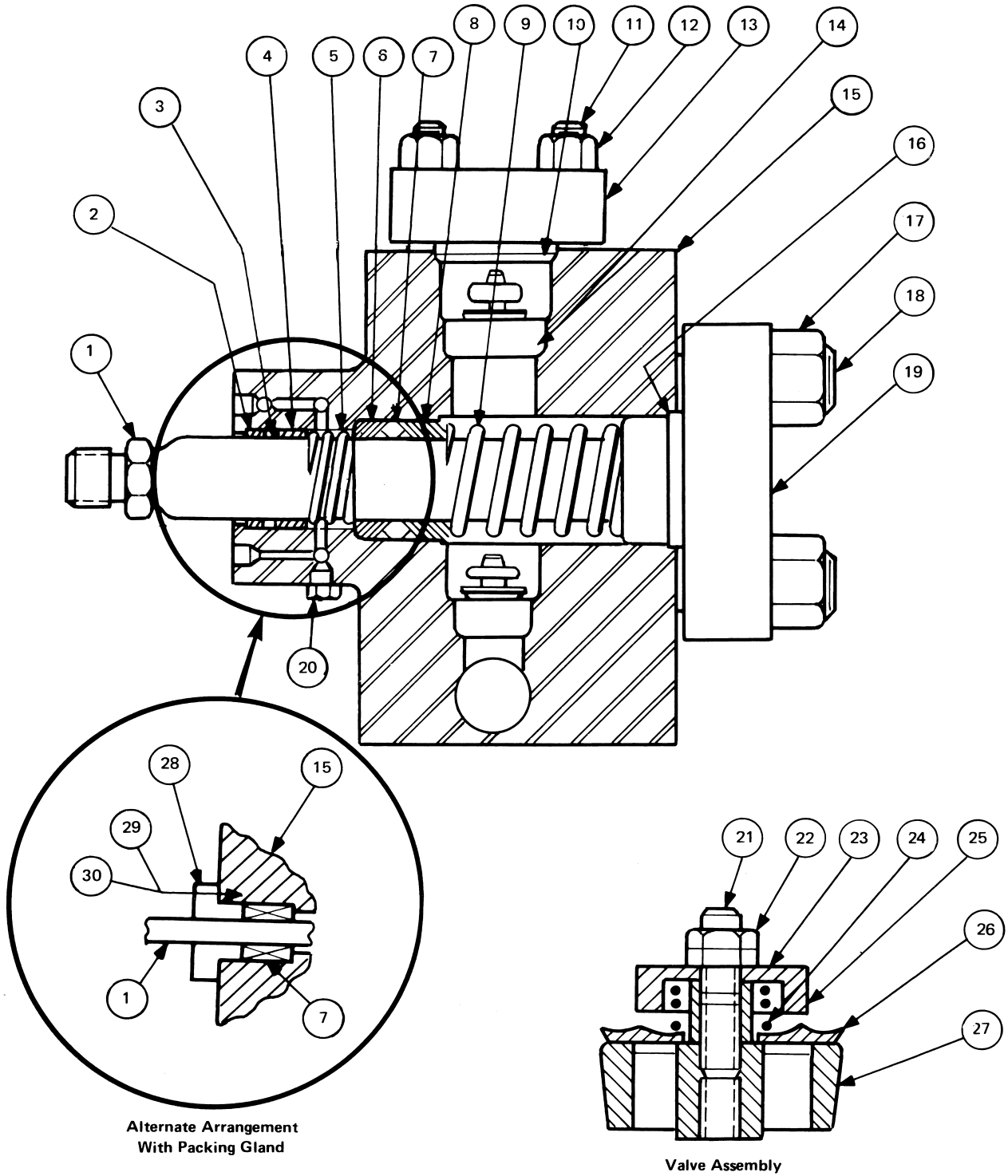


Figure U-1500-6
Reciprocating Plunger Pump (NC-3400 and ND-3400) (Cont'd)

Key to Fig. U-1500-6

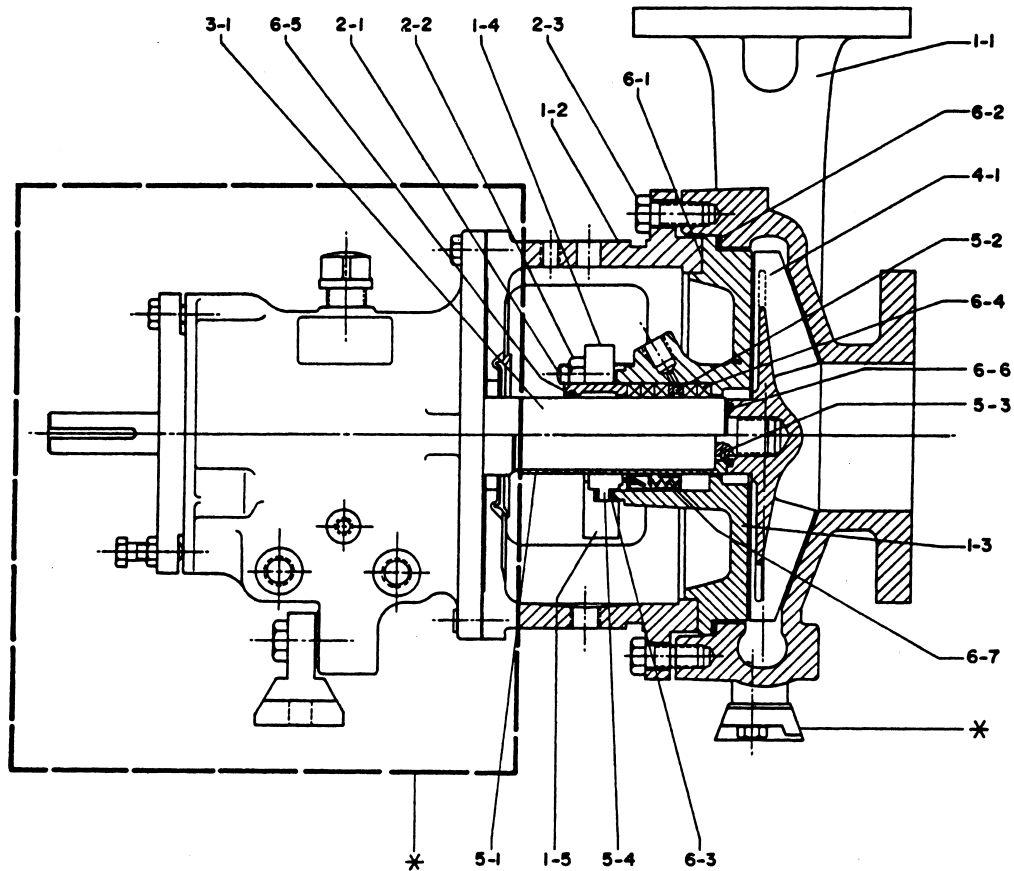
Cat.	Pump Item	Name (Typical)
1	13	Upper Cap
	15	Cylinder Block
	19	Front Cap
	28	Packing Gland
2	11	Upper Cap Stud
	12	Upper Cap Stud Nut
	17	Front Cap Stud Nut
	18	Front Cap Stud
	29	Stud (Gland)
	30	Nut (Gland)
3	1	Plunger
5	2	Packing Adj. Ring (Secondary)
	4	Plunger Ring (Secondary)
	5	Packing Spring (Secondary)
	6	Packing Adj. Ring (Primary)
	8	Plunger Ring (Primary)
	9	Packing Spring (Primary)
	14	Valve Assembly
	20	Plug
	21	Stud
	22	Double Locknut
	23	Sleeve
	24	Spring
	25	Guard Stop
6	26	Valve
	27	Seat
	3	Packing (Secondary)
	7	Packing (Primary)
	10	Upper Cap Gasket
	16	Front Cap Gasket

the contents of the report are correct and accurate, and that all operations performed by him or his subcontractors are in compliance with the requirements of the material specification and this Appendix. Alternatively, the Material Manufacturer shall provide a Certified Material Test Report for operations he performed and at least one Certified Material Test Report from each of his subcontractors for operations they performed. Chemical analysis, tests, examinations, and heat treatments required by the material specification that were not performed shall be listed on the Certified Material Test Report. A Material Manufacturer's Certificate of Compliance with the material specification, grade, class, and heat treated condition, as applicable, may be provided in lieu of a Certified Material Test Report for material used for pumps with inlet

connections 2 in. nominal pipe size and less, and bolting 1 in. nominal diameter and less. Material identification including any marking code (see U-1316) shall be described in the Certified Material Test Report or Certificate of Compliance as applicable.

(b) Certification by Material Supplier. The Material Supplier who completes any operation not performed by the Material Manufacturer shall provide a Certified Material Test Report for all operations performed by him. This certification affirms that the contents of the report are correct and accurate, and that all test results and operations performed by him or his subcontractors are in compliance with the material specification and the applicable material requirements of this Section as designated by the purchaser.

Figure U-1500-7
Typical for Type A and C Pumps (NC-3400 and ND-3400)



Key to Fig. U-1500-7

Cat.	Pump Item	Name (Typical)	Cat.	Pump Item	Name (Typical)
1	1-1	Casing	5	5-1	Shaft Sleeve
	1-2	Frame Adapter		5-2	Lantern Ring
	1-3	Stuffing Box Cover		5-3	Pin—Sleeve
	1-4	Gland—Packing		5-4	Mech. Seal (Metallic Parts)
	1-5	Gland—Mech. Seal	6	6-1	Gasket—Adapter/Cover
2	2-1	Stud-Gland		6-2	Gasket—Casing
	2-2	Nut—Gland Stud		6-3	Gasket—Gland
	2-3	Cap Screw—Case		6-4	Packing—Stuffing Box
3	3-1	Shaft		6-5	Packing—Gland
				6-6	O-Ring—Shaft Sleeve
4	4-1	Impeller		6-7	Mech. Seal (Nonmetallic Parts)

*Part not covered by this Appendix

Table U-1610-1
Materials for Pump Internal Items for Class 1, 2, and 3 Pumps

Material	Product Form	Spec. No. (8)	Type or Grade	Notes	Diameter or Thickness, in. (mm)
1Cr-0.2Mo	Bar, Rod	A-331	4140	(2)(5)	Up to 2 (50) incl.
	Bar, Rod	A-331	4340	(2)(5)	
1Cr-0.2Mo-0.3Pb	Bar, Rod	A-331	41L40	(2)(5)	
13Cr	FFBS	A-182	F6	(1)(4)(5)	Up to 8 (200) incl.
	Plate	A-240	410, 410S	(1)(4)(5)	Up to 8 (200) incl.
	Tube	A-268	TP410	(4)(5)	Up to 8 (200) incl.
	Bar, Shapes	A-276	403, 410	(4)(5)	
	Billets, Bars	A-314	403, 410	(4)(5)	Up to 8 (200) incl.
	Billets, Bars	A-314	416, 416Se	(2)(4)(5)	Up to 8 (200) incl.
	Castings	A-217	CA15	(4)(5)	
	Bar, Shapes	A-479	410	(4)(5)	Up to 8 (200) incl.
	Bars	A-582	416, 416Se	(2)(4)(5)	Up to 8 (200) incl.
	Bars	A-582	403, 410	(4)(5)	Up to 8 (200) incl.
	Bar, Shapes	A-276	414	(4)(5)	Up to 8 (200) incl.
	Castings	A-296	CA15	(4)(5)	
	Castings	A-296	CA40	(4)(5)	
	Castings	A-487	CA15M	(4)(5)(6)	
13Cr-4Ni	Castings	A-296	CA6NM		
	Castings	A-487	CA6NM		
13Cr	Bar, Shapes	A-276	420	(2)	
	Bar, Shapes	A-276	420	(1)(2)(5)	
18Cr	Bar, Shapes	A-276	440A	(1)(2)(5)(7)	Up to 8 (200) incl.
	Bar, Shapes	A-276	440C	(1)(2)(5)(7)	Up to 8 (200) incl.
13Cr	Bar, Shapes	A-565	615	(2)(5)(7)	
	Bar, Shapes	A-565	616	(2)(5)(7)	
17Cr-4Ni-4Cu	Bar, Shapes	A-564	630	(1)(2)(4)(5)	8 (200) max
	Castings	AMS 5355B		(1)(2)(4)(5)	
		AMS 5398A		(2)(4)(5)	
				(2)(4)	
				(2)(4)	
15Cr-6Ni-Cu-Mo	Bar, Shapes	A-564	XM-25	(2)(5)	8 (200) max
	Bar, Shapes	A-564	XM-25	(2)	8 (200) max
	Bar, Shapes	A-564	XM-25	(2)	8 (200) max
25Ni-15Cr-2Ti	Bolting	A-453	660	(2)	
25Ni-15Cr-2Ti	Bar, Shapes	A-638	660	(2)(5)	
16Cr-12Ni-2Mo	Bar, Shapes	A-276	316	(5)	Up to $\frac{3}{4}$ (19) $\frac{3}{4}$ to 1 (19 to 25) 1 to $1\frac{1}{4}$ (25 to 32) $1\frac{1}{4}$ to $1\frac{1}{2}$ (32 to 38) Up to 2 (50) incl.
Phos-Bronze	Bar, Rod, Shapes	B-139	544		
Al-Bronze	Castings	B-148	955		
		B-148	952		
Pb-Sn-Bronze	Castings	B-584	932		
		B-584	937		
Cu-Sn-Pb-Zn	Castings	B-584	836		
Ni-Cr-Fe	Bar, Shapes	B637-84A	718	(2)(5)	4 (100) max
	Bar, Shapes	B637-84A	688 Type 2	(2)(5)	

2013 SECTION III APPENDICES

Table U-1610-1
Materials for Pump Internal Items for Class 1, 2, and 3 Pumps (Cont'd)

Cond.	Tensile Strength, psi (MPa), min	Yield Strength, psi (MPa), min	Minimum Tempering or Aging Temp., °F (°C)	Spec No.
HT	180,000 (1241)	155,000 (1069)	800 (425)	A-331
HT	178,000 (1227)	156,000 (1076)	1000 (540)	A-331
HT	180,000 (1241)	155,000 (1069)	800 (425)	A-331
	165,000 (1138)	132,000 (910)	900 (480)	A-182
	136,000 (938)	112,000 (772)	1025 (550)	A-240
	125,000 (862)	100,000 (690)	1075 (580)	A-268
	120,000 (827)	90,000 (620)	1100 (595)	A-276
	116,000 (800)	92,000 (634)	1125 (605)	A-314
	110,000 (758)	90,000 (620)	1175 (635)	A-314
	100,000 (690)	80,000 (552)	1275 (690)	A-217
	90,000 (620)	65,000 (448)	1375 (745)	A-479
				A-582
				A-582
				A-276
				A-296
				A-296
				A-487
	110,000 (758)	80,000 (552)	1100 (595)	A-296
	110,000 (758)	80,000 (552)	1100 (595)	A-487
A	95,000 (655)	50,000 (345)	NA (NA)	A-276
QT	250,000 (1724)	195,000 (1345)	750 (400)	A-276
HT	285,000 (1965)	275,000 (1896)	500 (260)	A-276
HT	285,000 (1965)	275,000 (1896)	600 (315)	A-276
HT	140,000 (965)	110,000 (758)	1150 (620)	A-565
HT	140,000 (965)	110,000 (758)	1150 (620)	A-565
PH	190,000 (1310)	170,000 (1172)	900 (480)	A-564
PH	155,000 (1069)	145,000 (1000)	1025 (550)	AMS 5355B
PH	145,000 (1000)	125,000 (862)	1075 (580)	AMS 5398A
PH	140,000 (965)	115,000 (793)	1100 (595)	
PH	135,000 (931)	105,000 (724)	1150 (620)	
PH	180,000 (1241)	170,000 (1172)	900 (480)	A-564
PH	125,000 (862)	75,000 (517)	1150 (620)	A-564
STr	125,000 (862)	95,000 (655)	NA (NA)	A-564
A or B	130,000 (896)	85,000 (586)	1325 (720)	A-453
1 or 2	130,000 (896)	85,000 (586)	1300 (700)	A-638
B	125,000 (862)	100,000 (690)	NA (NA)	A-276
B	115,000 (793)	80,000 (552)	NA (NA)	
B	105,000 (724)	65,000 (448)	NA (NA)	
B	100,000 (690)	50,000 (345)	NA (NA)	
Hard	50,000 (345)	35,000 (241)		B-139
	90,000 (620)	40,000 (276)	NA (NA)	B-148
	65,000 (448)	25,000 (172)	NA (NA)	B-148
	30,000 (207)	14,000 (97)	NA (NA)	B-584
	25,000 (172)	12,000 (83)	NA (NA)	B-584
	30,000 (207)	14,000 (97)	NA (NA)	B-584
PH	185,000 (1276)	150,000 (1034)	1325 (720)	B-637
PH	170,000 (1172)	115,000 (793)	1350 (730)	B-637

Table U-1610-1
Materials for Pump Internal Items for Class 1, 2, and 3 Pumps (Cont'd)

Material	Product Form	Spec. No. (8)	Type or Grade	Notes	Diameter or Thickness, in. (mm)
Ni-Cu	Bar, Rod	B-164	400 Class A		
	Pipe, Tube	B-165	400		Up to 5 (125)
	Pipe, Tube	B-165	400	(2)	Over 5 (125)
	FFBS	AMS 4676	Alloy 500	(5)	12 (300) max
	Rounds	AMS 4676	Alloy 500	(5)	4 (100) max
Cr-Ni-Cu-Mo	Casting	A-351	CN7M		

TABLE U-1610-1
MATERIALS FOR PUMP INTERNAL ITEMS FOR CLASS 1, 2, AND 3 PUMPS (CONT'D)

Cond.	Tensile Strength, psi (MPa), min	Yield Strength, psi (MPa), min	Minimum Tempering or Aging Temp., °F (°C)	Spec No.
A	70,000 (483)	25,000 (172)	NA	B-164
A	70,000 (483)	28,000 (193)	NA	B-165
A	70,000 (483)	25,000 (172)	NA	B-165
PH	140,000 (965)	100,000 (690)		AMS 4676
PH	135,000 (931)	95,000 (655)		AMS 4676
SHT	62,000 (427)	25,000 (172)	NA	A-351

NOTES:

- (1) Not to be used with Category 3 items, except for pumps with inlet piping connections NPS 2 (DN 50) and less.
- (2) Welding of these materials is not permitted.
- (3) Where the tensile strength, yield strength, tempering temperature, or aging temperature listed in Table U-1610-1 differ from the requirements of the material specification, the minimum requirements listed in Table U-1610-1 shall apply. The material shall be identified with this Appendix number in addition to the requirements for identification of para. U-1316.
- (4) Cross-bracketing indicates that any of the bracketed materials may be used with any of the bracketed properties.
- (5) The maximum tensile strength shall not exceed the minimum specified tensile strength listed in this Table by more than 40,000 psi (275 MPa).
- (6) Welding of this material is permitted provided the carbon content is 0.25% or less.
- (7) Service temperatures shall not exceed temperatures of 100°F (38°C) below the aging or tempering temperature.
- (8) Material shall conform to Edition specified by N-Certificate Holder.

(c) Certified Material Test Reports or Material Manufacturer or Material Supplier Certificates of Compliance are not required for Category 5 and 6 items.

U-1315 Welding and Brazing Materials

All welding and brazing materials used on Category 3 and 4 items shall meet the requirements of NB-2400, NC-2400, or ND-2400, as applicable.

U-1316 Material Identification

U-1316.1 Class 1 and 2 Pump Items.

(a) The identification of Materials for Category 3 items used for Class 1 and 2 pumps shall consist of marking or tagging the material with the applicable specification number, grade, heat number or heat code, and any additional marking required to facilitate traceability of the reports of the results of all tests and examinations performed on the material, except that heat number identification is not required for pumps with inlet connections 2 in. nominal pipe size (DN 50) and less. Alternatively, a

Certification Mark and/or code may be used which identifies the material with the Materials Certification and such Certification Mark and/or code shall be explained in the certificate (see U-1314). For identification and marking during fabrication by the Pump Manufacturer, see U-1420.

(b) The identification of Materials for Category 4 items used for Class 1 and 2 pumps shall consist of marking or tagging the material or its container in accordance with the marking requirements of the applicable material specification. Category 5 items shall be identified as set forth in the Manufacturer's Quality System Program.

(c) Materials may be marked by any method which will not result in any harmful contamination or sharp discontinuities. Stamping, when used, shall be done with blunt-nosed-continuous or blunt-nosed-interrupted-dot die stamps.

U-1316.2 Class 3 Pump Items. The identification of materials for Category 3 through 5 items used for Class 3 pumps shall consist of marking the material or its container in accordance with the requirements of the Manufacturer's Quality System Program.

U-1316.3 Welding and Brazing Material Identification. Welding and brazing materials shall be clearly identified by legible marking on the package or container so that they are identifiable as acceptable material until the material is actually consumed in the process.

U-1320 FRACTURE TOUGHNESS REQUIREMENTS FOR CATEGORY 3 MATERIALS

U-1321 Materials to Be Impact Tested

Impact tested specimens shall be representative of the final heat treatment of the finished part.

U-1321.1 Materials for Which Impact Testing Is Required. Materials for Category 3 items for Class 1 pumps, and for Class 2 and 3 pumps when required by the Design Specification, shall be impact tested in accordance with the requirements of [U-1320](#) except that the following materials do not require impact testing:

- (a) all thickness of materials for pumps with a nominal inlet pipe size 6 in. (150 mm) diameter and smaller
- (b) materials for pumps with all pipe connections of $\frac{5}{8}$ in. (16 mm) nominal pipe wall thickness and less
- (c) materials with a nominal section thickness of $\frac{5}{8}$ in. (16 mm) and less
- (d) bars with a nominal cross-sectional area of 1 in.² (650 mm²) and less
- (e) austenitic stainless steels
- (f) nonferrous materials

U-1321.2 Impact Test Procedure.

U-1321.2.1 Charpy V-Notch Tests. The Charpy V-Notch Test shall be performed in accordance with SA-370. Specimens shall be in accordance with SA-370, Figure 11, Type A. A test shall consist of a set of 3 full-size 10 × 10 mm specimens. The test temperature and lateral expansion shall be reported in the Certified Material Test Report.

U-1321.2.2 Location and Orientation of Test Specimens. Impact test specimens shall be removed from the locations and orientations specified by the material specification for tensile test specimens in each product form.

U-1321.2.3 Test Requirements and Acceptance Standards. Three Charpy V-Notch specimens shall be tested at a temperature equal to or lower than the lowest service temperature. All three specimens shall meet or exceed 15 mils (0.45 mm) lateral expansion. Lowest service temperature is the minimum temperature of the fluid retained by the pump. The lowest service temperature shall be specified in the Design Specification.

U-1321.2.4 Retests. One retest at the same temperature may be conducted provided

- (a) not more than one specimen per test is below the minimum requirements, and
- (b) the specimen not meeting the minimum requirements is not lower than 5 mils (0.13 mm) below the specified requirements

U-1400 FABRICATION REQUIREMENTS

Category 3 through 5 items shall be fabricated in accordance with the requirements of [U-1400](#) and shall be manufactured from materials which meet the requirements of [U-1300](#).

U-1410 CERTIFICATION OF MATERIALS AND FABRICATION BY PUMP MANUFACTURER

The pump manufacturer shall provide certification that all treatments, tests, repairs, or examinations performed on pump items are in compliance with the requirements of this Appendix. Reports of all required treatments and the results of all required tests, repairs, and examinations performed shall be maintained in accordance with NCA-3862.1 or NCA-4134.17.

U-1420 MATERIAL IDENTIFICATION

Material for Category 3 and 4 items for Class 1 and 2 pumps shall carry identification markings, including heat treatment grade either directly on the item or on a separate tag that accompanies the item, which will be maintained during and after assembly.

U-1430 EXAMINATION OF MATERIALS

Bars and forgings for Category 3 pump shafting for Class 1 pumps, shall be ultrasonically examined in accordance with NB-2542 and NB-2547. Materials for Class 1 pumps, Category 3 items and Category 4 impellers for Class 1 pumps shall be examined on all external and accessible internal surfaces by the magnetic particle or liquid penetrant method in accordance with Section V. The examination may be performed by the Material Manufacturer, Material Supplier, or the pump manufacturer (see [U-1313](#)). Acceptance standards for magnetic particle and/or liquid penetrant examination shall be as follows.

- (a) Only indications with major dimensions greater than $\frac{1}{16}$ in. (1.5 mm) shall be considered relevant.
- (b) The following relevant indications are unacceptable:
 - (1) any linear indications greater than $\frac{1}{16}$ in. (1.5 mm) long for materials less than $\frac{5}{8}$ in. (16 mm) thick; greater than $\frac{1}{8}$ in. (3 mm) long for materials from $\frac{5}{8}$ in. (16 mm) thick to under 2 in. (50 mm) thick; and $\frac{3}{16}$ in. (5 mm) long for materials 2 in. (50 mm) thick and greater;

(2) rounded indications with dimensions greater than $\frac{1}{8}$ in. (3 mm) for thicknesses less than $\frac{5}{8}$ in. (16 mm) and greater than $\frac{3}{16}$ in. (5 mm) for thicknesses $\frac{5}{8}$ in. (16 mm) and greater;

(3) four or more indications greater than $\frac{1}{16}$ in. (1.5 mm) in a line separated by $\frac{1}{16}$ in. (1.5 mm) or less edge to edge;

(4) ten or more indications greater than $\frac{1}{16}$ in. (1.5 mm) in any 6 in.² (4000 mm²) of area whose major dimension is no more than 6 in. (150 mm) with the dimensions taken in the most unfavorable location relative to the indications being evaluated;

(5) linear nonaxial indications.

Materials for Category 4 and 5 items for Class 1 pumps and for Category 3, 4, and 5 items for Class 2 and 3 pumps shall be examined in accordance with the material specification. When Category 4 or 5 material forms an integral or welded extensions to Category 1 material (see U-1120), the examination required for the Category 4 or 5 portion shall be in accordance with this Appendix. In addition, the requirements for the examination of the Category 1 portion shall be applied for a distance of at least $2t$ (t is thickness of the Category 4 or 5 material) from the Category 1 boundary as defined by the pump designer.

U-1431 Time of Examination

Magnetic particle or liquid penetrant examination shall be performed in the finished conditions, after all heat treatment operations and postweld heat treatment, except that threaded items may be examined prior to threading. Examinations shall be performed prior to any coating or plating. Lapping of seating surfaces to reduce leakage or lapping of bearing surfaces shall not require reexamination.

U-1432 Elimination of Surface Defects

(a) Unacceptable surface defects shall be removed by grinding or machining, provided:

(1) the remaining thickness of the section is not reduced below the minimum required by the design;

(2) the depression, after grinding or machining, is blended uniformly into the surrounding surface and the depression does not affect the function of the item;

(3) after grinding or machining, the area is examined by the method which originally disclosed the defect to assure that the defect has been removed or the indication reduced to an acceptable size.

(b) If grinding or machining reduces the thickness of the section below the minimum required by the design, the item may be repaired and returned to an acceptable size.

U-1440 REPAIR BY WELDING OF CATEGORY 3, 4, AND 5 PUMP ITEMS

(a) Category 3, 4 and 5 items for Class 1, 2 and 3 pumps may be repaired by welding using the provisions of Section IX for materials in Table U-1610-1 with assigned P-numbers provided the requirements of the following subparagraphs are met.

(b) Until such time as P-numbers are assigned, welding of those materials without P-numbers shall be separately qualified as required by Section IX.

(c) Bolts, studs, nuts, and material for which welding is prohibited by Note (2) of Table U-1610-1 shall not be repair welded.

U-1441 Defect Removal

The defect shall be removed or reduced to an acceptable size by suitable mechanical or thermal cutting or gouging methods and the cavity prepared for repair.

U-1442 Qualification of Welding Procedures and Welders

(a) When impact tests of Category 3 items are required, the impact testing requirements of U-1321.2.3 shall be met in the heat affected zone regardless of filler metal used and in the weld metal for all material except austenitic and nonferrous filler metal.

(b) Except as permitted in (c) below, the welding procedure and welders or welding operators shall be qualified in accordance with Section IX.

(c) Heat-treated materials listed in Table U-1610-1 which are not capable of passing bend tests required by Section IX for procedure or performance qualification may be qualified by a Fillet Weld Test in accordance with QW-180. In addition, a minimum of two cross sections of the qualification test plate (assembly) shall be ground and etched with a suitable etchant and visually examined at 10 × magnification. The weld metal and adjacent base material of the ground and etched cross sections shall be free of cracks.

U-1443 Blending of Repaired Areas

After repair, the surface shall be blended into the surrounding surface.

U-1444 Examination of Repair Welds

Each repair weld of materials for Category 3 items for Class 1 pumps shall be examined by the method that originally disclosed the defect. The finished surface shall be examined by either the magnetic particle or liquid penetrant method in accordance with Section V. The acceptance standards shall be those specified in U-1430(a) and U-1430(b). Repair welds of materials for the other categories shall be in accordance with the material specification.

U-1445 Heat Treatment After Repair

(a) Materials listed in [Table U-1610-1](#) which are repaired by welding shall be heat treated in accordance with the requirements of NB-4600, NC-4600 and ND-4600 as applicable, or as allowed by (c) below.

(b) Materials listed in [Table U-1610-1](#) which are repaired by welding shall be heat treated and tempered or aged after repair, except as allowed by (c) below. The minimum tempering or aging temperature shall be as specified in [Table U-1610-1](#) for the finished item.

(c) Repair weld procedures for welds not exceeding the lesser of $\frac{3}{8}$ in. (10 mm) or 10% of the section thickness shall be qualified by test weld specimens meeting the tensile and bend test requirements of Section IX, without PWHT, for the material repaired.

U-1446 Repair Weld Report

A record shall be made of each defect repair of Category 3 items for Class 1 pumps in which the depth of the repair cavity exceeds the lesser of $\frac{3}{8}$ in. (10 mm) or 10% of the section thickness. The record shall include the location and size of the repaired cavity, the welding material, the welding procedure, the heat treatment, and the examination results.

U-1450 WELDING REQUIREMENTS

Except as permitted in [U-1442](#) and [U-1451](#), all welds shall be made using qualified welding procedures and welders or welding operators in accordance with Section IX.

U-1451 Special Fabrication Welds

Fillet welds and partial penetration welds $\frac{1}{4}$ in. (6 mm) and less in size may be made in the fabrication of pump items or between pump items where either of the items is a material listed in [Table U-1610-1](#) provided welding is not prohibited by Note (2) in [Table U-1610-1](#), and the procedures and welders are qualified as follows:

(a) a test assembly shall be made for each combination of materials to be welded.

(b) the test assembly shall be a duplicate of the production weld joint or a groove butt weld $\frac{1}{4}$ in. (6 mm) minimum thickness.

(c) the test assembly shall be sectioned (a minimum of four cross sections), ground, etched with a suitable etchant, and visually examined at 10 × magnification. All surfaces of the weld and adjacent base material(s) shall be free of cracks.

U-1452 Examination of Welds

All welds shall be examined by the magnetic particle or liquid penetrant method in accordance with Section V. The time of examination shall be in accordance with NB-5120, NC-5120 and ND-5120. Acceptance standards shall be as follows.

U-1453 Acceptance Standards

(a) Only indications with major dimensions greater than $\frac{1}{16}$ in. (1.5 mm) shall be considered relevant.

(b) The following relevant indications are unacceptable:

(1) any cracks or linear indications

(2) rounded indications with dimensions greater than $\frac{3}{16}$ in. (5 mm)

(3) four or more rounded indications in a line separated by $\frac{1}{16}$ in. (1.5 mm) or less edge to edge

(4) ten or more rounded indications in any 6 in.² (4000 mm²) of surface with the major dimensions of this area not to exceed 6 in. (150 mm), with the area taken in the most unfavorable location relative to the indications being evaluated

U-1454 Heat Treatment of Welds

(a) Postweld heat treatment of welds which join materials listed in [Table U-1610-1](#) shall be in accordance with the postweld heat treatment requirements of NB-4620, NC-4620 or ND-4620, as applicable.

(b) Postweld heat treatment of welds which join materials listed in [Table U-1610-1](#) shall be in accordance with the postweld heat treatment requirements of NB-4620, NC-4620, or ND-4620, as applicable. Special techniques, such as local postweld heat treatment, may be necessary to avoid changing the base material properties of the item in location not adjacent to the weld. A change in the specified postweld heat treatment temperature will require requalification of the WPS in accordance with Section IX, QW-407.1.

(c) Postweld heat treatment of welds which join materials listed in [Table U-1610-1](#) to materials listed in Section II, Part D, Table 1A shall be in accordance with the postweld heat treatment requirements of NB-4620, NC-4620 or ND-4620, as applicable. Special techniques, such as local postweld heat treatment, may be necessary to avoid changing the base material properties of the item in location not adjacent to the weld. A change in the specified postweld heat treatment temperature will require requalification of the WPS in accordance with Section IX, QW-407.1.

(d) Postweld heat treatment of welds for joining materials listed in [Table U-1610-1](#) shall be in accordance with the heat treatment specified for the material of the finished item, i.e., the heat treatment required to obtain the tensile strength and/or yield strength listed in [Table U-1610-1](#).

(e) Materials listed in [Table U-1610-1](#) subject to material specification heat treatment after repair welding shall be welded to a procedure which shall demonstrate that the required strength can be met in the weld without affecting the properties of the base material.

(f) For fillet welds and partial penetration welds $\frac{1}{4}$ in. (6 mm) and less in size, postweld heat treatment is neither required nor prohibited, provided the requirements of [U-1451](#) are met.

NONMANDATORY APPENDIX W

ARTICLE W-1000 ENVIRONMENTAL EFFECTS ON COMPONENTS

W-1100 INTRODUCTION

This Appendix presents guidance summaries for potential service degradation mechanisms not explicitly covered by ASME Section III Code design requirements. Although such service degradation is not addressed in the Code, Section III recognizes the need for the Owner to provide requirements to limit material deterioration to acceptable levels for the life of the component. The following excerpt from NCA-1130 of Section III expresses the intent of the Code in this respect.

“The rules of this Section provide requirements for new construction and include consideration of mechanical and thermal stresses due to cyclic operation. They do not cover deterioration, which may occur in service as a result of radiation effects, corrosion or erosion. These effects shall be taken into account with a view to realizing the design or the specified life of the components.”

With the preceding requirements to address service degradation mechanisms in mind, the objective of this appendix is to provide further guidance in that area. The limitations and scope of that guidance are addressed below.

Summaries of potential damage mechanisms, due to service and environmental effects, are presented on the following pages. Prevention of component degradation, beyond acceptable limits, by these mechanisms should be considered in addition to explicit Code design requirements. The following summaries provide descriptions and experience for each mechanism, illustrating specific areas of possible concern based on past experience. This experience description does not include degradation that has not been significant to date, or that may be newly introduced by system modifications or new applications. Such components should also be thoroughly evaluated for service degradation when damage mechanisms are thought to be possible.

Material selection, treatment and testing, design limitations, and mitigating actions based on a review of the literature and industry practice are also summarized for each damage mechanism.

W-1110 SCOPE

The guidance on service degradation mechanisms in this Appendix is not intended to give final details, but rather to provide summaries and references as starting points for more detailed evaluations of specific applications. This intent is discussed further at the beginning of the next section. It may be found that such detailed evaluations, on a case-by-case basis, may justify different material selections, design and operation limits than those summarized here.

Also, emphasis is placed on relatively well known problem areas, based on past experience, and additional areas of concern (not covered in this Appendix) may emerge in the future. Further theoretical projections and laboratory test programs outside the scope of this Appendix may gain some insight as to emerging and future problems.

W-1200 SECTION XI AND PLEX APPLICATIONS

Along with providing guidance and a starting point for ASME Section III material and design considerations to minimize service degradation, this report is relevant to ASME Section XI and plant life extension (PLEX) evaluations. Consideration of these damage mechanisms and the service degradation of components should be a part of establishing Section XI in-service inspection (ISI) and surveillance programs, and in projecting plant life extension limits.

ARTICLE W-2000

SUMMARIES OF CORROSION DAMAGE MECHANISMS

W-2100 STRESS CORROSION CRACKING

W-2110 GENERAL DESCRIPTION

Stress corrosion cracking (SCC) is a corrosion mechanism that forms cracks in susceptible material in the presence of an aggressive environment and tensile stresses (refs. [1]–[3]). SCC can result in component failure, component leakage, plant shutdowns, augmented inservice inspection, analysis and repairs, and eventual replacement, depending on the component.

Stress corrosion cracks may be intergranular or transgranular in nature, depending on the material, level of stress, and the environment. Intergranular stress corrosion cracking (IGSCC) is characterized by preferential attack along grain boundaries and can occur in pure water or in aqueous environments in the presence of contaminants, such as halogens, sulfates, and lead. Transgranular stress corrosion cracking (TGSCC) is characterized by preferential attack along certain crystallographic planes within grains in metals (refs. [1] and [2]) and is normally associated with contaminants.

Virtually all alloys used in nuclear power plant construction are susceptible to SCC, depending on the environment. For example, chlorides may cause both IGSCC and TGSCC in austenitic stainless steels (ref. [4] and pressurized water reactor (PWR) steam generator Alloy 600 tubes (ref. [5]) have suffered IGSCC on both the primary and the secondary side of the tubes with the secondary side cracking related to concentration of chemical contaminants in crevices and in sludge.

W-2120 PREVENTIVE MEASURES

(a) SCC occurs when the following three conditions exist simultaneously:

- (1) susceptible material
- (2) tensile stress (applied and residual)
- (3) an environment that can provide the chemical driving force for corrosion reaction

A graphic representation of this phenomena is shown in Figure W-2100-1. The elimination of any one of these factors or reduction of one of these factors to some threshold level eliminates SCC (ref. [6]).

(b) Numerous preventive measures and remedies have been proposed and studied with regard to SCC. These remedies are generally categorized into three areas, combating the three necessary ingredients for SCC.

(1) careful selection of materials aimed at preventing the use of materials susceptible to SCC in the service environment

(2) employing design features and processing techniques aimed at tensile stress reduction

(3) eliminating or reducing the severity of the corrosive environment through proper in-service chemistry control

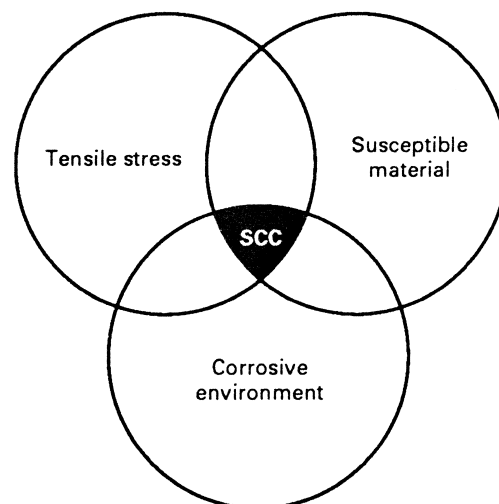
These remedies are discussed below.

W-2130 MATERIALS

Control and prevention of SCC can be achieved through judicious material selection. Considerable testing (refs. [3] and [7]) has been performed to justify the use of certain materials in potential IGSCC situations that have been observed in BWR and PWR applications.

For example, the low carbon austenitic stainless steels such as Type 316 Nuclear Grade, Type 347 Nuclear Grade, and CF3 and CF3M castings have been described (Refs. [7] and [8]) as being resistant to IGSCC in high temperature water environments. Types 304NG, 304L and 316L are also considered (ref. [8]) adequate. At least 8% delta ferrite is desired in stainless steel castings and 308L weld metal to provide resistance to IGSCC in boiling water reactor (BWR) applications (ref. [8]). High toughness, low

Figure W-2100-1
Environmental Conditions Required for SCC



strength carbon steel grades, with specified minimum yield strength of 35 ksi or lower is also an alternative, where suitable (ref. [8]). Alloy 82 nickel base weld metal is considered to be resistant in most of the BWR and PWR environments (refs. [8], [9], and [19]). Alloy 600, Alloy X-750, and Alloy 182 weld metal have been shown to be susceptible to IGSCC under some BWR and PWR conditions]. Pickling of Ni-Cr-Fe alloys has been shown to reduce material resistance to SCC (ref. [9]).

In austenitic stainless steels, sensitization and cold work from bending and forming render the material susceptible to stress corrosion cracking in certain environments. These materials, therefore, should be solution heat treated and water quenched after welding and hot or cold forming, although this requirement may be mollified when procedures are qualified for the SCC-resistant materials listed above (ref. [9]). Cold-worked austenitic stainless steels have been used successfully in PWRs in a variety of special applications, particularly fasteners, pins and drive mechanisms. Additional process controls should be implemented in such applications to negate the increased risk of IGSCC (ref. [9]). Grinding of welds in areas exposed to the environment should be minimized (ref. [3]). Grinding can produce very small fissures, increase the exposed surface area and produce a cold-worked microstructure. All three of these conditions can increase material susceptibility to SCC (ref. [9]). Fabrication procedure qualifications and material acceptance testing for resistance to IGSCC may be in accordance with the constant extension rate method (ref. [3]) or ASTM A262 practices A and E (refs. [10] and [11]).

Increased resistance to IGSCC may be gained for ferritic bolting materials by specifying reduced yield strength (less than approximately 120 ksi (800 MPa) minimum specified) material where practical (ref. [12]) and by utilizing materials produced by high purity melting procedures (ref. [13]). For certain environmental conditions and applications, some materials may be shown by specific evaluations to be acceptable at higher strength levels than the above guideline. Proper heat treatment in materials such as X-750 is important as well as strength. Control of bolting lubricants and degreasing to prevent contamination with sulfur and other harmful environments is essential (ref. [12]).

In PWR steam generator tubes, changing the fabrication heat treatment has reduced material susceptibility to certain environments. In addition, changes in alloy element concentrations can influence susceptibility of the material to certain environments. For instance, thermally treated Alloy 600 has improved resistance to primary water SCC over mill annealed Alloy 600 and Alloy 690, with a higher chromium content than Alloy 600, is considered immune to primary water SCC.

A particular case where IGSCC has occurred is in austenitic stainless steels in oxidizing environments, such as the BWR coolant (ref. [1]). IGSCC in Type 304 stainless steel piping as well as in Type 304 and Alloy 600 safe-ends has been a concern (refs. [1] and [3]).

Bolting materials, such as SA 320, Gr L43 and X-750 are also susceptible to IGSCC and can suffer relatively rapid failure (ref. [12]). PWR boron injection tank nozzles, man ways, boric acid piping and high pressure feedwater heaters (ref. [14]), as well as Alloy 600 tubes (ref. [5]) have suffered IGSCC.

Mitigating actions for condenser tubes made of copper base alloys include eliminating ammonia SCC by using resistant tube alloys (such as 70-30 copper nickel), maintaining low ammonia, and minimizing the ingress of oxygen and carbon dioxide (ref. [15]). Sulfates, nitrites, and nitrates should be minimized (ref. [15]). Keeping the tubing clean and reducing residual stresses are also important (ref. [15]). For stainless steel and titanium condenser tubes, the actions include keeping the tubing clean and avoiding certain chloride environments (ref. [5]).

Design changes to reduce SCC susceptibility should not impact on other design considerations. For example, cladding of the inside surface of an existing pipe with a corrosion resistant material such as 308L with at least 8% delta ferrite (refs. [8] and [16]) may provide corrosion resistance, but it may also, under certain conditions, compromise the inspectability of welds by ultrasonic testing (UT).

W-2140 STRESS CONTROL

Minimizing applied loads and residual stresses (such as induced by welding) reduces one of the major driving forces for IGSCC (ref. [1]). Stresses associated with the corrosion process are often linked to fabrication and installation. Welding (ref. [1]) residual tensile stresses can be major contributors to the driving force for stress corrosion cracking. Cold working, grinding, bending and high heat input welding should be minimized, unless followed by a qualified heat treatment procedure (as for shop-fabricated components).

Methods such as shot peening, heat sink welding (HSW), induction heating stress improvement (IHSI), and mechanical stress improvement (refs. [8], [16], and [17]) can be used to place the inside diameter of piping in a compressive residual stress state to resist SCC.

W-2150 ENVIRONMENT

Keeping materials clean during fabrication is also important (ref. [13]).

Stagnant fluid regions should be avoided, as should crevices (ref. [8]). Weld joints should be designed to avoid integral back-up rings or back-up bars that are left in place and can create a crevice.

During plant startup or extended downtime due to repair, replacement or long refueling outages, plant systems cool down and the water can become quite oxidizing

unless the primary and secondary systems are made inert (ref. [2]). Remedies include oxygen control during startup, de-aeration, and hydrogen water chemistry methods for BWRs (refs. [8], [16], and [17]). The presence of sulfur, chlorine, or fluorine in the water, possibly as a result of resin intrusions or other off-chemistry conditions, can create further problems.

Sulfide ions can cause hydrogen embrittlement in the higher strength steels (e.g., bolting) even at low temperatures in oxidizing environments (ref. [2]).

For temperatures below 200°F, IGSCC is not a significant concern for austenitic stainless steels in oxidizing water environments (refs. [1] and [8]), in the absence of chlorides and sulfates. In the presence of chlorides (ref. [18]), or sulfides (ref. [2]) IGSCC can occur at temperatures as low as room temperature (Ref. [18]). The presence of sulfates and chlorides or fluorides can lead to transgranular stress corrosion cracking in all austenitic stainless steels, even in the absence of oxygen.

For PWRs, improved chemistry control and contaminant reduction can reduce the occurrence of SCC. In addition, sludge removal from steam generators can be effective as it eliminates sites where corrosive chemical species can concentrate and form an aggressive environment.

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W-2200 GENERAL CORROSION AND WASTAGE

W-2210 GENERAL DESCRIPTION

General corrosion is the thinning or loss (wastage) of a metal, more or less uniformly by chemical attack (dissolution) at the surface by an aggressive environment (ref. [1]). Some consequences of such damage are wall thinning (which in the case of tubes or pipes can cause rupture or leakage), reduction of cross-sectional area of bolting (leading to failure), metal removal from valve seats (causing leakage), and buildup of corrosion products which can result in turbidity and other problems. Steam generators have experienced wall thinning (refs. [2]–[4]) as have carbon and steel tubes in feedwater heaters (ref. [5]), and tubes in steam condensers (ref. [6]). Steam separators, reheaters and steam piping have also experienced general corrosion, but more in the mode of flow accelerated corrosion (ref. [7]).

W-2220 MATERIALS

Materials and environment combinations have been categorized (ref. [1]) with regard to surface metal removal (generation) rates. Generally, corrosion at less than 0.005 in./yr (0.127 mm/yr) is considered reasonably low, depending on design specifics of the component being considered. Components requiring high tolerances in design such as valve seats and pump shafts will need more

corrosion resistance. In some cases, tanks, piping and valve bodies can operate with higher corrosion rates, when compensatory wall thickness is added in design. However corrosion rates at greater than 0.005 in./yr (0.127 mm/yr) should generally be avoided by judicious material selection and control of the environment.

Higher Cr, Mo, and Al content in alloys can serve to reduce corrosion (ref. [1]). Also, the use of small additions of Cu or Ni to carbon steel for added corrosion resistance is common, especially for applications with impure water. In cases of extreme general corrosion, stainless steel may be required; however, this can lead to other potential problems such as stress corrosion cracking in cases of relatively high oxygen water at temperatures greater than approximately 200°F (95°C) (ref. [8]). Other alloys that are considered to be passivated and resistant to general corrosion, similar to stainless steel, are alloys with greater than 30 to 40% Ni and greater than 8% Cr (ref. [1]), although these alloys may also experience problems in certain applications (ref. [6]). Materials may also be made resistant to corrosion by cladding, coating, and cathodic protection, where appropriate (ref. [9]).

Qualification of cladding procedures should be performed to demonstrate freedom from underclad cracking, when cladding is to be deposited on materials that are known to be susceptible to this form of degradation. Cladding may also be examined to ensure freedom from underclad cracking or adequate bond to the base metal as required for the specific application. Proper control of base metal dilution, as-deposited cladding chemistry, and ferrite content should also be considered to ensure adequate resistance to corrosion or SCC.

The life of equipment can sometimes be assessed based on simple immersion tests. However, at times the plant operational conditions can often lead to environments that may be more severe than those anticipated.

Additional information on materials requirements may be found in ref. [11].

W-2230 DESIGN

A number of general design rules should be followed, where possible, to avoid increasing the potential for corrosion problems (ref. [9]). Crevices and sharp notches should be avoided. Tanks should be designed for easy drainage, cleaning and replacement of critical parts that may experience degradation. Electrical contact should be avoided between dissimilar metals. Design should not include sharp bends in piping and residual strains due to hot or cold forming should be alleviated. Local heating and hot spots during operation should be prevented, air (oxygen and carbon dioxide) should be excluded where possible, and heterogeneities such as dissimilar metals should be avoided where possible. Long-term leakage and incompatibility with the environment can lead to bolting wastage. Other factors to consider in engineering include lowering the temperature and fluid velocity (although stagnant flow regions should be avoided) to

practical limits in order to reduce corrosion tendencies (ref. [9]). The analysis of water composition may also be performed in order to identify possible corrosion processes.

W-2240 MITIGATING ACTIONS

In addition to prudent material selection and design features, temperature and fluid conditions can also be controlled during operation. Water purity campaigns can also be implemented. Sulfates and halogens, which can exacerbate corrosion, should be avoided. Corrosion inhibitors (and oxygen scavengers) can be added, although this must be done carefully as other problems can arise. It has been learned (ref. [4]) that phosphates, such as sodium phosphate, can build up in sludge deposits and cause severe corrosion when added to secondary side water in steam generators. Proper pH and venting of CO₂ and H₂S gases, where applicable, can also mitigate corrosion (refs. [9] and [10]). Oxidizing environments during plant startup or extended downtime should be avoided when possible or the oxygen content should be minimized before temperature reaches the values critical for the initiation of damage. The system should be protected by an inert environment during extended outages, where practical.

W-2250 REFERENCES

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W-2300 PITTING CORROSION

W-2310 GENERAL DESCRIPTION

Pitting corrosion is a form of localized attack, with greater corrosion rates at some locations than at others (refs. [1]–[3]). Pitting can be very insidious and destructive, with sudden failures (especially in tubes) occurring by perforation (refs. [1] and [2]). This form of corrosion essentially produces “holes” of varying depth to diameter ratios in the materials. These pits are, in many cases, filled with oxide debris, especially for ferritic materials such as carbon steel. Deep pitting is more common with passive metals, such as austenitic stainless steels, than with non-passive materials (refs. [1] and [2]). Pits are generally elongated in the direction of gravity (refs. [1] and [2]). In many cases, corrosion-erosion, fretting corrosion, and crevice corrosion can also lead to pitting (ref. [1]).

Corrosion pitting is an anodic reaction, and is an autocatalytic process. That is, the corrosion processes within a pit produce conditions that stimulate the continuing activity of the pit (ref. [2]). Pits can first develop as a result of differential aeration cells (as within crevices or under sludge) and then convert to passive-active cells, where the pit is an anode and the surrounding surface is a cathode. High concentrations of impurity anions, such as chlorides and sulfates, tend to concentrate in the oxygen-depleted pit region, giving rise to concentrated aggressive solution in this zone (ref. [2]).

Pitting has been found on the outside diameter of steam generator tubes, where sludge or tube scale was present. This has been postulated to occur due to air exposure and chlorides, with aggravation by copper species, during both lay-ups and full power (refs. [4] and [5]). Pitting has also occurred in condenser tubes and other heat exchangers under deposits such as sludge (ref. [6]). It can also occur at locations of relatively stagnant coolant or water, such as in carbon steel pipes for service water lines, and at crevices in stainless steel, such as at the stainless steel cladding between vessel closure flanges.

W-2320 MATERIALS

Cu-Ni condenser tubes are generally preferred over brass for pitting resistance in brackish or polluted waters (ref. [1]). However, if the polluted waters contain sulfides, rapid attack may occur on Cu-Ni alloys (refs. [10] and [11]). Additions of iron (about 1.0–1.75 wt. %) and manganese (0.75 wt. % maximum) can help resist pitting in these tubes, as can optimizing the Cu-Ni balance (ref. [1]). Titanium tubes are considered to be quite resistant to pitting for condenser tubes (ref. [1]).

Austenitic stainless steels should be solution heat treated above 1800°F (980°C) and water quenched for the best resistance to pitting (ref. [2]). Type 316 is considered superior to Type 304 in pitting initiation resistance (because of added molybdenum). Low carbon grades are also more resistant to pitting corrosion than Type 304 or 316

in the non-solution anneal condition (ref. [2]). A rough surface finish should be avoided to minimize pitting (ref. [2]).

Ordinary carbon and low alloy steels can show less pitting than stainless steels in some applications, but their general corrosion rates may be prohibitive (ref. [2]). Adding small amounts of copper (about 0.30 wt. %) to carbon steel, or using high strength low alloy steels can increase resistance to pitting above that for plain carbon steel (ref. [9]).

Additional information on materials requirements may be found in ref. [12].

W-2330 DESIGN

Stagnant fluid conditions, possible during extended outages and in some designs, should be avoided (ref. [2]). Dead legs also result in stagnant fluid conditions, they too should be avoided. Crevices, which lead to such stagnant conditions, should also be avoided (ref. [1]).

Pitting is a strong function of oxygen level and chloride ion concentration for both carbon steels (ref. [8]) and steam generator and heat exchanger tubing materials (refs. [4] and [5]). Thus, the ingress of chlorides, air and copper should be minimized into steam generators and other systems (ref. [4]). In fact, heavy chlorination of water by chemical additives in shock treatment to combat MIC (micro biologically induced corrosion) should be evaluated for tendencies to lead to pitting (refs. [2], [3], and [9]). Sulfur ions can also exacerbate pitting (ref. [3]). Alternate wetting and drying can concentrate aggressive ion species, such as sulfates, chlorides, hydroxides and nitrates, which cause pitting in carbon steels, alloy steels, stainless steels and other alloys (refs. [1] and [2]). As a rule, temperatures should be kept as low as possible (ref. [6]).

W-2340 MITIGATING ACTIONS

A number of mitigating actions are possible, including those discussed above. Cathodic protection may be implemented, where practical (ref. [1]). In general, crevices and stagnant conditions should be avoided, as should oxygen and chlorides (ref. [1]). Operation should be at the lowest temperature possible (ref. [1]). Also, the surface can be cleaned periodically with alkaline cleaners (ref. [1]). It should be noted that one class of inhibitors, namely anodic inhibitors, such as chromates, can actually increase pitting intensity, in some cases (ref. [2]), and must be used with care, if at all.

Sludge and tube scale can be minimized by actions such as sludge lancing, flushing, or chemical leaching (ref. [4]). Mill scale on carbon steel can also exacerbate pitting and should be removed (ref. [9]).

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W-2400 CREVICE CORROSION AND DENTING

W-2410 GENERAL DESCRIPTION

Crevice corrosion is intense, localized corrosion within crevices or shielded areas (ref. [1]). It is associated with a small volume of stagnant solution caused by holes, gasket surfaces, lap joints, crevices under bolt heads, surface deposits, designed crevices for attaching thermal sleeves to safe-ends, and integral weld backing rings or back-up bars (refs. [1] and [2]). The crevice must be wide enough to permit liquid entry and narrow enough to maintain stagnant conditions, typically a few thousands of an inch or less (ref. [1]). Crevice corrosion is closely related to pitting corrosion and can initiate pits in many cases (ref. [3]), as well as lead to stress corrosion cracking (ref. [2]).

In an oxidizing environment, a crevice can set up a differential aeration cell and concentrate an acid solution within the crevice (refs. [1] and [2]). The differential aeration cell causes a difference in electrochemical potential, drawing anions (such as chlorides) to the oxygen depleted

zone in the crevice (ref. [1]). The dissolution of metal within the crevice leads to an excess positive charge, resulting in the migration of negative ions (anions, such as chlorides) to maintain electrical neutrality, thus further exacerbating the autocatalytic process (similar to pitting) (ref. [1]). Even in a reducing environment, alternate wetting and drying, as at steam generator tube/tubesheet or tube support crevices, can concentrate aggressive ionic species to cause pitting, crevice corrosion, denting, intergranular attack, or stress corrosion cracking (ref. [2]).

Denting is due to the crevice corrosion of tube support plate and tubesheet materials. When carbon steel corrodes under reducing conditions, magnetite is produced as a corrosion product that has approximately twice the volume of uncorroded carbon steel, and thus squeezes the tube outside diameter (ref. [4]). Such denting decreases the thermal efficiency of steam generators (ref. [4]) and can cause cracking of the tube.

Steam generators, condensers and feedwater heaters all experience similar problems with regard to crevice corrosion, especially between tubes and tube sheets or tube support plates (refs. [4]–[6]). Boiling water reactor (BWR) stainless steel piping and stainless steel and Alloy 600 safe-ends have shown crevice corrosion (ref. [7]). Bolting has a built-in crevice in every case, and thus is also considered with regard to crevice corrosion (ref. [8]).

W-2420 MATERIALS

All materials are prone to crevice corrosion to some extent, even nuclear grade stainless steel, depending on the environment and crevice (ref. [7]). Carbon steels and low alloy steels also can crevice corrode (ref. [2]). Thus, simply replacing a material with a more corrosion resistant material may not be sufficient, designs should be modified to avoid crevices where possible, and water chemistries should be controlled to mitigate aggressive environments. However, for given applications, certain materials are preferred in order to resist crevice corrosion.

Titanium tubing is generally very good at resisting crevice corrosion, Ni-Cu alloys are less resistant, and Type 304 or 400 Series stainless steels are very susceptible to attack at crevices (ref. [1]). Titanium and high nickel alloys (such as Hastelloy C) are very resistant, whereas Type 430 stainless steel is quite susceptible to crevice corrosion (Ref. [1]). High chromium stainless steel reduces the crevice corrosion of steam generator support plates, and thus mitigates the denting of tubes (ref. [4]). Titanium is generally better than high alloy stainless steels, which are better than copper alloys for condenser tube corrosion resistance (refs. [5] and [6]). However, depending on the particular service, copper alloys may be adequate, and the longer life of titanium tubes would not be needed.

For bolting applications, the material yield strength should be kept as low as possible, lubricants should not be aggressive (avoid sulfides, especially), and degreasing materials should be removed (ref. [5]), as discussed with regard to stress corrosion cracking mitigation. In all cases,

cold work and grinding can serve to exacerbate corrosion and serve as initiation sites, and should be minimized (ref. [7]).

Additional information on materials requirements may be found in ref. [8].

W-2430 DESIGN

In general, vessels and other components should be designed for complete drainage, and to avoid stagnant areas and sharp corners. The capability for washing should be included in the design, when practical, to prevent solids from settling on the bottom of the vessel in crevices or in other areas (ref. [1]). Vessels should be able to be inspected periodically and deposits removed.

Sound welds should be made with complete penetration; crevices should be closed in any lap joints; integral fit-up and weld back-up rings or bars should not be used (if avoidable), and weld back-up bars should be removed after welding (refs. [1] and [7]). Welding should be employed instead of rolling in tubes in tube sheets, where practical (ref. [1]). The contact area between the tube and support plate should be reduced, and seal welds should be employed for heat exchanger design and fabrication (refs. [4] and [6]). The most susceptible regions in these components are at the end of the rolled-in portion of a tube and tubesheet and at the intersection of the tube with the tubesheet (ref. [6]).

W-2440 MITIGATING ACTIONS

The foremost mitigating action is to avoid crevices and deposits, such as dirt, sludge, and corrosion products (refs. [1] and [4]). The corrosion of steam generator tube outside diameters is now generally controlled using hydrazine and ammonia to control pH (ref. [4]). Denting is also related to the ingress of chlorides and oxidants (such as oxidized copper) into steam generators (ref. [4]). In such cases, chlorides can be concentrated up to 60,000 times in crevices, and form an acid corrosive environment (ref. [4]). Therefore, impurity levels in feedwater should be reduced. Denting has been shown to be arrested by adding boric acid to secondary feedwater for steam generators (ref. [4]). In condensers, chlorides, hydrogen sulfide, and manganese dioxide have been found to accelerate corrosion, and should be controlled (ref. [6]). Ferrous sulfate can also reduce the corrosive attack of copper alloys in condensers (ref. [6]).

In general, operation temperatures should be reduced as low as possible, and any crevices should be flushed frequently (ref. [4]). Alloys should be kept clean and free of deposits, decaying marine life, sludge, etc. (ref. [6]). Optimum velocities (greater than about 4 ft/sec) should be maintained; cleaning, flushing, and drying should be performed for extended outages; and heat and biocide programs should be implemented to minimize slime, algae, bacteria and marine growth (ref. [6]).

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W-2500 INTERGRANULAR CORROSION ATTACK

W-2510 GENERAL DESCRIPTION

Intergranular corrosion attack (IGA) is localized corrosion at or adjacent to grain boundaries, with relatively little corrosion of grains (ref. [1]). It is caused by impurities in the grain boundaries, or the enrichment or depletion of alloying elements at grain boundaries, such as the depletion of chromium at austenitic stainless steel grain boundaries (ref. [1]). When the grain boundary layer is anodic to the grains, the layer is preferentially attacked; when the layer is cathodic to the grains, a narrow layer around the grains is attacked, producing the same effective result (ref. [2]).

IGA is very similar to intergranular stress corrosion cracking (IGSCC), except in IGA, there is widespread grain boundary attack and grain dropout, with no single crack. This attack can lead to a significant loss of strength and ductility, leading to failure (ref. [3]). Stress is not necessarily required for IGA (refs. [1], [3], [4], and [5]). Although susceptibility to IGA and susceptibility to IGSCC are generally linked, a microstructure highly susceptible to IGA can be resistant to IGSCC (ref. [6]).

Crevices are associated with IGA in many cases, as at tubes/tube sheets, safe-ends/thermal sleeves, and at crevices formed at pipe welds by incomplete penetration (ref. [4]).

Like IGSCC, the conditions that will produce IGA relate to a specific alloy in a specific condition being attacked by a specific corrodent. Test methods have been devised to measure the resistance of various alloys to IGA. ASTM test methods / practices include A262 (etching tests) G108 (Electrochemical Reaction) for austenitic stainless steels; G 67 (weight loss for wrought 5XXX alloys) and G110 (etching tests) for aluminum alloys; and G28 (weight loss) for Ni-rich, Cr-bearing alloys such as Alloy 600 (ref. [7]).

W-2520 MATERIALS

W-2521 Stainless Steels

When austenitic stainless steels are heated into, or slow cooled through, the temperature range of approximately 750 to 1,500°F (400 to 815°C) chromium carbides can be formed, thus depleting the grain boundaries of chromium and decreasing their corrosion resistance (refs. [1], [3], and [5]). This “sensitized” microstructure causes susceptibility to IGA. As sensitization is often associated with welding, this attack is often called “weld decay” (ref. [1]).

High chromium ferritic stainless steels, such as Type 430, also experience susceptibility to IGA (refs. [1], [3], and [5]).

Higher carbon (about 0.04 wt. % C or above) austenitic stainless steels have shown IGA when in a susceptible condition and exposed to an aggressive environment (refs. [4], [12], and [13]).

W-2522 Aluminum Alloys

Section II Part D allows for the use of some aluminum alloys. Of particular concern are the 5000 series of aluminum alloys that are allowed for Class 3 construction and are known to be susceptible to IGA, due to the formation of a continuous bend of $Mg_2 Al_3$ (ref. [7]).

W-2523 Nickel Alloys

Nickel alloys such as Alloy 600 experience IGA in the presence of certain sulfur environments (ref. [5]) or when austenitic stainless steel weld filler metal is inadvertently used on Ni-Cr-Fe alloys.

IGA has also been found in steam generators on the secondary side, in areas where impurities concentrated due to boiling under restricted flow such as the tube to tube-sheet crevice or underneath the sludge pile (ref. [8]). In this case, a de-aerated caustic solution is developed. The presence of sulfur also seems to be deleterious (ref. [8]).

Accelerated corrosion tests have also shown that lead contaminated PWR secondary water (ref. [9]) and chloride-based solutions (Ref. [10]) can also lead to IGA of Alloy 600.

It has also been demonstrated that Alloy 690, despite its improvements over Alloy 600, can still be susceptible to IGA in caustic environments depending on the level of aeration (ref. [11]).

W-2524 Other Alloys

As noted previously, virtually every alloy will be susceptible to IGA, depending on the environment. Therefore, the susceptibility of any alloy used in construction should be carefully considered with respect to the environment that the alloy will be exposed to. Microbiological corrosion, for instance, can influence IGA (ref. [6]).

W-2530 DESIGN

Crevice should be avoided in design and in fabrication (ref. [4]). Cold and hot forming should be limited or followed by solution heat treatment, unless tested to be shown resistant to IGA through procedure qualifications. Alternate wetting and drying, as in steam generator tube/tubesheet or the support plate crevices can concentrate aggressive species to cause IGA or other forms of corrosion (ref. [4]).

Water chemistry should be controlled to limit impurities such as chlorides, sulfur-related ions, phosphates and oxygen. Intrusion of sulfur-related ions during startup and extended outages, with oxidizing conditions, is to be avoided (ref. [4] and [13]).

W-2540 MITIGATING ACTIONS

Post-weld heat treatment (PWHT) usually is not performed for welds of austenitic stainless steel or Alloy 600 (ref. [1] and [3]) since it can result in sensitizing a susceptible material. Bimetallic welds, as in the case of joining an austenitic stainless steel to a ferritic piping material, can receive PWHT as a requirement for the ferritic material in some cases (ref. [11]). Care must be taken to either use a nonsusceptible material (ref. [12]) or to avoid PWHT in these cases. Welding heat input should also be kept to the lowest practical levels to mitigate residual stresses and sensitization (ref. [1]).

Alternate materials may be selected for resistance to IGA (efs. 12 and 15). Three main alternatives are considered when decreasing the susceptibility of austenitic stainless steels to IGA (and IGSCC): (1) solution heat treatment, (2) lower carbon content, and (3) adding elements to stabilize carbon (and prevent chromium carbide precipitation) (refs. [1] and [3]). Generally L-grade stainless steels with 0.03 wt. % C or less are considered (refs. [1] and [3]) quite resistant to IGA, although in some cases nuclear grade Type 316 or other alloys are preferred (refs. [12] and [15]) with 0.02 wt. % C or less. Elements such as Nb, Ta, or Ti are added to grades of austenitic stainless steel (Types 347 and 321) to stabilize the carbon (refs. [1] and [3]). It must be recognized though that even these materials are not immune to IGA when in an improper condition. A phenomenon known as “knife-line attack” can occur in stabilized grades when Nb (or Ti or Ta) and C dissolve at very high temperatures [as at 2,600°F (1 425°C), when welding] and stay in solution when water quenched or rapidly cooled. Upon reheating or slow cooling within the sensitization range [750°F to 1,500°F (400°C

to 815°C)] for austenitic stainless steels, the Nb stays in solution, but chromium carbides precipitate and deplete chromium from a narrow band immediately adjacent to the weld (ref. [1]). This depletion can occur during a subsequent post-weld heat treatment (PWHT), which is not advised for these materials (ref. [1]).

The heat input of welding should be kept to a minimum, consistent with producing high quality, sound weldments. Oily or greasy rags should not be employed to clean material prior to welding. Preservatives or lubricants containing high levels of halogens (especially chlorides) or low melting point constituents such as Pb, Zn, Sn, and Bi should be avoided. Sulfur ions, as introduced by thiosulfate, etc., are particularly detrimental to Alloy 600 and high nickel alloys (ref. [13]). Polyvinyl chloride (PVC) liners and other halogen-containing packaging or insulation should be minimized. When materials are pickled to remove heat treat scale, the components should be in the solution heat treated condition, and all pickling acid should be removed by rinsing with high purity water, such as demineralized water. Water chemistry should be controlled in all cases for material processing, hydrotests, and operation, with the goal of controlling the ions discussed above. Components stored or used near the sea-coast should be protected from the salt environment or washed frequently (ref. [5]).

Solution heat treatment, at approximately 1,800°F to 2,100°F (980°C to 1,150°C), followed by water quenching, is recommended (refs. [12] and [15]) for austenitic stainless steels in all practical cases, even for materials that are considered resistant to IGA. A maximum solution heat treatment temperature of 1,950°F (1,065°C) is sometimes specified for the stabilized grades to minimize “knife-line attack.” Solution annealing is prudent since factors such as crevices, weld repairs, grain size, cold work and grinding, trace impurities and the overall composition balance can impair IGA resistance (ref. [12]).

Crevices should be avoided where possible, as discussed above. Cold work and grinding should be minimized, or followed by solution heat treatment when practical. Tube/tubesheet welds can be sealed by welding or explosive expansion of the tube into the tubesheet below damaged regions (ref. [13]).

W-2550 REFERENCES

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W-2600 MICROBIOLOGICALLY-INDUCED CORROSION AND FOULING

W-2610 DESCRIPTION AND EXPERIENCE

Acceleration of corrosion of materials can occur as a result of the influence of microbiological activity. Sulfate reducing bacteria, sulfur oxidizers, and iron-oxidizing bacteria are most commonly associated with corrosion effects. Microbiologically-induced corrosion (MIC) most often results in pitting, followed by excessive deposition of corrosion products. Stagnant or low flow areas are most susceptible. Any system that uses untreated water, or is buried, is particularly susceptible. Consequences range from leakage to excessive ΔP and flow blockage. Essentially all systems and most commonly used materials are susceptible. Temperatures from about 50°F to 120°F

(10°C to 50°C) are most conducive to MIC. The experience of MIC in virtually all large industries is common. Nuclear experience is relatively new, but widespread occurrences are being reported (refs. [1]–[5]).

In particular, service water flow rates have been found (ref. [6]) below design values. This flow impediment was caused by fouling and partial blockage in sections of small diameter pipes supplying cooling water coolers (ref. [6]). Sedimentation aggravates the MIC problem in standby service water basins, as well as increasing pipe roughness and resistance to flow (ref. [6]). Slime-forming bacteria can produce a biofilm, which attract and hold suspended solids contained in the water (ref. [6]), and reduce the ability of heat exchanger tubes to transfer heat.

Fouling in heat exchangers has occurred in raw water systems and can decrease the ability to reject heat to the ultimate heat sink (ref. [7]). Flow rates through heat exchangers have been reduced to values below design values by a combination of organic and inorganic compounds that produce fouling (ref. [7]). Clams, mussels and debris from other shellfish have also been found to cause problems (ref. [7]). Affected heat exchangers include containment spray, component cooling water, and control room chiller heat exchangers (ref. [7]). Reactor coolant pump motor coolers may also be affected. Service water is used to cool the lube oil coolers for charging pump gearboxes (part of the emergency core cooling system), and fouling can result in overheating these pump motors (ref. [7]).

W-2620 MATERIALS

All materials commonly used in nuclear power plants are susceptible to MIC, with the possible exception of titanium and some high Ni-Cr-Mo alloys. Equipment is particularly vulnerable in the time period between the system or component hydrostatic test and operation. In addition, materials may be vulnerable in the entire construction phase and in all lay-up periods. Some systems, such as service water systems and tanks for standby-type systems, are susceptible throughout plant life. Typical treatments include mechanical cleaning, use of biocides, additions to increase pH to > 10 or 10.5, or procedural controls (periodic flow). Methods of detection include monitoring of plant parameters (e.g., service water flow and related heat exchanger performance), detection of small leaks (e.g., during repair activities), measurement of total organic carbon in water chemistry, or visual observation. Routine sampling of process streams for microbial activity is rare and generally ineffective. Few components have side stream sampling or coupon monitoring stations either (refs. [1] through [5]).

Additional information on materials requirements may be found in Ref. [8], with general information available on the subject of MIC in ref. [9].

W-2630 DESIGN

MIC does not now limit design, although design and operational remedies are often proposed to mitigate MIC concerns (elimination of stagnant areas, drain and dry provisions for all pipe runs, and requirements for periodic operation of pumps in all lines to assure at least some flow).

Piping system configurations should be designed to avoid deposits and low flow, through system flow balancing (ref. [6]). Flush and drain connections should be included in the design to facilitate routine future cleaning (ref. [6]). In some cases, larger diameter pipes may be appropriate to reduce flow resistance (refs. [6] and [7]). Intake screens of the proper size can also be employed to filter out suspended solids and silt (ref. [7]).

W-2640 MITIGATING ACTIONS

Water treatment is probably the most common mitigating step for prevention and treatment of MIC. Use of biocides (chlorine, hypochlorite, ozone, and hydrogen peroxide), agents to increase the pH of the system to > 10 or 10.5, and dispersants to break up deposits on metals are all used, often in combination. Note that such treatments, notably chloride additions, can result in potential problems, as discussed for other damage mechanisms. Mechanical cleaning is also used and generally mandatory to remove deposits so that the water treatment agent can get to the metal surface. Hydrolazing and flushing can be used to reduce blockage (ref. [6]). Procedural and design controls to eliminate, or at least minimize stagnant areas, are also commonly utilized. Valves are adjusted to increase flows in some cases (ref. [6]). Key components are sometimes coated with cement or epoxy following thorough cleaning to afford protection (ref. [1]). In severe environments, coatings may be inadequate to protect against MIC attack.

Flow monitoring instrumentation and trending information is part of the mitigative actions taken for MIC and fouling (ref. [6]). In general, water treatment (as discussed above) should be considered, including a general corrosion inhibitor, a biocide, and a dispersant (ref. [6]). Biofouling surveillance should be initiated in cases where this degradation mechanism may exist (ref. [7]).

W-2650 REFERENCES

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W-2700 CORROSION FATIGUE AND CRACK GROWTH

W-2710 DESCRIPTION AND EXPERIENCE

The behavior of materials under cyclic loading conditions is commonly considered as consisting of two broad categories of material properties. One category relates to the cyclic life for the formation of a fatigue crack in a smooth test specimen, the so-called S-N fatigue properties. The second relates to the growth of a preexisting crack. The fatigue design procedures of Section III are based on the S-N properties. Fatigue crack growth from preexisting flaws is a primary concern in Section XI's flaw evaluation procedures and material properties for crack growth analysis are given in Appendix A of that Section.

Laboratory tests have shown that LWR coolant water can have a detrimental effect on both S-N fatigue properties and fatigue crack growth. This Article focuses on the effects of LWR coolant water on the S-N fatigue behavior of pressure boundary materials since effects on fatigue crack growth are included in Appendix A of Section XI. The specific LWR environments of concern are the total circuit in a Boiling Water Reactor (BWR) plant and the primary circuit in a Pressurized Water Reactor (PWR) plant although secondary side water environments in a PWR system could affect the fatigue behavior of secondary side components.

To date, there has been no documented instance of a fatigue failure in an operating LWR plant where the primary cause of the failure could be ascribed to a reduction in S-N fatigue life due to LWR coolant environmental effects. However, since laboratory tests do show that a reduction in S-N life can occur under certain conditions, it is prudent to consider environmental effects in design. The test results suggest that environmental effects might occur in the secondary side of a PWR as well as in the primary side.

W-2720 MATERIALS AND CRITICAL PARAMETERS

Test results show that the magnitude of the environmental effect of LWR coolant water on S-N fatigue life is dependent on the combined effects of water chemistry and mechanical parameters. The main water chemistry factors are temperature and dissolved oxygen content and possibly water flow velocity. The principal mechanical parameters are strain amplitude and strain rate. The largest environmental effects are observed in tests that have a combination of high temperatures (up to nominal LWR coolant temperatures), high dissolved oxygen content, large strain amplitude, and low strain rate.

The test results also indicate that the S-N life of all of the commonly used pressure boundary structural materials is reduced to some degree for severe combinations of water chemistry and mechanical parameters. Materials that have been tested include carbon and low alloy steels, austenitic stainless steels, and high nickel alloys. Some tests have been performed on carbon and low alloy steel welds and the results indicate no greater or lesser effect than on the base metal for identical test conditions.

Test results for austenitic steels show that severe sensitization increases sensitivity to environmental effects. There are no definitive results on heat treatment effects for nickel alloys. Coolant water flow rate is an additional parameter that is believed to be an important factor that can affect the magnitude of environmental effect. In the case of fatigue crack growth tests, high flow rates have been found to reduce or minimize the environmental effect. Test results to validate this effect on S-N fatigue life are lacking but a threshold value derived from fatigue crack growth results has been assumed to also apply to S-N life.

W-2730 DESIGN

Analysis of the test results indicate only a moderate environmental reduction in S-N life when a threshold value for any one of the water chemistry or mechanical parameters is not transgressed. Careful designs that minimize transients that transgress these threshold values will reduce the environmental effects. The same considerations also apply to plant operating conditions. Reference [9] describes the results of applying an interim version of environmentally adjusted fatigue curves in the fatigue reevaluations of design transients in operating plants. It was concluded that when conservative assumptions were removed and the anticipated numbers of cycles were used, the cumulative usage factor (CUF) could be reduced to below 1.0 for most components using the interim fatigue environmentally adjusted curves in both older and newer vintage plants.

A detailed analysis methodology that could be implemented in Section III design to determine adjustments for LWR environmental effects has been proposed in the

form of a nonmandatory Appendix. The major elements of this methodology are described in ref. [10] and may possibly be considered for Code implementation.

W-2740 MITIGATING ACTIONS

The major mitigating actions are the design precautions described in W-2730 and careful attention in plant operation to avoid exposure to conditions exceeding the threshold values. There appears to be no easy material selection or fabrication steps that can be used to mitigate the environmental effects on S-N fatigue for currently used types of pressure boundary materials.

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W-2800 FLOW ACCELERATED CORROSION

W-2810 GENERAL DESCRIPTION

Flow accelerated corrosion (FAC) is characterized by a localized increase in the rate of dissolution of a material as a result of a flowing corrosive environment. This accelerated corrosion is the result of the continuous or cyclic localized removal of the protective oxide film. This phenomena is also referred to as flow assisted corrosion or erosion corrosion. This phenomena differs from erosion, which is a mechanical removal of base material by the fluid.

Cavitation is normally associated with erosion, but may be a factor in FAC. Cavitation is localized damage produced by the collapse of steam bubbles on a material surface. Cavitation bubbles form in the flowing single phase high energy water as a result of a localized sharp pressure drop such as downstream of a control valve. The pressure drop causes vapor bubbles to form. The resulting bubbles coalesce on the material surface and then collapse, imparting large forces in a very small area. Vigorous localized flow can accelerate corrosion dissolution of the oxide film, in the case of FAC, or mechanically deform and remove oxide and base material, in the case of erosion (ref. [1]).

In nuclear power plants, FAC typically occurs in carbon steel piping systems containing flowing water or wet steam. It does not occur in dry steam since liquid water is necessary for the corrosion process to occur. The rate of flow accelerated corrosion is influenced by a complex interaction between a number of variables including material composition, temperature, steam quality, pH, oxygen content, fluid velocity, and geometry (refs. [2] through [4]).

Because FAC causes a gradual thinning in a fairly localized area, the carbon steel pipe has a tendency to rupture when the thinned wall can not support a pressure transient rather than develop a small leak. This has resulted in a number of ruptures of piping carrying high energy single phase water or two phase steam, with accompanying loss of life, injuries, and equipment damage (refs. [5] through [8]).

W-2820 MATERIALS

The stability of the protective oxide film, which forms at the interface between the material and the fluid, is a strong function of the material composition. Alloying elements such as chromium, copper, and molybdenum greatly increase the resistance of the oxide film, with chromium having the largest effect.

Stainless steels with 12% or greater chromium content are virtually immune to FAC. Alloy steels with on the order of 1% to 2% chromium can exhibit a factor of 4 to a factor of 10 or more reduction in FAC rate over plain carbon steel. Even trace levels of chromium, on the order of 0.1% can have a significant effect (ref. [4]).

Carbon steel, such as A106 Gr. B, typically used in PWR secondary systems or equivalent BWR systems is procured without a specification for chromium or alloying elements other than carbon. Depending on the manufacturing method, a range of alloying element levels could be present. Examination of 38 heats of A106 Gr. B material in one survey revealed a range in chromium concentration from 0.03% to 0.28%, with a range of 0.01% to 0.06% for molybdenum and 0.06% to 0.34% for copper. A factor of 16 difference in FAC rate was estimated for this range of compositions (ref. [9]).

W-2830 ENVIRONMENT

In stagnant or low flow oxygenated water environments carbon steel will corrode and form an oxide film at the interface between the water and the metal. This film acts as a protective barrier against further corrosion. As the fluid velocity is increased, local turbulent regions are created, especially at geometric discontinuities. This increases the corrosion rate by increasing the mass transfer of soluble corrosion products from the vicinity of the turbulence. The protective film becomes thinner or is dissolved, allowing the base material to continue to corrode. In either case, additional base metal is consumed in the corrosion process and the material becomes progressively thinner in a highly localized region (ref. [4]).

The adherence of the oxide film is influenced by the pH, oxygen level, and temperature of the water, as well as the material influence previously mentioned. Changes in these three variables affect the solubility of the oxide, the structure of the oxide, and the rate the corrosion reaction. The effects of changes in these parameters are interrelated and are not linear. General trends are indicated below.

Increasing pH, in the range from 7 to 10, decreases the solubility of the oxide and decreases the rate of FAC. An order of magnitude decrease in FAC rate was observed when pH was increased from 8.0 to 9.0 in laboratory tests (ref. [10]).

Increasing oxygen concentration above a threshold of 10 ppb to 20 ppb reduces the rate of FAC. This effect plateaus above 100 ppb to 300 ppb. The exact values of the threshold and plateaus are temperature and pH dependent. Laboratory tests have shown a factor of 100 decrease in the rate of FAC over a range of oxygen concentration from 1 ppb to 200 ppb (refs. [4], [9], [11], and [12]).

Increasing temperature initially increases and then decreases the rate of FAC. This is a result of competing influences of increased reaction rates, changes in solubility, and changes in the oxide film structure. The temperature associated with the peak FAC rate is approximately 250°F to 350°F (120°C to 175°C) and can vary significantly depending on whether the water is single or two phase and other parameters discussed above (ref. [4]).

W-2840 DESIGN

The geometry of piping and components has a significant effect on the location and rate of FAC. Sharp changes in direction, reducers, expanders, tees, and orifices cause turbulence and eddy in the flow. This can result in much higher local velocities than would be indicated by the bulk fluid velocity. The high local velocities increase mass transfer of soluble corrosion products. In addition, the flow may impinge directly on the surface. Rating systems have been developed to qualitatively assess various geometries (refs. [4] and [13]).

It is not practical to design piping systems without changes in direction or other geometric features. However, the designer should minimize these features where practical and consider resistant materials in areas where they are required and the other conditions of the system indicate potential susceptibility to FAC.

Materials are available that are resistant to FAC. In selection of a material, the designer must consider the level of resistance required in comparison with the service life of a component. For example, a 2% chromium alloy steel may provide sufficient resistance for a given application and the additional resistance provided by stainless steel may not be required. In addition, the designer must consider any increased costs, fabrication complexity, pre and post weld heat treatments, and susceptibility to other damage mechanisms in selection of a material. Carbon steel may provide adequate service in a given application provided the other variables affecting FAC are controlled and appropriate in-service inspections are performed.

Various water chemistry control protocols have been developed to reduce FAC in PWR secondary systems. These include intentional injection of oxygen and increasing pH (refs. [9] and [12]). A number of amines have been examined to control pH. Of course, effects on other corrosion mechanisms must be considered.

The amine characteristics will determine the relative concentrations in the liquid and vapor phases and will influence the pH in various systems around the steam cycle. For instance, morpholine has better high temperature and separation characteristics than ammonia. In an extraction line, this would result in a relative increase in the pH.

W-2850 MITIGATING ACTIONS

The mitigation of FAC is not always straight forward, due to the ramifications of modifying some of the causative parameters. For example, oxygen injection may not be desirable due to the potential for increased intergranular stress corrosion cracking in a BWR and increased steam generator tube corrosion in a PWR. Neutral pH is required for operation of a BWR, while modification of the pH to the higher levels in a PWR may impact condensate polisher operation and the corrosion of system components containing copper.

Material changes to a less susceptible material, and re-design of piping to reduce velocities and avoid geometric discontinuities can alleviate the FAC problem. However, these solutions entail increased material and fabrication costs, potential increased complexity, and may create susceptibility to other damage mechanisms.

Because of the complex interaction between the variables influencing the rate of flow accelerated corrosion and the difficulty in ascertaining the condition of each variable at all locations of potentially susceptible piping systems, prediction of high wear locations and rates accurately is quite difficult.

Therefore, periodic inspection of components in susceptible systems is essential. A number of predictive models are available to assist in the identification of components to be inspected. However, since it is rare that all information is known about every component in a system, care should be used in applying any predictive model. For example, a sufficient number of locations should be examined to insure that differences in material composition do not mask a potential problem.

W-2860 REFERENCES

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W-2900 EROSION AND EROSION-CORROSION

W-2910 GENERAL DESCRIPTION

The classical metallurgical definition of erosion is the destruction of metals or other materials by the abrasive action of moving fluids, usually accelerated by the presence of solid particles or matter in suspension. When corrosion occurs simultaneously, the term erosion-corrosion (or corrosion erosion) is often used (ref. [1]). More simply stated, erosion is the mechanical abrasion by solids suspended in a fluid (ref. [2]).

Frequently, components fail in service due to the combined effects of mechanical or hydraulic factors and corrosion. These failure mechanisms fall into three types: stress-corrosion, corrosion-fatigue, and liquid velocity effects (corrosion-erosion and cavitation). Flynn/Trojan (ref. [3]) provide their definition for the two terms associated with liquid velocity effects.

(a) *Erosion* — the wearing away of a surface because of the combination of mechanical and corrosive effects.

(b) *Cavitation* — stress corrosion induced by the local collapse of vapor bubbles against a surface such as a pump impeller.

Erosion is also considered low-stress scratching abrasion where the stress on the abrading particles does not exceed their crushing strength (ref. [4]).

Thus, erosion and erosion-corrosion may involve the loss of metal where it is exposed to mixing fluids. Factors affecting its extent will briefly be discussed in the following paragraphs on materials (W-2920), environment (W-2930), and design (W-2940).

W-2920 MATERIALS

There are no published lists of materials that are resistant to all aspects of erosion or erosion-corrosion. Engineering judgment must be used to select materials that may be subjected to conditions leading to such damage in service. The ASM Metals Handbook (ref. [4]) does have a useful section on "Damage Resistance of Metals." Some of the more helpful points include:

(a) ASTM G32, "Standard Method for Vibratory Cavitation Erosion Test," may be used to screen materials for a particular service.

(b) Various materials properties such as hardness, true stress at fracture, strain energy to fracture, work-hardening rate, and “ultimate resilience” are some indicators of resistance to erosion damage. The higher these values for a given material, the more it will be able to resist erosion.

(c) Thermal treatments that increase toughness will generally improve erosion resistance.

(d) Fine grain size and fine dispersion of hard second-phase particles both enhance erosion resistance.

(e) A classification of 22 alloys is provided (in ref. [4]) showing normalized erosion resistance relative to 18CR-8Ni austenitic stainless steel with a hardness of 170 dph (which is equivalent to a Brinell hardness of 162 or a tensile strength of 79 ksi (543 MPa), which is typical for Type 304 stainless steel commonly used in Code construction).

In general, some of the strongest and tougher steels, precipitation hardened stainless steels and nickel-base alloys, and cobalt-based materials would be good choices when erosion conditions are expected. The weaker and softer materials obviously would be expected to “wear” faster.

W-2930 ENVIRONMENT

Environment from the standpoint of erosion and erosion-corrosion, must be considered to include the chemistry of the interacting fluid, the fluid velocity, temperature of the fluid and the base metal, and the nature of other particulate substances that might be present in the fluid.

Erosion can occur in the absence of corrosion, but case histories clearly show that corrosion can generally accelerate the erosion process. It is when corrosion is a factor that the erosion process becomes an erosion-corrosion problem. Factors affecting erosion and erosion-corrosion are frequently “a given” once a plant or new component is conceived. Thus materials must be properly selected to deal with those conditions — or some of the conditions may be dealt with by other design features. The very important message here is to thoroughly determine the total environment that will be experienced during the expected life of the component or plant. The word “total” is used since there are conditions during startup testing or post-maintenance testing that may differ from those expected during normal operation. There are also static conditions (no fluid flow) during hydrostatic testing or plant lay-up that can profoundly affect the subsequent corrosion resistance of materials. All of these factors of environment must be considered as the design process progresses.

W-2940 DESIGN

The design process provides the opportunity to affect many of the conditions that lead to erosion or erosion-corrosion. Since the rate of “wear” or erosion is affected by fluid velocity, the designer should explore suitable ways to reduce the velocity of impinging fluids. Where

cavitation is a concern, the designer needs to reduce the hydrodynamic intensity by increasing the radius of the flow path or by removing surface discontinuities. Both of these parameter changes can reduce the probability of cavitation.

When fluids are known to contain particulates, baffles can be strategically placed to redirect flows to areas less prone to damage or where damage can be better tolerated.

The designer must have complete knowledge of the environmental conditions so that the most erosion or erosion-corrosion resistant materials can be specified. The issues involved in materials selection were discussed earlier in W-2920, and this is generally the only way that the designer can deal with the “environment,” since that is usually a “given” condition.

W-2950 MITIGATION ACTIONS

Much of the earlier discussion on materials, environment, and design inferred the mitigative actions for erosion and erosion-corrosion control are taken by the supplier of a component or system. In many instances, modifications must be made after some period of operation — corrective actions to address what was either overlooked or not known during the design phase.

Reference [4], under “Prevention of Erosion Damage” cites that damage of the erosion-type can be prevented or minimized by:

(a) reducing the intensity of cavitation or liquid impingement;

(b) using erosion-resistant metals; or, under certain circumstances, using elastomeric coatings.

The protection afforded by these coatings is brought about by their ability to absorb large amounts of energy by elastic deflection. In the use of erosion-resistant materials, consideration needs to extend beyond the base metal through application of weld overlay or other surface altering treatments (see W-4100 on Fretting and Wear). Whenever a new substance is introduced into the system, whether it is a different metallic coating or an elastomeric coating, some thought needs to be given to the problems that might be experienced if the protective coating comes loose and enters the system. Of course, equal thought also has to be given to the effects of the base metal that wears away and now moves about the system. A thorough assessment of erosion and erosion-corrosion, like many other failure or deterioration concerns, is not an easy task, and is one that must be taken seriously.

W-2960 REFERENCES

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- [3] Flinn, R. A., and Trojan, P. K. *Engineering Materials and Their Applications*, Houghton Mifflin Co., 1986.
- [4] ASM Metals Handbook, 8th Edition, Volume 10, *Failure Analysis and Prevention*, 1986.

ARTICLE W-3000

SUMMARIES OF EMBRITTLEMENT DAMAGE MECHANISMS

W-3100 IRRADIATION-ASSISTED STRESS CORROSION CRACKING (IASCC)

W-3110 GENERAL DESCRIPTION

Environmentally induced intergranular cracking of stainless steel and nickel-base alloys exposed to the radiation and high-temperature aqueous environment of a light water reactor (LWR) is known as irradiation-assisted stress corrosion cracking (IASCC). IASCC does not require the presence of a sensitized microstructure or high tensile stresses (refs. [1], [2], and [3]). The IASCC phenomena has been observed in some boiling water reactor (BWR) in-core components which have been exposed to fluences of above 5×10^{20} n/cm² (> 1 MeV) (refs. [1], [2], and [5]). In PWR systems, the environmental cracking of irradiated stainless steels was observed when the neutron fluence was higher than 2×10^{21} n/cm² (ref. [8]). Components which have experienced IASCC include neutron source holders, control rod absorber tubes, and incore instrument tubes. With sufficient exposure core support structures may become susceptible to IASCC as well (ref. [4]). The cause of cracking is thought to be due to radiation, water chemistry, and neutron induced diffusion of elements (Cr) and impurities (S, Si, P) preferentially to the grain boundaries in oxidizing high temperature water environments (ref. [5]). Stress from service loads or fabrication processes is a contributing factor.

W-3120 MATERIALS

Not enough is known about the IASCC phenomenon to positively identify which austenitic stainless steels may be most susceptible. However, Cr depletion is believed to be an important damage mechanism. Since impurity elements are believed to play a role in the IASCC mechanism, these elements should be minimized in high neutron irradiation applicants (refs. [1] and [5]). Annealing of components following welding to reduce the possibly synergistic effects of sensitization and residual stress should be considered as well.

W-3130 DESIGN

As IASCC appears to be a strong function of neutron irradiation fluence levels, actions which can be taken to reduce exposure should be considered. Such actions include water gaps between the core and structural elements and design for replacement of in-core components. A threshold irradiation of 5×10^{20} n/cm² (> 1 MeV) is generally accepted as the minimum fluence necessary for

susceptibility to IASCC (refs. [1] and [2]). Service loads and fabrication residual stresses should be minimized for high fluence components, and features, which contribute to intergranular stress corrosion cracking in the absence of neutron irradiation (such as sensitization, surface cold work, and crevices), should be minimized.

W-3140 MITIGATING ACTIONS

Stringent water environment controls will reduce the susceptibility to IASCC of stainless steel components. IASCC susceptibility of 304 and 316 austenitic stainless steels decreases with a decrease in the dissolved oxygen content of the water (refs. [1], [2], and [6]). Water chemistry controls that ensure low dissolved oxygen content could mitigate IASCC in relatively high fluence regions (refs. [2] and [6]). Hydrogen injection, normally used to reduce electrochemical potential as a preventive measure against intergranular stress corrosion cracking (IGSCC) in boiling water reactor (BWR) recirculation piping system, may also mitigate IASCC (ref. [1]). Experimental results have shown that, for irradiated Type 304 stainless steel, the protection potential is ~ 0.210 V_{SHE} (refs. [2] and [7]).

W-3150 REFERENCES

- [1] "ASM Handbook, Volume 13, Corrosion," December 1992, pp. 935-936.
- [2] Indig, M. E., et al. Investigation of Protection Potential Against IASCC. *Proceedings: Fifth International Symposium on Environmental Degradation of Materials in Nuclear Power Systems-Water Reactors*. 1991 August 25-29; Monterey, California: American Nuclear Society, Inc.: pp. 936-939.
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- [4] Gerber, T. L., et al. "Evaluation of BWR Top Guide Integrity," EPRI NP-4767, November 1986.
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- [7] “Advanced Light Water Reactor Utility Requirements Document,” Volume II, ALWR Evolutionary Plant, Chapter 1: Overall Requirements, EPRI NP-6780-L, December 1993, pp. 1.5-1 to 1.5-40.
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- [9] Stultz, S. C., and Kitto, J. B. *Steam*, 40th Edition, Babcock and Wilcox, 1992.

W-3200 THERMAL AGING EMBRITTLEMENT

W-3210 GENERAL DESCRIPTION

Thermal aging embrittlement refers to the loss in notch impact properties and fracture toughness of materials as a result of long term exposure to elevated temperatures. The specific temperatures at which reductions can occur is material dependent including factors such as alloy type and grade, heat treatment, and fabrication procedures. The consequence of the embrittlement is an increase in the risk of nonductile fracture failure. The embrittlement can be manifested as a reduction in toughness at the exposure temperature and at temperatures below the exposure temperatures. The embrittlement may be accompanied by changes in other properties such as yield and tensile strength and ductility. The embrittlement that occurs is generally defined as that due to the singular effect of temperature only and apart from the additional effects of other environmental factors such as sustained or cyclic stresses and strains, neutron radiation, and aggressive chemical attack.

The extent of embrittlement is measured by pre- and post-exposure testing. The Charpy impact test is often used but fracture mechanics toughness test data are more useful and applicable for quantitative evaluation of embrittlement effects.

W-3220 MATERIALS AND CRITICAL PARAMETERS

The normal upper limit of operating service temperatures for pressure boundary materials in light-water reactors (LWRs) is generally about 700°F (370°C). A practical aspect of identifying and quantitatively determining thermal aging effects is the long testing times required to obtain data useful for assessing the effects of exposures comparable to typical service lives. Experimentally, exposures at higher temperatures to compensate for shorter

test times are often utilized and the results are equated for long service times through empirical or theoretical time-temperature equivalence models.

Within these experimental limitations, test results for carbon and low alloy steels typically used for LWR piping and reactor vessels show that exposures within the range of LWR temperatures and service lifetime would be expected to have only small or negligible embrittlement, (refs. [1] through [4]). This appears to be particularly true of low alloy steel components that have had a postweld heat treatment (PWHT) during fabrication. The inservice exposure has very little additional thermal aging effect relative to the effect of the PWHT. Exceptions to this lack of embrittlement sensitivity may occur in the weld heat affected zone (HAZ) of the lower alloy grades of steels and in the base materials of some grades of higher strength, higher alloy content steels such as SA 508, Class 4. The increased sensitivity can be due to temper embrittlement that can occur at temperatures beginning at about 600°F (315°C) and extending up to about 1,000°F (540°C). Susceptibility to this type of embrittlement is increased with the presence of impurity elements such as phosphorous and tin in the steel. Another exception to the low sensitivity to thermal aging embrittlement of carbon and low alloy steels at LWR operating temperatures may occur when cold work has been introduced into the component during fabrication. Subsequent thermal exposures can cause embrittlement due to a phenomenon termed “strain age embrittlement.” Code rules recognize this possibility and limit the amount of cold straining that can be performed during fabrication.

The sensitivity of austenitic stainless steels to thermal aging embrittlement is quite complex depending on product form (wrought vs. cast) alloy content, and metallurgical factors. The standard Type 300 wrought alloys such as the 304, 316, and 347 grades seem to be basically insensitive to embrittlement at LWR temperatures as indicated in Refs. [5] and [6]. However, at higher temperatures, embrittlement due to sigma phase can occur as described in Article A-340, Appendix A, Section II, Part D of the Code.

The cast grades, the weld metals, and other duplex microstructure grades of austenitic stainless steels may exhibit thermal aging embrittlement at LWR temperatures. The responsible metallurgical factor is the presence of the ferrite phase in the microstructure and the sensitivity is primarily dependent on the amount of ferrite with a secondary effect from carbon content and the amount of additional alloying elements. Precautionary guidelines defining the effect of ferrite content on thermal aging embrittlement in austenitic and austenitic-ferritic stainless steels are given in Article A-360, Appendix A, Section II, Part D of the Code. Additional details about the embrittlement mechanisms, data, and analyses can be found in refs. [5] through [10].

The available information indicates that the non-hardenable grades of wrought nickel-base and nickel-chromium-iron-base alloys do not exhibit thermal aging

embrittlement at LWR temperatures (refs. [5] and [6]). There appears to be no data regarding the embrittlement susceptibility of weld metals used to weld high nickel alloys. Also, there are no data regarding the sensitivity of the cast grades of high nickel alloys but their usage for LWR components is relatively minor.

The age hardenable grades of nickel-base alloys are susceptible to thermal aging embrittlement at temperatures higher than the upper limit of LWR temperatures but not at LWR temperatures.

W-3230 DESIGN

Several material selection and processing considerations can be utilized to minimize thermal aging embrittlement. In the case of carbon and low alloy steels, fabrication procedures that minimize cold straining should be specified. For the case of wrought austenitic stainless steel products, the use of low carbon and L grades combined with low ferrite content welds will minimize embrittlement susceptibility. Cast austenitic steels having low carbon and ferrite contents consistent with strength requirements should be used to minimize embrittlement. Careful consideration should be given to the use of duplex microstructure wrought stainless steels to ensure that they will not be exposed to embrittling temperature. In particular, the designer should be fully aware of the notes accompanying the allowable stresses in Section II, Part D citing sensitivity to embrittlement.

W-3240 MITIGATING ACTIONS

The principal mitigating actions are in utilizing the design precautions listed in W-3230. Some mitigation can be obtained in specific instances by decreasing the operating temperature of the component.

W-3250 REFERENCES

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W-3300 RADIATION EMBRITTLEMENT

This section is under development.

W-3400 HYDROGEN DAMAGE EMBRITTLEMENT AND DELAYED CRACKING

W-3410 GENERAL DESCRIPTION

Hydrogen damage or embrittlement is a mechanical-environmental failure mechanism resulting from the initial presence or absorption of excessive amounts of hydrogen in metallic materials, generally associated with significant residual or applied tensile stresses. This type of cracking is usually restricted to high strength steels and certain other high-strength alloys. Hydrogen cracking may occur at the tip of pre-existing cracks or at subsurface sites where triaxial stresses are the highest. When the critical stress is exceeded, a crack initiates and propagates through the region of high hydrogen concentration.

The following are specific types of hydrogen damage, whose occurrence may be limited to certain materials or alloy systems:

- (a) cracking from hydrogen charging
- (b) hydrogen-induced blistering
- (c) hydrogen-induced cracking from decarburization
- (d) cracking from hydrogen-induced slow strain rate embrittlement
- (e) hydrogen-induced cracking from static fatigue
- (f) cracking from hydride formation
- (g) cracking from exposure to molecular hydrogen gas
- (h) cracking from exposure to hydrogen sulfide
- (i) cracking from exposure to water and dilute aqueous solutions

Each of the nine types of hydrogen damage is described in considerable detail in Volume 10 of the *Metals Handbook* (ref. [1]). The role of hydrogen in various power industry applications is also extensively described in a series of ten papers published in 1964 (ref. [2]).

Embrittlement by hydrogen damage manifests itself as a decrease in tensile ductility (percent elongation and reduction of area), decrease in notched tensile strength, and delayed failure by fracture under static loading. In metals that have high notch sensitivity, the extent of crack growth is usually quite small, and the probability of detecting a crack before complete failure occurs is correspondingly small. Delay in fracture apparently results, because of the time required for hydrogen to diffuse to specific area near a crack nucleus until the concentration reaches a damaging level (ref. [3]). Another mechanism is the interaction between hydrogen atoms and dislocations in the metal. The crack tip may be embrittled by hydrogen atoms from the aqueous reactions.

W-3420 MATERIALS

Carbon and low alloy steels are subject to eight of the hydrogen damage mechanisms described above (not from hydriding), particularly when tensile strengths exceed about 150 ksi (1 000 MPa). Failure by hydrogen damage (except by blistering or decarburization) is seldom a problem in those materials whose tensile strengths are below 100 ksi (700 MPa). When corrosion occurs in aqueous systems involving carbon steel, hydrogen is generated and may react with the carbon to form methane gas. This leads to localized decarburization with corresponding weakening of the metal. The methane collects at grain boundaries and other discontinuities with the metal. As the gas pressure builds, small fissures are formed, eventually resulting in a through-wall failure (ref. [4]).

Austenitic stainless steels are almost completely resistant to failure by hydrogen damage. These face-centered cubic structures are relatively impermeable to the diffusion of atomic hydrogen. The resulting low hydrogen content in the metal lattice thus has little impact on the material's ductility.

Ferritic stainless steels, in the annealed condition, are extremely resistant to hydrogen damage, because of their low hardness or strength levels. When these same materials are hardened by cold working or when they are used in the as-welded condition, they are susceptible to hydrogen damage.

Martensitic and precipitation-hardened steels are susceptible to hydrogen damage. As the yield strengths increase, the propensity for hydrogen-induced cracking is increased. Almost any corrosive environment capable of evolving hydrogen can cause cracking in these materials.

Heat-resisting nickel-base alloys generally do not experience hydrogen damage. However, there have been isolated cases with alloys such as N07718 where some

damage was experienced under conditions of pure hydrogen, pressure as high as 5000 psi (35 MPa), and a temperature of 1,250°F (675°C).

Aluminum alloys occasionally experience hydrogen damage, but the problems are generally traced to voids in ingots that contained hydrogen gas prior to working. Hydrogen damage has not been considered an industrial problem with this alloy family.

Titanium alloys can experience hydrogen damage (embrittlement) due to absorbed hydrogen. The hydrogen may originate from water vapor, pickling acids, or hydrocarbons. The amount of hydrogen absorbed depends primarily on the titanium oxide film on the titanium surface, and an adherent unbroken film can significantly retard hydrogen absorption. Hydrogen can be removed from titanium alloys by vacuum annealing.

Zirconium alloys can pick up substantial amounts of hydrogen during exposure to high-pressure steam. Like titanium alloys, the hydrogen can be removed by vacuum annealing.

W-3430 DESIGN LIMITATIONS

Design issues for hydrogen embrittlement are addressed under W-3440, Mitigating Actions.

W-3440 MITIGATING ACTIONS

The *Metals Handbook* (ref. [1]) advises, under a paragraph entitled, "Prevention of Failures:"

"... the effectiveness of a proposed change in part design or material in preventing hydrogen damage can be determined, and any possible harmful effects on the changed part can be detected, by processing sample parts using modified procedures and evaluating the reaction of the parts under simulated or actual service conditions."

The following are possible steps to be taken to mitigate hydrogen damage.

(a) Alloy changes, particularly using a lower strength material, may eliminate the hydrogen embrittlement problem.

(b) Lowering stresses will lower the propensity for cracking. This may be accomplished by annealing (lowering the material strength), thickening the section (lowering applied stress), or reducing design loads.

(c) Design changes may involve elimination of sharp corners or other stress raisers. Eliminating sites for crevice corrosion (where hydrogen might be generated) is also helpful.

(d) Heat treatments, such as tempering of high-strength steels, lower the strength level and result in tougher, more hydrogen damage resistant materials.

(e) Surface treatments that result in compressive residual stresses also improve resistance to hydrogen-induced cracking in normally susceptible materials.

(f) Coatings can be applied to susceptible materials, providing they are compatible with the service environment and they are applied such that no hydrogen is generated.

W-3450 REFERENCES

- [1] *Metals Handbook*, Volume 10, "Failure Analysis and Prevention," 8th Edition, 1975, American Society for Metals, pp. 230–240.
- [2] Transactions of the ASME Journal of Engineering for Power, July 1964, pp. 299–352.
- [3] Uhlig, H. H. *Corrosion and Corrosion Control*, J. Wiley & Sons, New York, 1964, p. 123.
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ARTICLE W-4000

SUMMARIES OF OTHER DAMAGE MECHANISMS

W-4100 FRETTING AND WEAR

W-4110 GENERAL DESCRIPTION

Fretting is a wear phenomenon that occurs between two mating surfaces. Fretting generally occurs when two tight-fitting surfaces are subjected to a cyclic motion of extremely small amplitude (as in vibration). The term *fretting* covers numerous forms of deterioration including fretting corrosion, false brinelling, friction oxidation, chafing fatigue, molecular attrition, and wear oxidation. Fundamentally, the fretting process includes

- (a) initial adhesion
- (b) oscillation accompanied by the generation of debris
- (c) fatigue and wear in the region of contact.

Wear is the damage to a solid surface caused by removal or displacement of material by the mechanical action of another solid, a liquid, or a gas (or various combinations). All mechanical components that undergo sliding or rolling contact are subject to some degree of wear. Wear may range from mild polishing over a long period of time to rapid and severe removal of material with accompanying surface roughening. There are numerous wear modes and they may change in service as a component wears.

Adhesive wear occurs generally under non-lubricated conditions when both contacting surfaces are metallic. It is also known as scoring, galling, seizing, or scuffing. Microscopic projections from the mating surfaces bond at the sliding interface under very high local pressure. As the bonds are broken, material may be torn from one surface or loose particles may be formed that then attribute to abrasive wear.

Abrasive wear occurs when hard particles of some origin slide or roll under pressure across a surface, cutting grooves in the surface. Both of the mating sliding surfaces may wear, or the particles may become embedded in one of the surfaces, causing abrasive wear to the mating surface. Abrasive wear may be grinding abrasion, or low-stress scratching abrasion.

Corrosive wear is a form of abrasive wear in which chemical or electrochemical reactions accelerate the metal loss between mating surfaces where sliding occurs. In this mode of wear, it is unclear whether mechanical wear precedes chemical actions or vice versa.

Surface fatigue is another mode of wear in which particles of metal are detached from a surface under high cyclic contact stresses, causing pitting and spalling.

Two additional modes of wear — erosive wear and erosion-corrosion — are covered as unique damage mechanisms in [W-2900](#) of this Appendix.

W-4120 MATERIALS

Volume 10 of the *Metals Handbook* (ref. [\[1\]](#)) provides numerous case histories of fretting and wear failures. These failures occurred for a variety of reasons including initial base metal selection, heat treatment of the material, or alteration of the surface. Surface finish can also be a factor.

W-4130 DESIGN

Fretting and wear failures can be brought about by design factors that allow for relative motion between mating surfaces, allow high stresses in areas that must move with respect to one another, or allow for the entry of aggressive species into mating surfaces. Many of these causative factors are described in the case histories presented in volume 10 of the *Metals Handbook* (ref. [\[1\]](#)).

W-4140 MITIGATING ACTIONS

Wear of material may be countered by altering the service conditions or selecting a more wear resistant material. Material selection may simply involve choosing the best base metal or it may involve one of many surface alteration methods applied to the base metal(s). In many applications, use of suitable lubricants can significantly lessen the effects of wear.

Altering service conditions include

- (a) design features to reduce the amount of relative vibration or motion between mating surfaces
- (b) improved surface finishes to minimize projections
- (c) means for excluding aggressive chemicals from the mating surfaces
- (d) redesigns to lower stresses between parts that will be moving with respect to one another

There are no simple guidelines on selection of materials for wear resistance. An excellent place to start is a chapter in Volume 1 of the *Metals Handbook* (ref. [\[2\]](#)) entitled, "The Selection of Steel for Wear Resistance." This chapter also addresses many ways in which heat treatments can be used to achieve a more wear resistant material with selected materials. The wear resistance of specific materials can also be found in newer versions of Volumes 1 and 2 of the *Metals Handbook* (refs. [\[3\]](#) and [\[4\]](#)).

Surfaces of normally wear-prone materials can be altered to significantly improve wear resistance. These surface enhancements include the more conventional carburizing and nitrating treatments. Surface can also be coated by a variety of wear resistant material applied by

(a) chemical vapor deposition (CVD) or physical vapor deposition (PVD) (ref. [5])

(b) hardcoat anodizing (ref. [6])

(c) plasma arc spraying (ref. [7])

(d) ion implantation (ref. [8])

(e) detonation-gun or electro-spark processes (ref. [9]), or

(f) conventional hard chrome plating

W-4150 REFERENCES

- [1] *Metals Handbook*, Volume 10, "Failure Analysis and Prevention," 8th Edition, 1975, American Society for Metals, pp. 134–160.
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W-4200 THERMAL FATIGUE

This section is under development.

W-4300 DYNAMIC LOADING — VIBRATION, WATER HAMMER AND UNSTABLE FLUID FLOW

This section is under development.

W-4400 CREEP

W-4410 GENERAL DESCRIPTION

As temperature increases, creep failures become more likely. Creep failure is primarily an intergranular fracture phenomenon (that occurs at relatively high temperatures) (ref. [1]), defined as the progressive deformations of a material at constant stress (ref. [1]). Creep failure (fracture) is known as stress rupture, and reflects the effect of temperature on long-term load-bearing characteristics (ref. [1]). Three stages of creep exist, and can be considered in design:

(a) primary creep due to initial, transient loading

(b) secondary, or steady state creep, where the creep rate is measured

(c) tertiary creep, leading to instability and failure (ref. [1]).

Stress relaxation is a related phenomenon in which the stress in a member decreases, when a constant amount of deformation is applied, due to creep (ref. [1]). In this case, stress relaxation in bolted joints and shrunk or press-fit components can result in leaks or loss of function (ref. [1]).

Creep is not generally a major consideration in light water reactors (LWRs), due to their operation at approximately 550°F (290°C), somewhat below the creep range for most ASME Code materials in ASME Section III. Temperature limits below which design for creep time dependency is not required are about 700°F (370°C) for ferritic steels and 800°F (425°C) for austenitic steels and alloys (ref. [2]). However, creep and creep-fatigue are very important in the design of components for use in nuclear power plants such as liquid metal fast breeder reactors (LMFBRs) and high temperature gas reactors (HTGRs), where temperatures are in the creep range as defined by ASME Section III, Subsection NH.

In-pile ductility and creep ductility may suffer after irradiation in boiling water reactors (BWRs) and pressurized water reactors (PWRs), for austenitic stainless steels and nickel alloys (ref. [3]). An influence of irradiation on creep and stress relaxation behavior has been shown for materials stressed at or above the yield strength, and exposed to significant irradiation [on the order of 1×10^{21} n/cm² (> 1 MeV)]. However, especially in BWRs, intergranular stress corrosion cracking (IGSCC) is usually of more concern for these components than creep (ref. [3]). Some typical reactor vessel components for which irradiation-enhanced creep may occur are austenitic stainless steel components such as cladding of control rods, absorber rods and neutron sources, Alloy X-750 components such as spring and bolts, Alloy 718 springs, and Alloy 625 cladding of control rod (ref. [3]).

W-4420 MATERIALS

Selection of materials for creep rupture resistance is usually based upon creep test results. However, this is not always a simple matter, since it is physically difficult

to perform creep-rupture tests at relatively low service temperature. At low temperatures, the stress must exceed the material yield strength in order to produce failure in a reasonable time (ref. [2]). Thus, time-temperature parameters are employed to extrapolate results to lower temperatures, and some uncertainty can exist (ref. [2]). ASME Code data bases and design factors account for such uncertainties (ref. [2]).

Environmental effects, such as carbon transport in the liquid sodium coolant of LMFBRs (ref. [5]) can be significant and should be considered. Significant carburization or decarburization can occur in austenitic/ferritic alloy piping systems, and influence creep rupture and fatigue properties (ref. [5]). The effects of helium on the creep properties of Type 304 and 316, and other heat exchanger alloys, have been shown to be relatively small for HTGRs (ref. [6]).

Significant stress relaxation has been demonstrated in tests for Type 304 stainless steel, Alloy X-750 and Alloy 718 during irradiation in-pile to fluences equal to or greater than 5×10^{21} n/cm² (> 1 MeV) (ref. [4]). As high as 60 to 100% relaxation of stresses occurred in the preceding cases, for irradiation at 140 to 600°F (60 to 315°C) (ref. [4]). Stress relaxation also occurred in Type 304 bolts of from 20 ksi (140 MPa) cold preloaded stress to 10 ksi after reactor heatup to 550°F and irradiation at 6×10^{19} n/cm² (> 1 MeV) (ref. [4]).

Creep-fatigue testing of 2 $\frac{1}{4}$ Cr-1Mo steel and of austenitic stainless steels for LMFBRs has been reported to show increased damage when creep and fatigue loadings are interspersed (ref. [7]). In this case, the creep resistance is decreased by imposing transient plastic strains (ref. [7]). In general, more stable microstructures are thought to result in better long-term resistance to creep rupture (ref. [8]). Stable microstructures, such as those produced to complete annealing or normalizing and tempering, can show lower rupture strength at short times, but are more resistant to microstructure transformations which can degrade properties at long times (ref. [8]). However, it must be mentioned that there is a sparsity of creep-rupture data at long times and properties in this regime are determined by extrapolation methods in most cases (ref. [2]).

W-4430 DESIGN

Limitations on operation and design are specified by the ASME Section III, Subsection NH, for elevated temperature applications where creep is a significant factor. This subsection recognizes three creep properties for elevated temperature design: rupture stress, stress for 1% strain, and stress for beginning of tertiary creep (ref. [2]).

If significant creep takes place locally (due to a local hot spot, etc.), large residual tensile stresses are frozen into the metal after cooldown (ref. [9]). The stresses can be greater than ASME Code allowable stresses and are most important in fatigue, where damage is caused by the repeated stress reversals which accompany startup and shutdown operation (ref. [9]). Significant stress relaxation

can cause equipment functional problems and possible shortened fatigue life in bolts due to decreased preload (ref. [4]).

W-4440 MITIGATING ACTIONS

Nuclear equipment, which operates at elevated temperatures in the creep range, must be designed in accordance with Section III, Subsection NH for Class 1 components. Creep rupture and creep-fatigue evaluations are required. In other cases, not covered by Subsection NH, prediction of long-term properties may be pursued by time-temperature extrapolation of test data (ref. [1]). For example, STRAP is a computer program for steam turbine rotors which predicts rotor lifetime considering the duty cycles and ultrasonic test (UT) results (ref. [10]). This program contains fracture toughness, stress rupture, yield strength, and fatigue crack growth rate data for air melted 1Cr-Mo-V forgings (ref. [10]).

The accurate, or conservative, prediction of creep rupture and creep-fatigue damage, and design in accordance with these concepts, is the most effective mitigating action or remedy. However, other parameters can also affect creep and stress relaxation and should be controlled where possible. Irradiation of susceptible parts should be minimized and cold work (which can undergo stress relaxation) minimized, where practical. Bolting applications in high radiation fields should be designed for possible reduced preload, in order to account for stress relaxation.

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NONMANDATORY APPENDIX Y

ARTICLE Y-1000 EVALUATION OF THE DESIGN OF RECTANGULAR AND HOLLOW CIRCULAR CROSS SECTION WELDED ATTACHMENTS ON CLASS 1, 2, AND 3 PIPING

Y-1100 INTRODUCTION

Y-1110 SCOPE

(a) The Articles of this Appendix provide rules and service limits which may be used to evaluate the design of both rectangular cross section and hollow circular cross section welded attachments on pipe.

(b) The rules presented here are not intended to exclude other methods such as finite element analyses.

ARTICLE Y-2000

PROCEDURE FOR EVALUATION OF THE DESIGN OF RECTANGULAR CROSS SECTION ATTACHMENTS ON CLASS 1 PIPING

Y-2100 INTRODUCTION

Y-2110 SCOPE

This Article provides rules and service limits which may be used to evaluate the design of rectangular cross section welded attachments on Class 1 pipe under Section III, Division 1.

Y-2200 LIMITATIONS TO APPLICABILITY

(a) The attachment shall be welded to the pipe by a full penetration weld (see Figure Y-2300-1).

(b) The attachment material and pipe material shall have essentially the same moduli of elasticity and coefficients of thermal expansion.

(c) $\beta_1 \leq 0.5$, $\beta_2 \leq 0.5$, and the product $\beta_1 \times \beta_2 \leq 0.075$, where β_1 and β_2 are defined in Y-2300.

(d) The attachment is made on straight pipe, with the nearest edge of the attachment weld located at a minimum distance of \sqrt{RT} from any other weld or other discontinuity; R and T are defined in Y-2300. For multiple attachments located at a distance less than \sqrt{RT} to each other, the stress effects for each individual attachment shall be superimposed.

(e) $D_o/T \leq 100$.

Y-2300 NOMENCLATURE AND DEFINITIONS (SEE FIGURE Y-2300-1)

The nomenclature defined below is used in the equations and figures of this Article.

D_o = outside diameter of pipe, in. (mm)

R = mean pipe radius, in. (mm)

T = nominal pipe-wall thickness, in. (mm)

$\beta_1 = L_1/R$, L_1 is defined in Y-2300-1

$\beta_2 = L_2/R$, L_2 is defined in Y-2300-1

$\gamma = R/T$

$A_l = 4L_1L_2$, in.² (mm²)

$B_L = (\frac{2}{3})C_L$, but not less than 1.0

$B_N = (\frac{2}{3})C_N$, but not less than 1.0

$B_T = (\frac{2}{3})C_T$, but not less than 1.0

$C_L = 0.26 (\gamma)^{1.74} \beta_1 \beta_2^2 \eta^{4.74} \geq 1.0$

$C_N = 0.38 (\gamma)^{1.90} \beta_1^2 \beta_2 \eta^{3.40} \geq 1.0$

$$C_T = 3.82 (\gamma)^{1.64} \beta_1 \beta_2 \eta^{1.54} \geq 1.0$$

$E\alpha$ = modulus of elasticity, E , times the mean coefficient of thermal expansion, α , both at room temperature, psi/°F (kPa/°C)

K_l = 2.0 for as-welded full penetration welds and 1.3 for ground fillet welds as per Figure Y-2300-1

L_a = lesser of L_2 and T , in. (mm)

L_b = lesser of L_1 and T , in. (mm)

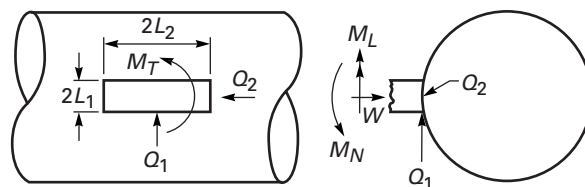
L_c = lesser of L_1 and L_2 , in. (mm)

L_d = greater of L_1 and L_2 , in. (mm)

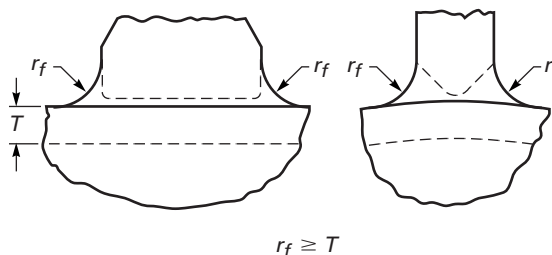
M_L = bending moment applied to the attachment as shown in Figure Y-2300-1, in.-lb (kN·m)

M_N = bending moment applied to the attachment as shown in Figure Y-2300-1, in.-lb (kN·m)

Figure Y-2300-1
Nomenclature Illustration



Sketch (a): Graphic representation of L_1 , L_2 , W , M_L , M_N , Q_1 , Q_2 , and M_T ; L_1 and L_2 are to be measured along the surface of the run pipe. Welds or fillet radii between attachment and pipe are not to be included.



Sketch (b): Ground weld or integrally cast attachment; $K_1 = 1.3$. For as-welded or fillet radii $r_f \leq T$, use $K_1 = 2.0$.

- M_T = torsional moment applied to the attachment as shown in Figure Y-2300-1, in.-lb (kN·m)
 \bar{M}_T = greater of $M_T / \{L_c L_d T [1 + (L_c/L_d)]\}$ and $M_T / \{[0.8 + 0.05(L_d/L_c)] L_c^2 L_d\}$, psi (kPa)
 Q_1 = shear load applied to the attachment as shown in Figure Y-2300-1, lb (kN)
 Q_2 = shear load applied to the attachment as shown in Figure Y-2300-1, lb (kN)
 S_m = allowable design stress intensity, psi (kPa) (lesser of attachment or pipe material)
 S_y = yield strength at temperature, psi (kPa) (lesser of attachment or pipe material)
 T_t = average temperature of that portion of the attachment within a distance of $2T$ from the surface of the pipe, °F (°C)
 T_w = average temperature of the portion of the pipe under the attachment and within a distance of \sqrt{RT} from the edge of the attachment, °F (°C)
 W = thrust load applied to the attachment as shown in Figure Y-2300-1, lb (kN)
 $Z_{IL} = (\frac{4}{3})L_1(L_2)^2$, in.³ (mm³)
 $Z_{IN} = (\frac{4}{3})(L_1)^2L_2$, in.³ (mm³)
 $\eta = -(X_1 \cos \theta + Y_1 \sin \theta) - (1/A_0)(X_1 \sin \theta - Y_1 \cos \theta)^2$

Load	A_0	θ	X_0	Y_0
Thrust	2.2	40°	0	0.05
Longitudinal moment	2.0	50°	-0.45	-0.55
Circumferential moment	1.8	40°	-0.75	-0.60

$$X_1 = X_0 + \log_{10} \beta_1$$

$$Y_1 = Y_0 + \log_{10} \beta_2$$

M_L , M_N , M_T , Q_1 , Q_2 , and W are determined at the surface of the pipe. The values of attachment loads used in the stress evaluation (Y-2400) are based on the loads used in the different Code equations.

M_L^{**} , M_N^{**} , M_T^{**} , Q_1^{**} , Q_2^{**} , and W^{**} are absolute values of maximum loads occurring simultaneously.

Y-2400 EVALUATION PROCEDURE

The loads on the attachment cause stresses in the pipe wall. Equations are provided in Y-2410(a) to determine these stresses. The attachment stresses are then added to the piping system stresses at the attachment. The piping system stresses are determined by eqs. (9), (10), (11), (12), (13), and (14) of NB-3650 for straight pipe. The Code equations including the attachment stress terms are given in Y-2410(b). The attachment stresses, S_{mb} , S_{nb} , and S_{pl} are to be calculated for the loading conditions corresponding to the NB-3650 equations. For example, in calculating S_{ml} for use in NB-3652 eq. (9), W , M_L , M_N , Q_1 , Q_2 , and M_T are the loads on the attachment due to mechanical loads.

There are additional equations given in Y-2410(c) that also must be checked for attachment stresses. These are based on the absolute values of maximum loads occurring simultaneously under all service loading conditions.

Y-2410 ANALYSIS OF ATTACHMENT

(a) Calculate the stresses S_{mb} , S_{nb} , S_{pl} , and S_{nl}^{**}

$$S_{ml} = B_T W / A_l + B_L M_L / Z_{IL} + B_N M_N / Z_{IN} + Q_1 / 2L_1 L_a + Q_2 / 2L_2 L_b + \bar{M}_T \quad (1)$$

$$S_{nl} = C_T W / A_l + C_L M_L / Z_{IL} + C_N M_N / Z_{IN} + Q_1 / 2L_1 L_a + Q_2 / 2L_2 L_b + \bar{M}_T \quad (2)$$

$$S_{pl} = K_1(S_{nl}) + K_1 E \alpha |T_t - T_w| \quad (3)$$

NOTE: For thermal transients with fluid temperature changes greater than 100°F (37.8°C) and rate of change greater than 10°F/min., $|T_t - T_w|$ may be conservatively taken as one-half of the difference between the initial metal temperature and the transient fluid temperature during a temperature transient.

$$S_{nl}^{**} = C_T W^{**} / A_l + C_L M_L^{**} / Z_{IL} + C_N M_N^{**} / Z_{IN} + Q_1^{**} / 2L_1 L_a + Q_2^{**} / 2L_2 L_b + \bar{M}_T^{**} \quad (4)$$

(b) The following modified Code equations shall be satisfied. All terms except attachment stresses, or where otherwise noted, are defined in NB-3652 and NB-3653.

(1) NB-3652 eq. (9) becomes

$$B_1 \frac{PD_0}{2t} + B_2 \frac{D_0}{2l} M_i + S_{ml} \quad (\text{NB-9})$$

- $\leq 1.5S_m$ for Design and Service Level A loadings
- $\leq 1.8S_m$ but not greater than $1.5S_y$ for Level B loadings
- $\leq 2.25S_m$ but not greater than $1.8S_y$ for Level C loadings
- $\leq 3.0S_m$ but not greater than $2.0S_y$ for Level D loadings

where

$$B_1 = 0.5$$

$$B_2 = 1.0 \text{ for straight pipe.}$$

(2) NB-3653.1 eq. (10) becomes

$$S_n = C_1 \frac{P_0 D_0}{2t} + C_2 \frac{D_0}{2l} M_i + S_{nl} \leq 3S_m \quad (\text{NB-10})$$

where

$$C_1 = C_2 = 1.0 \text{ for straight pipe.}$$

If S_n , as calculated by eq. (NB-10), exceeds $3S_m$, NB-3653.6 eqs. (12) and (13) and the thermal stress ratchet check of NB-3653.7 must be satisfied; S_{nl} need not be included in these checks. However, the value of K_e shall be determined from S_n , including S_{nl} .

(3) NB-3653.2 eq. (11) becomes

$$S_p = K_1 C_1 \frac{P_0 D_0}{2t} + K_2 C_2 \frac{D_0}{2l} M_i + \frac{1}{2(1 - \nu)} \left[K_3 E \alpha \left| \Delta T_1 \right| + \frac{1}{1 - \nu} E \alpha \left| \Delta T_2 \right| + S_{pl} \right] \quad (\text{NB-11})$$

where $K_1 = K_2 = K_3 = 1.0$ for straight pipe.

(4) NB-3653.6 eq. (14) becomes:

$$S_{\text{alt}} = K_e \frac{S_p}{2} \quad (\text{NB-14})$$

where S_n and S_p are as calculated by eqs. (2)(NB-10) and (3)(NB-11) of this Article.

(c) In addition to the Code equations, the following equations shall also be satisfied:

$$S_{nl}^{**} \leq 2S_y \quad (5)$$

$$Q_1^{**} / 2L_1 L_a + Q_2^{**} / 2L_2 L_b + \bar{M}_T^{**} \leq S_y \quad (6)$$

Y-2500 ANALYSIS DOCUMENTATION

Analyses demonstrating compliance with this Article shall be included in the Design Report for the piping system.

ARTICLE Y-3000

PROCEDURE FOR EVALUATION OF THE DESIGN OF RECTANGULAR CROSS SECTION ATTACHMENTS ON CLASS 2 OR 3 PIPING

Y-3100 INTRODUCTION

Y-3110 SCOPE

This Article provides rules and service limits which may be used to evaluate the design of rectangular cross section welded attachments on Class 2 or 3 pipe under Section III, Division 1.

Y-3200 LIMITATIONS TO APPLICABILITY

- (a) The attachment shall be welded to the pipe by
- (1) a full penetration weld
 - (2) a fillet or partial penetration weld along at least three sides of the attachment
 - (3) a fillet or partial penetration weld along the two long sides of the attachment, where the length of the long side is at least three times the length of the short side
- (b) The attachment material and pipe material shall have essentially the same moduli of elasticity and coefficients of thermal expansion.
- (c) $\beta_1 \leq 0.5$, $\beta_2 \leq 0.5$, and the product $\beta_1 \times \beta_2 \leq 0.075$, where β_1 and β_2 are defined in Y-3300.
- (d) The attachment is made on straight pipe, with the nearest edge of the attachment weld located at a minimum distance of \sqrt{RT} from any other weld or other discontinuity; R and T are defined in Y-3300. For multiple attachments located at a distance less than \sqrt{RT} to each other, the stress effects for each individual attachment shall be superimposed.
- (e) $D_o/T \leq 100$.

Y-3300 NOMENCLATURE AND DEFINITIONS (SEE FIGURE Y-3300-1)

The following nomenclature is used in the equations and figures of this Article.

- $A_l = 4L_1L_2$, in.² (mm²)
- A_w = total fillet weld throat area, in.² (mm²)
- $B_L = (\frac{2}{3})C_L$, but not less than 1.0
- $B_N = (\frac{2}{3})C_N$, but not less than 1.0
- $B_T = (\frac{2}{3})C_T$, but not less than 1.0
- $C_L = 0.26 (\gamma)^{1.74} \beta_1 \beta_2^2 \eta^{4.74} \geq 1.0$
- $C_N = 0.38 (\gamma)^{1.90} \beta_1^2 \beta_2 \eta^{3.40} \geq 1.0$

$$C_T = 3.82 (\gamma)^{1.64} \beta_1 \beta_2 \eta^{1.54} \geq 1.0$$

D_o = outside diameter of pipe, in. (mm)

K_l = 2.0 for as-welded full penetration welds and fillet or partial penetration welds welded on four sides
= 3.6 for fillet or partial penetration welds where the attachment is welded on two or three sides

L_a = lesser of L_2 and T , in. (mm)

L_b = lesser of L_1 and T , in. (mm)

L_c = lesser of L_1 and L_2 , in. (mm)

L_d = greater of L_1 and L_2 , in. (mm)

M_L = bending moment applied to the attachment as shown in Figure Y-3300-1, in.-lb (kN·m)

M_N = bending moment applied to the attachment as shown in Figure Y-3300-1, in.-lb (kN·m)

M_T = torsional moment applied to the attachment as shown in Figure Y-3300-1, in.-lb (kN·m)

\bar{M}_T = greater of $M_T / \{L_c L_d T [1 + (L_c/L_d)]\}$ and $M_T / \{[0.8 + 0.05(L_d/L_c)]L_c^2 L_d\}$, psi (kPa)

Q_1 = shear load applied to the attachment as shown in Figure Y-3300-1, lb (kN)

Q_2 = shear load applied to the attachment as shown in Figure Y-3300-1, lb (kN)

R = mean pipe radius, in. (mm)

S_y = yield strength at temperature, psi (kPa) (lesser of attachment material or pipe material)

T = nominal pipe-wall thickness, in. (mm)

W = thrust load applied to the attachment as shown in Figure Y-3300-1, lb (kN)

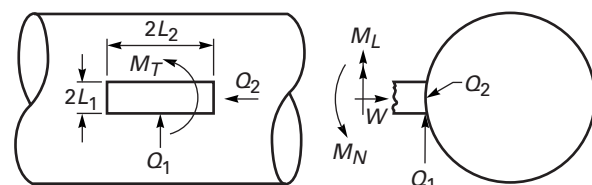
$$X_1 = X_0 + \log_{10} \beta_1$$

$$Y_1 = Y_0 + \log_{10} \beta_2$$

$$Z_{lL} = (\frac{1}{3})L_1(L_2)^2, \text{ in.}^3 \text{ (mm}^3\text{)}$$

$$Z_{lN} = (\frac{1}{3})(L_1)^2L_2, \text{ in.}^3 \text{ (mm}^3\text{)}$$

Figure Y-3300-1
Nomenclature Illustration



Z_{wd} = section modulus of fillet or partial penetration weld about the neutral axis of bending parallel to L_1 , in.³ (mm³)

Z_{wl} = section modulus of fillet or partial penetration weld about the neutral axis of bending parallel to L_2 , in.³ (mm³)

Z_{wt} = torsional section modulus of fillet or partial penetration weld for torsional loading, in.³ (mm³)

$\beta_1 = L_1/R$, L_1 is defined in Y-3300-1

$\beta_2 = L_2/R$, L_2 is defined in Y-3300-1

$\gamma = R/T$

$\eta = -(X_1 \cos \theta + Y_1 \sin \theta) - (1/A_0)(X_1 \sin \theta - Y_1 \cos \theta)^2$

Load	A_0	θ	X_0	Y_0
Thrust	2.2	40°	0	0.05
Longitudinal moment	2.0	50°	-0.45	-0.55
Circumferential moment	1.8	40°	-0.75	-0.60

M_L , M_N , M_T , Q_1 , Q_2 , and W are determined at the surface of the pipe. The values of attachment loads used in the stress evaluation (Y-3400) are based on the loads used in the different Code equations.

M_L^{**} , M_N^{**} , M_T^{**} , Q_1^{**} , Q_2^{**} , and W^{**} are absolute values of maximum loads occurring simultaneously.

Y-3400 EVALUATION PROCEDURE

The loads on the attachment cause stresses in the pipe wall. Equations are provided in Y-3410(a) to determine these stresses. The attachment stresses are then added to the piping system stresses at the attachment. The piping system stresses are determined by NC-3652 eq. (8), NC-3653.1 eq. (9), and NC-3653.2 eqs. (10), (10a), and (11) for straight pipe. The Code equations including the attachment stress terms are given in X-3410(b). The attachment stresses S_{mb} , S_{nb} , and S_{pl} are to be calculated for the loading conditions corresponding to NC-3652 eq. (8), NC-3653.1 eq. (9), and NC-3653.2 eqs. (10), (10a), and (11). For example, in calculating S_{ml} for use in NC-3652 eq. (8), W , M_L , M_N , Q_1 , Q_2 , and M_T are the loads on the attachment due to weight and other sustained loads. While NC is used below, the same rules apply for ND piping.

There are additional equations given in Y-3410(c) for all weld configurations and Y-3420(b) for attachments welded with fillet or partial penetration welds that also must be checked for attachment stresses. These are based on the absolute values for maximum loads occurring simultaneously for Level A, B, C, or D service loading conditions.

Y-3410 ANALYSIS OF ATTACHMENT WELDED TO PIPE WITH A FULL PENETRATION WELD

(a) Calculate the stresses S_{mb} , S_{nb} , S_{pl} , and S_{nl}^{**} :

$$S_{ml} = B_T W / A_l + B_L M_L / Z_{lL} + B_N M_N / Z_{lN} + Q_1 / 2L_1 L_a + Q_2 / 2L_2 L_b + \bar{M}_T \quad (1)$$

$$S_{nl} = C_T W / A_l + C_L M_L / Z_{lL} + C_N M_N / Z_{lN} + Q_1 / 2L_1 L_a + Q_2 / 2L_2 L_b + \bar{M}_T \quad (2)$$

$$S_{pl} = K_1(S_{nl}) \quad (3)$$

$$S_{nl}^{**} = C_T W^{**} / A_l + C_L M_L^{**} / Z_{lL} + C_N M_N^{**} / Z_{lN} + Q_1^{**} / 2L_1 L_a + Q_2^{**} / 2L_2 L_b + \bar{M}_T^{**} \quad (4)$$

(b) The following modified Code equations shall be satisfied. All terms except attachment stresses, or where otherwise noted, are defined in NC-3652.

(1) NC-3652 eq. (8) becomes

$$S_{SL} = B_1 \frac{PD_0}{2t_n} + B_2 \frac{M_A}{Z} + S_{ml} \leq 1.5S_h \quad (\text{NC-8})$$

where $B_1 = 0.5$ and $B_2 = 1.0$ for straight pipe.

(2) NC-3653.1 eq. (9) becomes

$$S_{OL} = B_1 \frac{P_{\max} D_0}{2t_n} + B_2 \frac{M_A + M_B}{Z} + S_{ml} \quad (\text{NC-9})$$

$\leq 1.8S_h$ but not greater than $1.5S_y$ for Level A and B loadings

$\leq 2.25S_h$ but not greater than $1.8S_y$ for Level C loadings

$\leq 3.0S_h$ but not greater than $2.0S_y$ for Level D loadings

(3) NC-3653.2 eq. (10) becomes

$$S_E = \frac{iM_C}{Z} + \frac{S_{pl}}{2} \leq S_A \quad (\text{NC-10})$$

(4) NC-3653.2 eq. (10a) becomes

$$\frac{iM_D}{Z} + \frac{S_{pl}}{2} \leq 3.0S_c \quad (\text{NC-10a})$$

(5) NC-3653.2 eq. (11) becomes

$$S_{TE} = \frac{PD_0}{4t_n} + 0.75i\left(\frac{M_A}{Z}\right) + i\left(\frac{M_C}{Z}\right) + S_{ml} + \frac{S_{pl}}{2} \leq (S_h + S_A) \quad (\text{NC-11})$$

In eq. (5)(NC-11), S_{ml} is the same as used in eq. (1)(NC-8), and S_{pl} is the same as used in eq. (3)(NC-10).

(c) In addition to the modified Code equations, the following equations shall also be satisfied.

$$S_{nl}^{**} \leq 2S_y \quad (5)$$

$$Q_1^{**}/2L_1L_a + Q_2^{**}/2L_2L_b + \bar{M}_T^{**} \leq S_y \quad (6)$$

Y-3420 ANALYSIS OF ATTACHMENT WELDED TO PIPE WITH FILLET OR PARTIAL PENETRATION WELDS

(a) The requirements of Y-3410 shall be met. For attachments welded on two or three sides, the value of K_1 used in calculating S_{pl} shall be 3.6.

(b) The following additional equations shall be satisfied.

$$W^{**}/A_W + M_L^{**}/Z_{wd} + M_N^{**}/Z_{wl} + 2(Q_1^{**} + Q_2^{**})/A_W + M_T^{**}/Z_{wt} \leq 2S_y \quad (7)$$

$$\{(W^{**}/A_W)^2 + 4[(Q_1^{**} + Q_2^{**})/A_W + M_T^{**}/Z_{wt}]^2\}^{1/2} \leq S_y \quad (8)$$

Y-3430 DIFFERENTIAL METAL TEMPERATURE EFFECTS

The potential for increased stress at the attachment welds, which may occur as a result of differential metal temperatures between the attachment and the run, should be considered in the design evaluation.

Y-3500 ANALYSIS DOCUMENTATION

Analyses demonstrating compliance with this Article shall be included in the Design Report for the piping system.

ARTICLE Y-4000

PROCEDURE FOR EVALUATION OF THE DESIGN OF HOLLOW CIRCULAR CROSS SECTION WELDED ATTACHMENTS ON CLASS 1 PIPING

Y-4100 INTRODUCTION

Y-4110 SCOPE

This Article provides rules and service limits which may be used to evaluate the design of hollow circular cross section welded attachments on Class 1 pipe under Section III, Division 1.

Y-4200 LIMITATIONS TO APPLICABILITY

(a) The attachment shall be welded to the pipe by a full penetration weld (see [Figure Y-4200-1](#)).

(b) The attachment material and pipe material shall have essentially the same moduli of elasticity and coefficients of thermal expansion.

(c) The constants defined in [Y-4300](#) fall within the following ranges:

(1) $4.0 \leq \gamma \leq 50.0$

(2) $0.2 \leq \tau \leq 1.0$

(3) $0.3 \leq \beta \leq 1.0$

(4) the axis of the attachment is perpendicular to the axis of the run pipe

(d) The attachment is made on straight pipe, with the nearest edge of the attachment weld located at a minimum distance of \sqrt{RT} from any other weld or other discontinuity (see [Y-4300](#) for definitions of R and T). For multiple

attachments located at a distance less than \sqrt{RT} to each other, the stress effects for each individual attachment shall be superimposed.

Y-4300 NOMENCLATURE AND DEFINITIONS (SEE [FIGURE Y-4300-1](#))

The nomenclature defined below is used in the equations and figures of this Article.

$$A_T = \pi (r_o^2 - r_i^2)$$

$$B_L = 0.5 (C_L), \text{ but not less than } 1.0$$

$$B_N = 0.5 (C_N), \text{ but not less than } 1.0$$

$$B_T = 0.5 (C_T), \text{ but not less than } 1.0$$

$$B_W = 0.5 (C_W), \text{ but not less than } 1.0$$

$$C = A_o (2\gamma)^{n_1} \beta^{n_2} \tau^{n_3} \text{ but not less than } 1.0$$

$$C_T = 1.0 \text{ for } \beta \leq 0.55$$

$$= C_N \text{ for } \beta = 1.0, \text{ but not less than } 1.0; C_T \text{ should be linearly interpolated for } 0.55 < \beta < 1.0, \text{ but not less than } 1.0$$

$$d_o = \text{outside diameter of attachment, in. (mm)}$$

$$D_o = \text{outside diameter of run pipe, in. (mm)}$$

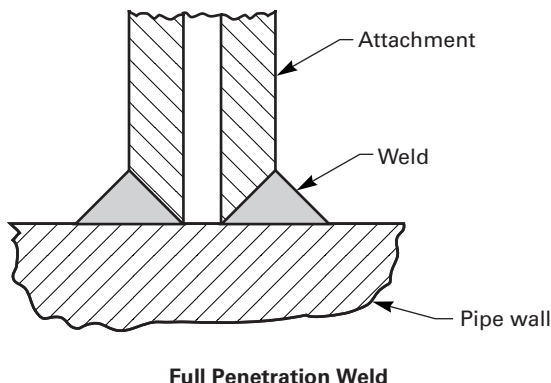
$$E\alpha = \text{modulus of elasticity, } E, \text{ times the mean coefficient of thermal expansion, } \alpha, \text{ both at room temperature, psi/}^\circ\text{F (kPa/}^\circ\text{C)}$$

$$I_T = \frac{\pi}{4} (r_o^4 - r_i^4)$$

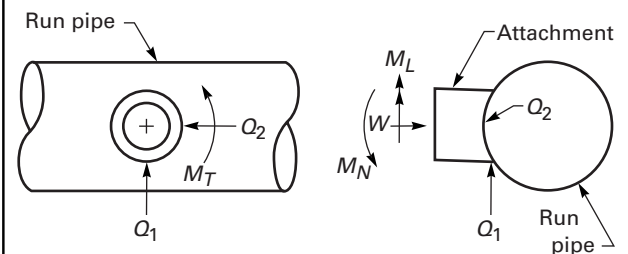
$$J = \text{lesser of } \pi r_o^2 T \text{ or } Z_T$$

$$K_T = 1.8 \text{ for full penetration welds}$$

**Figure Y-4200-1
Weld Type Illustration**



**Figure Y-4300-1
Nomenclature Illustration**



M_L	= bending moment applied to the attachment as shown in Figure Y-4300-1, in.-lb (kN·m)
M_N	= bending moment applied to the attachment as shown in Figure Y-4300-1, in.-lb (kN·m)
M_T	= torsional moment applied to the attachment as shown in Figure Y-4300-1, in.-lb (kN·m)
Q_1	= shear load applied to the attachment as shown in Figure Y-4300-1, lb (kN)
Q_2	= shear load applied to the attachment as shown in Figure Y-4300-1, lb (kN)
R	= mean run pipe radius, in. (mm)
r_i	= attachment inside radius, in. (mm)
r_o	= attachment outside radius, in. (mm)
R_o	= run pipe outside radius, in. (mm)
S_m	= allowable design stress intensity, psi (kPa) (lesser of attachment or pipe material)
S_Y	= yield stress at temperature, psi (kPa) (lesser of attachment or pipe material)
T	= run pipe wall thickness, in. (mm)
t	= attachment wall thickness, in. (mm)
T_T	= average temperature of that portion of the attachment within a distance of $2t$ from the surface of the pipe, °F (°C)
T_W	= average temperature of the portion of the pipe under the attachment and within a distance of \sqrt{RT} from the edge of the attachment, °F (°C)
W	= thrust load applied to the attachment as shown in Figure Y-4300-1, lb (kN)
Z_T	= I_T/r_o
β	= d_o/D_o
γ	= R_o/T
τ	= t/T

The equation for C shall be used to determine C_W , C_L , and C_N , based on the following table. Select the maximum value of the pipe and the attachment equations.

Index	Part	β range	A_o	n_1	n_2	n_3
C_W	Pipe attachment	0.3–1.0	1.40	0.81	(1)	1.33
		0.3–1.0	4.00	0.55	(2)	1.00
C_L	Pipe attachment	0.3–1.0	0.46	0.60	–0.04	0.86
		0.3–1.0	1.10	0.23	–0.38	0.38
C_N	Pipe attachment	0.3–0.55	0.51	1.01	0.79	0.89
		0.3–0.55	0.84	0.85	0.80	0.54
	Pipe attachment	> 0.55–1.0	0.23	1.01	–0.62	0.89
		> 0.55–1.0	0.44	0.85	–0.28	0.54

NOTES:
 (1) Replace β^{n_2} with $e^{(-1.2\beta^3)}$
 (2) Replace β^{n_2} with $e^{(-1.35\beta^3)}$

M_L , M_N , M_T , Q_1 , Q_2 , and W are determined at the surface of the pipe.

M_L^{**} , M_N^{**} , M_T^{**} , Q_1^{**} , Q_2^{**} , and W^{**} are absolute values of maximum loads occurring simultaneously under all service loading conditions.

Y-4400 EVALUATION PROCEDURE

The loads on the attachment cause stresses in the pipe wall. Equations are provided in X-4410(a) to determine these stresses. The attachment stresses are then added to the piping system stresses at the attachment. The piping system stresses are determined by NB-3652 eq. (9), NB-3653.1 eq. (10), NB-3653.2 eq. (11), and NB-3653.6 eqs. (12), (13), and (14). The Code equations including the attachment stress terms are given in Y-4410(b). The attachment stresses, S_{MT} , S_{NT} , and S_{PT} are to be calculated for the loading conditions corresponding to NB-3652 eq. (9), NB-3653.1 eq. (10), NB-3653.2 eq. (11), and NB-3653.6 eqs. (12), (13), and (14). For example, in calculating S_{MT} for use in NB-3652 eq. (9) for design conditions, W , M_L , M_N , Q_1 , Q_2 , and M_T are the loads on the attachment due to design mechanical loads.

There are additional equations given in Y-4410(c) that also must be checked for attachment stresses. These are based on the absolute values for maximum loads occurring simultaneously under all service loading conditions.

Y-4410 ANALYSIS OF ATTACHMENTS

(a) Calculate the stresses S_{MT} , S_{NT} , S_{PT} , and S_{NT}^{**} :

$$S_{MT} = \frac{B_W W}{A_T} + \frac{B_N M_N}{Z_T} + \frac{B_L M_L}{Z_T} + \frac{2Q_1}{A_T} + \frac{2Q_2}{A_T} + \frac{B_T M_T}{J} \quad (1)$$

$$S_{NT} = \frac{C_W W}{A_T} + \frac{C_N M_N}{Z_T} + \frac{C_L M_L}{Z_T} + \frac{2Q_1}{A_T} + \frac{2Q_2}{A_T} + \frac{C_T M_T}{J} + 1.7E\alpha \left| T_T - T_W \right| \quad (2)$$

NOTE: For thermal transients with fluid temperature changes greater than 100°F (37.8°C) and rate of change greater than 10°F/min., $|T_T - T_W|$ may be conservatively taken as one-half of the difference between the initial metal temperature and the transient fluid temperature during a temperature transient.

$$S_{PT} = K_T(S_{NT}) \quad (3)$$

$$S_{NT}^{**} = \frac{C_W W^{**}}{A_T} + \frac{C_N M_N^{**}}{Z_T} + \frac{C_L M_L^{**}}{Z_T} + \frac{2Q_1^{**}}{A_T} + \frac{2Q_2^{**}}{A_T} + \frac{C_T M_T^{**}}{J} \quad (4)$$

(b) The following modified Code equations shall be satisfied.

(1) NB-3652 eq. (9) becomes:

$$B_1 \frac{PD_0}{2t} + B_2 \frac{D_o}{2I} M_i + S_{MT} \quad (\text{NB-9})$$

$\leq 1.5S_m$ for Design and Service Level A loadings
 $\leq 1.8S_m$ but not greater than $1.5S_y$ for Level B loadings
 $\leq 2.25S_m$ but not greater than $1.8S_y$ for Level C loadings
 $\leq 3.0S_m$ but not greater than $2.0S_y$ for Level D loadings

where $B_1 = 0.5$ and $B_2 = 1.0$ for straight pipe.

(2) NB-3653.1 eq. (10) becomes:

$$S_n = C_1 \frac{P_o D_o}{2t} + C_2 \frac{D_o}{2l} M_i + S_{NT} \leq 3S_m \quad (\text{NB-10})$$

where

$C_1 = C_2 = 1.0$ for straight pipe.

If S_n , as calculated by eq. (NB-10), exceeds $3S_m$, NB-3653.6 eqs. (12) and (13) and the thermal stress ratchet check of NB-3653.7 must be satisfied; S_{nl} need not be included in these checks. However, the value of K_e shall be determined from S_n , including S_{nl} .

(3) NB-3653.2 eq. (11) becomes:

$$S_p = K_1 C_1 \frac{P_o D_o}{2t} + K_2 C_2 \frac{D_o}{2l} M_i + \frac{1}{2(1 - \nu)} K_3 E \alpha \left[\Delta T_1 + \frac{1}{1 - \nu} E \alpha \Delta T_2 \right] + S_{pT} \quad (\text{NB-11})$$

where

$K_1 = K_2 = K_3 = 1.0$ for straight pipe.

(4) NB-3653.6 eq. (14) becomes:

$$S_{ALT} = K_e \frac{S_p}{2} \quad (\text{NB-14})$$

where S_n and S_p are as calculated by eqs. (2)(NB-10) and (3)(NB-11) of this Article.

All terms except attachment stresses, or where otherwise noted, are defined in NB-3652 and NB-3653.

(c) In addition to the modified Code equations, the following equations shall also be satisfied:

$$S_{NT}^{**} \leq 2S_y \quad (5)$$

$$\frac{2Q_1^{**}}{A_T} + \frac{2Q_2^{**}}{A_T} + \frac{M_T^{**}}{J} \leq S_y \quad (6)$$

Y-4500 ANALYSIS DOCUMENTATION

Analyses demonstrating compliance with this Article shall be included in the Design Report for the piping system.

- S_c = basic material allowable stress at ambient temperature, psi (kPa) (lesser of attachment or pipe material allowable)
 S_h = basic material allowable stress at maximum (hot) temperature, psi (kPa) (lesser of attachment or pipe material allowable)
 S_y = yield stress at temperature, psi (kPa) (lesser of attachment or pipe material yield stress)
 T = nominal run pipe wall thickness, in. (mm)
 t = nominal attachment wall thickness, in. (mm)
 W = thrust load applied to the attachment as shown in Figure Y-5300-1, lb (kN)
 $Z_T = I_T/r_o$
 Z_{wl} = section modulus of fillet weld or partial penetration weld about the neutral axis normal to the run pipe center line, in.³ (mm³)
 Z_{wn} = section modulus of fillet weld or partial penetration weld about the neutral axis of bending parallel to run pipe centerline, in.³ (mm³)
 Z_{wt} = torsional section modulus of fillet weld or partial penetration weld for torsional loading, in.³ (mm³)
 $\beta = d_o/D_o$
 $\gamma = R_o/T$
 $\tau = t/T$

The equation for C shall be used to determine C_W , C_L , and C_N , based on the following table. Select the maximum value of the pipe and the attachment equations.

Index	Part	β range	A_o	n_1	n_2	n_3
C_W	Pipe	0.3–1.0	1.40	0.81	[Note (1)]	1.33
	attachment	0.3–1.0	4.00	0.55	[Note (2)]	1.00
C_L	Pipe	0.3–1.0	0.46	0.60	–0.04	0.86
	attachment	0.3–1.0	1.10	0.23	–0.38	0.38
C_N	Pipe	0.3–0.55	0.51	1.01	0.79	0.89
	attachment	0.3–0.55	0.84	0.85	0.80	0.54
	Pipe	> 0.55–1.0	0.23	1.01	–0.62	0.89
	attachment	> 0.55–1.0	0.44	0.85	–0.28	0.54

NOTES:
 (1) Replace β^{n_2} with $e^{(-1.2\beta^{n_3})}$
 (2) Replace β^{n_2} with $e^{(-1.35\beta^{n_3})}$

M_L , M_N , M_T , Q_1 , Q_2 , and W are determined at the surface of the pipe. The values of attachment loads used in the stress evaluation (Y-5400) are based on the loads used in the different Code equations.

M_L^{**} , M_N^{**} , M_T^{**} , Q_1^{**} , Q_2^{**} , and W^{**} are absolute values of maximum loads occurring simultaneously under all service loading conditions.

Y-5400 EVALUATION PROCEDURE

The loads on the attachment cause stresses in the pipe wall. Equations are provided in Y-5410(a) to determine these stresses. The attachment stresses are then added to the piping system stresses at the attachment. The piping system stresses are determined by NC-3652 eq. (8), NC-3653.1 eq. (9), and NC-3653.2 eqs. (10), (10a), and (11) for straight pipe. The Code equations including the attachment stress terms are given in Y-5410(b). The attachment stresses, S_{MT} , S_{NT} , and S_{PT} are to be calculated for the loading conditions corresponding to NC-3652 eq. (8), NC-3653.1 eq. (9), and NC-3653.2 eqs. (10), (10a), and (11). For example, in calculating S_{MT} for use in NC-3652 eq. (8), W , M_L , M_N , Q_1 , Q_2 , and M_T are the loads on the attachment due to weight and other sustained loads. While NC is used below, the same rules apply for ND piping.

There are additional equations given in Y-5410(c) for all weld configurations and Y-5420(b) for fillet weld or partial penetration weld attachments, that also must be checked for attachment stresses. These are based on the absolute values for maximum loads occurring simultaneously under all service loading conditions.

Y-5410 ANALYSIS OF ATTACHMENT WELDED TO PIPE WITH A FULL PENETRATION WELD

(a) Calculate the stresses: S_{MT} , S_{NT} , S_{PT} , and S_{NT}^{**}

$$S_{MT} = \frac{B_W W}{A_T} + \frac{B_N M_N}{Z_T} + \frac{B_L M_L}{Z_T} + \frac{2Q_1}{A_T} + \frac{2Q_2}{A_T} + \frac{B_T M_T}{J} \quad (1)$$

$$S_{NT} = \frac{C_W W}{A_T} + \frac{C_N M_N}{Z_T} + \frac{C_L M_L}{Z_T} + \frac{2Q_1}{A_T} + \frac{2Q_2}{A_T} + \frac{C_T M_T}{J} \quad (2)$$

$$S_{PT} = K_T(S_{NT}) \quad (3)$$

$$S_{NT}^{**} = \frac{C_W W^{**}}{A_T} + \frac{C_N M_N^{**}}{Z_T} + \frac{C_L M_L^{**}}{Z_T} + \frac{2Q_1^{**}}{A_T} + \frac{2Q_2^{**}}{A_T} + \frac{C_T M_T^{**}}{J} \quad (4)$$

(b) The following modified Code equations shall be satisfied, where all terms except attachment stresses are defined in NC-3652.

(1) NC-3652 eq. (8) becomes

$$S_{SL} = B_1 \frac{PD_o}{2t_n} + B_2 \frac{M_A}{Z} + S_{MT} \leq 1.5S_h \quad (\text{NC-8})$$

where $B_1 = 0.5$ and $B_2 = 1.0$ for straight pipe.

(2) NC-3653.1 eq. (9) becomes

$$S_{OL} = B_1 \frac{P_{\max} D_o}{2t_n} + B_2 \frac{M_A + M_B}{Z} + S_{MT} \quad (\text{NC-9})$$

$\leq 1.8S_h$ but not greater than $1.5S_y$ for Level A and B loadings

$\leq 2.25S_h$ but not greater than $1.8S_y$ for Level C loadings

$\leq 3.0S_h$ but not greater than $2.0S_y$ for Level D loadings

(3) NC-3653.2 eq. (10) becomes

$$S_E = \frac{iM_C}{Z} + \frac{S_{PT}}{2} \leq S_A \quad (\text{NC-10})$$

(4) NC-3653.2 eq. (10a) becomes

$$\frac{iM_D}{Z} + \frac{S_{PT}}{2} \leq 3.0S_c \quad (\text{NC-10a})$$

(5) NC-3653.2 eq. (11) becomes

$$S_{TE} = \frac{PD_o}{4t_n} + 0.75i\left(\frac{M_A}{Z}\right) + i\left(\frac{M_C}{Z}\right) + S_{MT} + \frac{S_{PT}}{2} \leq \left(S_h + S_A\right) \quad (\text{NC-11})$$

In eq. (NC-11), S_{MT} is the same as used in eq. (1)(NC-8), and S_{PT} is the same as used in eq. (3)(NC-10).

(c) In addition to the Code equations, the following equations shall also be satisfied.

$$S_{NT}^{**} \leq 2S_y \quad (5)$$

$$\frac{2Q_1^{**}}{A_T} + \frac{2Q_2^{**}}{A_T} + \frac{M_T^{**}}{J} \leq S_y \quad (6)$$

Y-5420 ANALYSIS OF ATTACHMENT WELDED TO PIPE WITH FILLET WELDS OR PARTIAL PENETRATION WELDS

(a) The requirements of Y-5410 shall be met.

(b) The following additional requirements shall be met.

$$\frac{W^{**}}{A_w} + \frac{M_L^{**}}{Z_{wl}} + \frac{M_N^{**}}{Z_{wn}} + \frac{\left[Q_1^{**2} + Q_2^{**2}\right]^{1/2}}{A_w} + \frac{M_T^{**}}{Z_{wt}} \leq 2S_y \quad (7)$$

$$\left\{ \left(\frac{W^{**}}{A_w} \right)^2 + 4 \left[\left(\frac{Q_1^{**} + Q_2^{**}}{A_w} \right) + \frac{M_T^{**}}{Z_{wt}} \right]^2 \right\}^{1/2} \leq S_y \quad (8)$$

Y-5430 DIFFERENTIAL METAL TEMPERATURE EFFECTS

The potential for increased stress at the attachment welds, which may occur as a result of differential metal temperatures between the attachment and the run, should be considered in the design evaluation.

Y-5500 ANALYSIS DOCUMENTATION

Analyses demonstrating compliance with this Article shall be included in the Design Report for the piping system.

NONMANDATORY APPENDIX Z

ARTICLE Z-1000 INTERRUPTION OF CODE WORK

Z-1100 INTRODUCTION

Z-1110 SCOPE

The scope of this Appendix is confined to those situations where the Code activities of a Certificate Holder are interrupted prior to completion of all Code assigned responsibilities. When this occurs, all completed in-process work must be clearly documented to ensure remaining activities and Code responsibilities are readily identifiable.

Z-1200 DEFINITIONS

interruption of Code activities: the cessation of Code work by a Certificate Holder prior to completion of all assigned Code responsibilities. The cessation may result from project suspension or transfer of Code responsibilities from one Certificate Holder to another Certificate Holder.

completed in-process work: a Code activity on an item, part, subassembly, appurtenance, component, or support which has been accepted by the ANI for the Certificate Holder that performed the Code activity, but which requires additional work before the item, part, subassembly, appurtenance, component, or support can be stamped with the applicable Certification Mark. Examples of completed in-process work include design activities, an individual weld or a weld pass, heat treatment of weld joints, and examination of welds or items after heat treatment.

Z-1300 DOCUMENTATION

(a) A memorandum of understanding shall be approved by the parties involved which documents the status of Code activities and acceptance of transfer responsibilities, where applicable. The memorandum shall be approved by the existing Certificate Holder and its ANI, or alternatively, a Certificate Holder who has permitted its Certificate to expire and its current ANI, employee by an Accredited Nuclear AIA, and the Certificate Holder that will provide continuity of responsibility and its ANI, where applicable. When the work is being performed at a nuclear power plant site, the memorandum shall also be approved by

the Owner and the jurisdictional enforcement authority having jurisdiction at the nuclear power plant site, if applicable.

(b) The memorandum of understanding shall reference Code Data Reports for completed work. For completed in-process work, Code Data Reports may also be used to document the status of Code activities. Additionally, the status of completed in-process work may also be indicated on drawings, diagrams, or other means and referenced in the memorandum.

Z-1400 OTHER CONSIDERATIONS

(a) If a quality program such as that used by a Certificate Holder is utilized during the period of interruption, a review of completed work will not be required by the Certificate Holder assuming responsibility for completion of Code activities. The Certificate Holder assuming responsibility or its Authorized Nuclear Inspector may require evidence that an acceptable quality program has been in place.

(b) Once completed work or completed in-process work is accepted by an ANI, it meets Code requirements and need not be scrutinized by another Certificate Holder or ANI.

(c) To meet Code requirements, completed work shall be documented on the applicable Code Data Report and stamped as applicable.

(d) Completion of the applicable Code Data Report and application of the Certification Mark, where applicable, expressly signifies the Code status of completed work and completed in-process work.

(e) Where the N-type Certificate Holder has permitted the Certificates to expire, and has returned the Certification Mark to the Society, and the Owner plans to contract with a new Certificate Holder to complete construction of the nuclear facility, the expired Certificate Holder may apply to the Society for Temporary Certificates of Authorization and such Certificates and applicable Certification Mark shall be issued by the Society subject to the following conditions:

(1) The scope of the certificates shall be limited to the Code Edition and Addenda to which the nuclear plant has been docketed. No new Code work may be performed under these Temporary Certificates. Repair welding of material imperfections and existing welds shall not be performed.

(2) An Accredited Authorized Nuclear Inspection Agency shall be employed to review the completed work previously performed and monitor and verify compilation and completion of all originally required documentation such as Data Report Forms and supporting Data Packages.

(3) The ANI shall certify all partial Data Reports and authorize the Temporary Certificate Holder to stamp the previously completed work with the appropriate Certification Mark.

(4) The Quality Assurance Program previously accepted by the Society shall be implemented (NCA-8140) and any revisions to the program shall be acceptable to the Authorized Nuclear Inspection Agency. All required revisions to the Quality Assurance Manual shall be reviewed and accepted by the ANI prior to implementation. The revised program shall govern all activities required to document and stamp all previously completed work.

(5) A survey or audit by the Society shall be required for the issuance of the requested Certificates and Stamps to the Expired Certificate Holder. Code activities performed prior to the issuance of the Temporary Certificates shall be subject to the acceptance of the Inspector (NCA-8153).

(6) The Owner shall apply to the Society for an Owners Certificate (NCA-8162), and the evaluation interview by the Society shall include a review of the Owner's planned scope of activities to be performed under the Temporary Certificates. A complete list of all work

remaining to be documented and stamped shall be provided to the AIA prior to the completion of all work. The Regulatory Authority and the Jurisdictional and Enforcement Authority (if applicable) shall be notified of the completion of these activities.

(7) The term of the Temporary Certificates shall be for 1 year, and may be extended once by the Society upon receipt of a request submitted by Certified mail for an additional period not to exceed 1 year. Subsequent renewals shall be treated as renewals of active Certificates.

(8) The Owner shall maintain the Owner's Certificate in accordance with existing Code requirements until all Code activity has been completed, and the N-3 Data Report Form has been completed and filed [NCA-8180(c)].

(9) The Temporary Certificates and Certification Mark shall be returned to the Society when all previously completed work has been documented and stamped.

Z-1500 RESUMPTION OF CODE ACTIVITIES

Resumption of Code activities may be undertaken at any time by updating the original memorandum of understanding documenting the status of Code activities. The updated memorandum shall be approved by any new parties involved such as the Certificate Holder assuming responsibility for completion of Code activities, the Certificate Holder's ASME Accredited Authorized Nuclear Inspection Agency, and the organization contracting for the completion of Code activities. For Code activities to be performed at a nuclear plant site, the Owner and jurisdictional and enforcement authority (when applicable) shall also approve the updated memorandum of understanding.

NONMANDATORY APPENDIX AA

GUIDANCE FOR THE USE OF U.S. CUSTOMARY AND SI UNITS IN THE ASME BOILER AND PRESSURE VESSEL CODE

AA-1000 USE OF UNITS IN EQUATIONS

The equations in this Nonmandatory Appendix are suitable for use with either the U.S. Customary or the SI units provided in [Mandatory Appendix XXIV](#), or with the units provided in the nomenclature associated with that equation. It is the responsibility of the individual and organization performing the calculations to ensure that appropriate units are used. Either U.S. Customary or SI units may be used as a consistent set. When necessary to convert from one system of units to another, the units shall be converted to at least three significant figures for use in calculations and other aspects of construction.

AA-2000 GUIDELINES USED TO DEVELOP SI EQUIVALENTS

The following guidelines were used to develop SI equivalents:

(a) SI units are placed in parentheses after the U.S. Customary units in the text.

(b) In general, separate SI tables are provided if interpolation is expected. The table designation (e.g., table number) is the same for both the U.S. Customary and SI tables, with the addition of suffix "M" to the designator for the SI table, if a separate table is provided. In the text, references to a table use only the primary table number (i.e., without the "M"). For some small tables, where interpolation is not required, SI units are placed in parentheses after the U.S. Customary unit.

(c) Separate SI versions of graphical information (charts) are provided, except that if both axes are dimensionless, a single figure (chart) is used.

(d) In most cases, conversions of units in the text were done using hard SI conversion practices, with some soft conversions on a case-by-case basis, as appropriate. This was implemented by rounding the SI values to the number of significant figures of implied precision in the existing U.S. Customary units. For example, 3,000 psi has an implied precision of one significant figure. Therefore, the conversion to SI units would typically be to 20 000 kPa. This is a difference of about 3% from the "exact" or soft conversion of 20 684.27 kPa. However, the precision of the conversion was determined by the Committee on a case-by-case basis. More significant digits were included

in the SI equivalent if there was any question. The values of allowable stress in Section II, Part D generally include three significant figures.

(e) Minimum thickness and radius values that are expressed in fractions of an inch were generally converted according to the following table:

Fraction, in.	Proposed SI Conversion, mm	Difference, %
$\frac{1}{32}$	0.8	-0.8
$\frac{3}{64}$	1.2	-0.8
$\frac{1}{16}$	1.5	5.5
$\frac{3}{32}$	2.5	-5.0
$\frac{1}{8}$	3	5.5
$\frac{5}{32}$	4	-0.8
$\frac{3}{16}$	5	-5.0
$\frac{7}{32}$	5.5	1.0
$\frac{1}{4}$	6	5.5
$\frac{5}{16}$	8	-0.8
$\frac{3}{8}$	10	-5.0
$\frac{7}{16}$	11	1.0
$\frac{1}{2}$	13	-2.4
$\frac{9}{16}$	14	2.0
$\frac{5}{8}$	16	-0.8
$\frac{11}{16}$	17	2.6
$\frac{3}{4}$	19	0.3
$\frac{7}{8}$	22	1.0
1	25	1.6

(f) For nominal sizes that are in even increments of inches, even multiples of 25 mm were generally used. Intermediate values were interpolated rather than converting and rounding to the nearest mm. See examples the following table. [Note that this table does not apply to nominal pipe sizes (NPS), which are covered below.]

Size, in.	Size, mm
1	25
$1\frac{1}{8}$	29
$1\frac{1}{4}$	32
$1\frac{1}{2}$	38
2	50
$2\frac{1}{4}$	57
$2\frac{1}{2}$	64
3	75
$3\frac{1}{2}$	89
4	100
$4\frac{1}{2}$	114
5	125
6	150
8	200

Table continued

Size, in.	Size, mm
12	300
18	450
20	500
24	600
36	900
40	1 000
54	1 350
60	1 500
72	1 800

Size or Length, ft	Size or Length, m
3	1
5	1.5
200	60

(g) For nominal pipe sizes, the following relationships were used:

U.S. Custom- ary Practice	SI Practice	U.S. Custom- ary Practice	SI Practice
NPS 1/8	DN 6	NPS 20	DN 500
NPS 1/4	DN 8	NPS 22	DN 550
NPS 3/8	DN 10	NPS 24	DN 600
NPS 1/2	DN 15	NPS 26	DN 650
NPS 3/4	DN 20	NPS 28	DN 700
NPS 1	DN 25	NPS 30	DN 750
NPS 1 1/4	DN 32	NPS 32	DN 800
NPS 1 1/2	DN 40	NPS 34	DN 850
NPS 2	DN 50	NPS 36	DN 900
NPS 2 1/2	DN 65	NPS 38	DN 950
NPS 3	DN 80	NPS 40	DN 1000
NPS 3 1/2	DN 90	NPS 42	DN 1050
NPS 4	DN 100	NPS 44	DN 1100
NPS 5	DN 125	NPS 46	DN 1150
NPS 6	DN 150	NPS 48	DN 1200
NPS 8	DN 200	NPS 50	DN 1250
NPS 10	DN 250	NPS 52	DN 1300
NPS 12	DN 300	NPS 54	DN 1350
NPS 14	DN 350	NPS 56	DN 1400
NPS 16	DN 400	NPS 58	DN 1450
NPS 18	DN 450	NPS 60	DN 1500

(h) Areas in square inches (in.²) were converted to square mm (mm²) and areas in square feet (ft²) were converted to square meters (m²). See examples in the following table:

Area (U.S. Customary)	Area (SI)
1 in. ²	650 mm ²
6 in. ²	4 000 mm ²
10 in. ²	6 500 mm ²
5 ft ²	0.5 m ²

(i) Volumes in cubic inches (in.³) were converted to cubic mm (mm³) and volumes in cubic feet (ft³) were converted to cubic meters (m³). See examples in the following table:

Volume (U.S. Customary)	Volume (SI)
1 in. ³	16 000 mm ³
6 in. ³	100 000 mm ³
10 in. ³	160 000 mm ³
5 ft ³	0.14 m ³

(j) Although the pressure should always be in MPa for calculations, there are cases where other units are used in the text. For example, kPa is used for small pressures. Also, rounding was to one significant figure (two at the most) in most cases. See examples in the following table. (Note that 14.7 psi converts to 101 kPa, while 15 psi converts to 100 kPa. While this may seem at first glance to be an anomaly, it is consistent with the rounding philosophy.)

Pressure (U.S. Customary)	Pressure (SI)
0.5 psi	3 kPa
2 psi	15 kPa
3 psi	20 kPa
10 psi	70 kPa
14.7 psi	101 kPa
15 psi	100 kPa
30 psi	200 kPa
50 psi	350 kPa
100 psi	700 kPa
150 psi	1 MPa
200 psi	1.5 MPa
250 psi	1.7 MPa
300 psi	2 MPa
350 psi	2.5 MPa
400 psi	3 MPa
500 psi	3.5 MPa
600 psi	4 MPa
1,200 psi	8 MPa
1,500 psi	10 MPa

(k) Material properties that are expressed in psi or ksi (e.g., allowable stress, yield and tensile strength, elastic modulus) were generally converted to MPa to three significant figures. See example in the following table:

Strength (U.S. Customary)	Strength (SI)
95,000 psi	655 MPa

(1) In most cases, temperatures (e.g., for PWHT) were rounded to the nearest 5°C. Depending on the implied precision of the temperature, some were rounded to the nearest 1°C or 10°C or even 25°C. Temperatures colder than 0°F (negative values) were generally rounded to the nearest 1°C. The examples in the table below were created by rounding to the nearest 5°C, with one exception:

Temperature, °F	Temperature, °C
70	20
100	38
120	50
150	65
200	95
250	120
300	150
350	175
400	205
450	230
500	260
550	290
600	315
650	345
700	370
750	400
800	425
850	455
900	480
925	495
950	510
1,000	540
1,050	565
1,100	595
1,150	620
1,200	650
1,250	675
1,800	980
1,900	1 040
2,000	1 095
2,050	1 120

AA-3000 SOFT CONVERSION FACTORS

The following table of “soft” conversion factors is provided for convenience. Multiply the U.S. Customary value by the factor given to obtain the SI value. Similarly, divide the SI value by the factor given to obtain the U.S. Customary value. In most cases it is appropriate to round the answer to three significant figures.

U.S. Customary	SI	Factor	Notes
in.	mm	25.4	...
ft	m	0.3048	...
in. ²	mm ²	645.16	...
ft ²	m ²	0.09290304	...
in. ³	mm ³	16,387.064	...
ft ³	m ³	0.02831685	...
U.S. gal.	m ³	0.00378541-2	...
U.S. gal.	liters	3.785412	...
psi	MPa (N/mm ²)	0.0068948	Used exclusively in equations
psi	kPa	6.894757	Used only in text and for nameplate
psi	bar	0.06894757	...
ft-lb	J	1.355818	...
°F	°C	$\frac{5}{9} \times (°F - 32)$	Not for temperature difference
°F	°C	$\frac{5}{9}$	For temperature differences only
R	K	$\frac{5}{9}$	Absolute temperature
lbm	kg	0.4535924	...
lbf	N	4.448222	...
in-lb	N·mm	112.98484	Use exclusively in equations
ft-lb	N·m	1.3558181	Use only in text
ksi√in.	MPa√m	1.0988434	...
Btu/hr	W	0.2930711	Use for boiler rating and heat transfer
lb/ft ³	kg/m ³	16.018463	...

NONMANDATORY APPENDIX BB METALLIC BRAIDED FLEXIBLE HOSE FOR CLASS 2 AND 3 SERVICE

ARTICLE BB-1000 SCOPE

BB-1100 RULES

This Appendix provides the rules for the construction of metallic braided flexible hose for use in Section III, Division 1, Class 2 and 3 applications.

The braided hose consists of a convoluted inner sheath pressure boundary, with outer reinforcing braided wire welded to end pieces.

ARTICLE BB-2000 MATERIAL

BB-2100 SHEATHS, END PIECES, AND BRAIDS

The inner sheath and end pieces shall be fabricated from materials conforming to NC/ND-2000.³⁹

The braid shall be made from stainless steel heat-resisting wire conforming to ASTM specification A580-98. Only wire of material types listed under SA-479 in Tables 1A, 1B, and 3 of Section II, Part D may be used for the braided sheath.

ARTICLE BB-3000 DESIGN

BB-3100 DESIGN FACTORS

(a) The design shall consider operating and design loads and movements including differential movement, vibration, and inertia effects when applicable.

(b) The Certificate Holder that manufactures the hose shall establish the pressure and temperature rating of the hose assembly by calculations and tests in accordance with the Expansion Joint Manufacturers Association Standard (EJMA). The minimum and maximum temperatures shall be within the limits listed in Section II, Part D. The hose rating shall be equal to or exceed the piping system design pressure and temperature.

(c) The rating of the hose assembly shall have a minimum design margin of 3.5 against burst and leakage.

BB-3200 GENERAL DESIGN REQUIREMENTS

(a) Braided flexible hoses with the convoluted hose element having a length to outside diameter ratio (L/D_o) of 3 or less shall comply with all the requirements of NC/ND-3649 for bellows expansion joints.

(b) Braided flexible hoses with the convoluted hose element having a length to outside diameter ratio (L/D_o) greater than 3 shall comply with the requirements of NC/ND-3649 with the following exceptions:

(1) The flexible wire braid shall act as an axial restraint for the hose and vibration dampener, and provide columnar stability against squirm.

(2) Subsections NC/ND-3649.1(a), NC/ND-3649.2(d), and NC/ND-3649.4(c) are not applicable.

(c) The Certificate Holder that manufactures the hose shall supply to the N-type Certificate Holder the maximum allowable end loads, inertia loads, displacements, minimum curvature between the hose ends, and the spring rates for the flexible hose. The calculated loads and displacements for the piping system shall be less than those supplied by the Certificate Holder that manufactures the hose.

BB-3300 SPECIAL DESIGN REQUIREMENTS

(a) Flow-induced vibration at design flow shall be evaluated and a sleeve specified when required per NC/ND-3649.2(f).

(b) The piping system layout, anchorage, guiding, and support shall avoid the imposition of end displacement, vibratory motions, or forces for the hose length, other than those for which the hose is designed.

(c) The hose end on one side of the installation shall not be oriented longitudinally concentric with the other hose end unless the minimum design curvature between the hose ends recommended by the Certificate Holder that manufactures the hose is maintained.

(d) Either annealed or cold-finished wire may be used for the wire braid, but the allowable stresses shall be those listed in Tables 1A, 1B, and 3 of Section II, Part D for the annealed or solution-treated condition for SA-479 of the same material type as the wire being specified.

As a minimum, the required number of strands shall be established from the equation

$$N = \frac{F}{SA \cos \alpha}$$

where

A = cross-sectional area of one wire, in.²

F = end load due to pressure, lbs acting on the effective area of the connector

N = minimum number of wires

S = allowable stress at the rated temperature, psi

α = wrap angle (the acute angle supported by the strand and the axis of the connector), as shown in

[Figure BB-3300-1](#)

$$F = \frac{\pi}{4} \left(\frac{D_o + D_i}{2} \right)^2 P$$

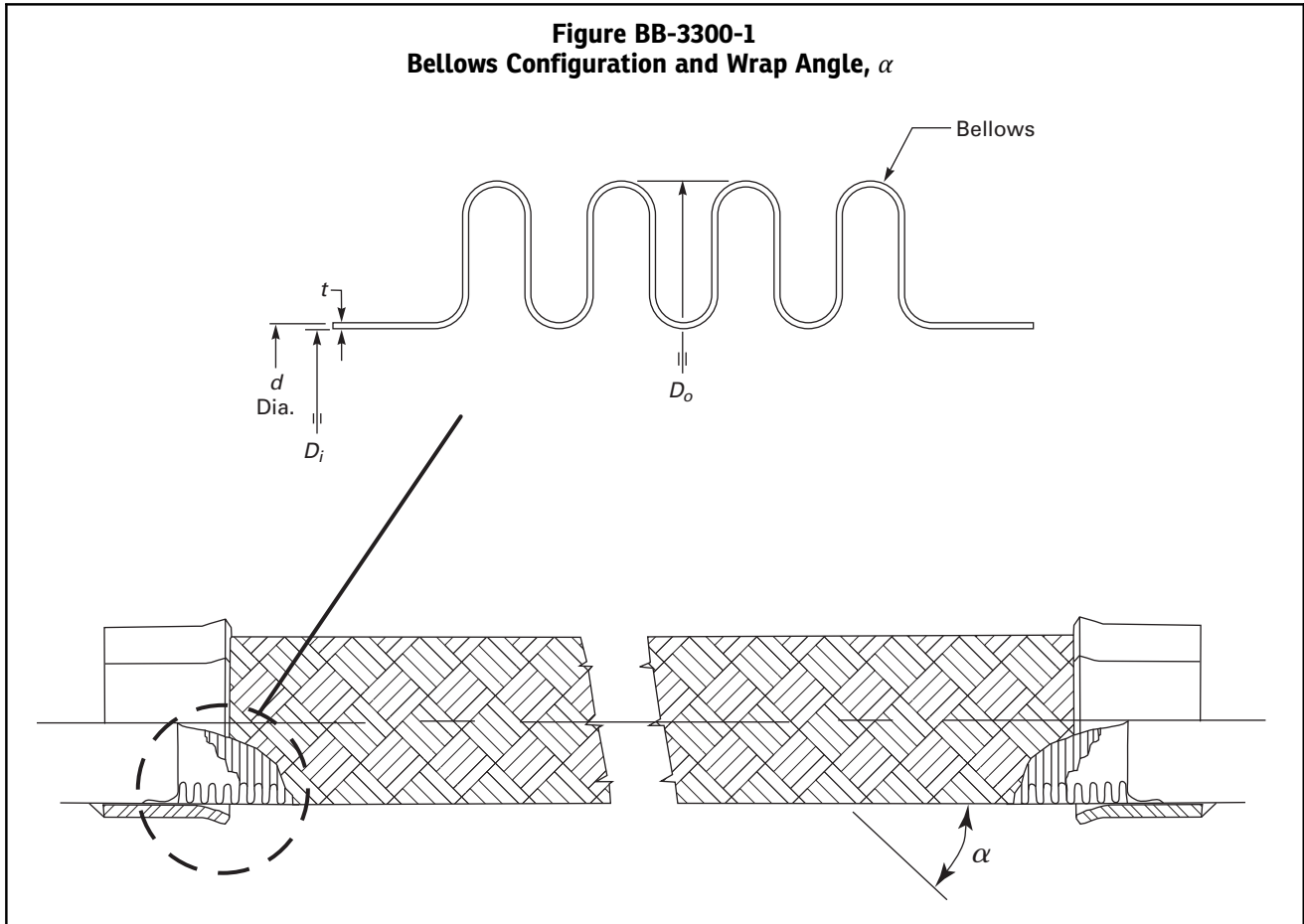
where

D_i = ID of convolution, [Figure BB-3300-1](#)

D_o = OD of convolution, [Figure BB-3300-1](#)

P = rated pressure, psi

Figure BB-3300-1
Bellows Configuration and Wrap Angle, α



ARTICLE BB-4000 FABRICATION

BB-4100 REQUIREMENTS

(a) All wire braid strands shall be welded to the welding collars of the convoluted hose connector per NC/ND-4800.

(b) All welds shall conform to NC/ND-4800.

(c) The end pieces, whether welded, flanged, or threaded, shall conform to NC/ND-3132 and NC/ND-3612.

(d) The inner sheath shall be attached to the end pieces utilizing circumferential welds of a butt type having full penetration through the thickness of the inner sheath.

ARTICLE BB-5000 EXAMINATION

BB-5100 PROCEDURES

- (a)* Examination requirements for expansion joints in accordance with NC/ND-5700 shall be satisfied.
- (b)* All butt welds greater than or equal to NPS 4 shall be examined by RT or UT, in accordance with NC/ND-5000.
- (c)* All butt welds less than NPS 4 shall be examined by PT or MT, as applicable, in accordance with NC/ND-5000.
- (d)* The wire-strand-to-collar welds shall be visually examined to detect unconnected wires.

ARTICLE BB-6000 TESTING

BB-6100 HYDROSTATIC AND PNEUMATIC TESTING

All braided flexible hoses shall be hydrostatically tested in accordance with NC/ND-6000, except that test pressure shall be not less than 1.5 times the design pressure at room temperature, and shall be so noted on Data Report Form NPP-1. Alternatively, the hose may be pneumatically

tested when submerged in water. The test of hoses with inlet piping connections of NPS 4 and smaller need not be witnessed by the Inspector. The Inspector's review of the Certificate Holder's test records will be his authority to sign the Data Report. Installed hose assemblies are subject to the piping system hydrostatic test.

ARTICLE BB-7000 CERTIFICATION

BB-7100 PROVISIONS

(a) The braided flexible hose shall be stamped with the Certification Mark with NPT Designator in accordance with NCA-8230. The design pressure and temperature shall be part of the required information per NCA-8211.

(b) The NPP-1 Data Report shall include the pressure and temperature rating, maximum allowable end loads, inertia loads, displacement, spring loads, and other loads specified by the design specifications.

(c) A single NPP-1 Data Report Form may be used for a lot of no more than 25 hose assemblies of the same nominal pipe size, length, and geometry.

(d) The N-type Certificate Holder shall demonstrate in the Design Report for the piping system that includes the hose, that hose assembly manufacturer's limits are not exceeded.

NONMANDATORY APPENDIX CC ALTERNATIVE RULES FOR LINEAR PIPING SUPPORTS

ARTICLE CC-1000 INTRODUCTION

CC-1100 INTRODUCTION

CC-1110 SCOPE AND GENERAL REQUIREMENTS

CC-1111 Scope of This Appendix

This Appendix provides alternative rules to the requirements of Subsections NCA and NF, for Linear Piping Supports that are constructed to ANSI/AISC N690-1994, "Specification for the Design, Fabrication, and Erection of Steel Safety-Related Structures for Nuclear Facilities," including Supplement 2, ANSI/AISC N690-1994 (R2004) S2, and the requirements of this Appendix.

CC-1112 General Requirements

(a) When this Appendix is used, the Owner or his designee shall provide a Design Specification (NCA-3252, NCA-3255) that permits the use of this Appendix and

identifies the loadings and combinations of loadings for which the supports are to be designed. The Design Specification shall contain sufficient detail to provide a complete basis for construction of the supports.

(b) The Owner or his designee shall perform a documented review of the calculations for each support to determine that all the specified loadings have been evaluated and that the acceptance criteria provided in this Appendix and in ANSI/AISC N690 have been considered. The responsibility for the method of analysis and the accuracy of the calculations remains with the designer.

(c) The supports shall be constructed under a Quality Assurance Program that meets the requirements specified by the Owner.

ARTICLE CC-2000 MATERIALS

CC-2100 MATERIAL REQUIREMENTS

CC-2110 SUPPORT MATERIAL

(a) Material shall conform to ANSI/AISC N690.

(b) In those instances where material may be subject to lamellar tearing, such as through-thickness transmission of tensile loads in thick plates, the Design Specification shall include the requirement that the material be ultrasonically examined in accordance with ANSI/AISC N690, Section Q1.4.

(c) The requirements of ANSI/AISC N690 Section Q1.4 or Q2.2 do not apply to bearings, bushings, gaskets, hydraulic fluids, seals, shims, slide plates, retaining rings, wear shoes, springs, washers, wire rope, spring end plates, thread locking devices, cotter pins, sight glass assemblies, spring hanger travel and hydro stops, nameplates,

nameplate attachment devices, or for compression dynamic stops used as stops⁴⁰ for seismic and other dynamic loads that are designed primarily for compressive loading and are not connected to the pressure boundary and do not provide support of the pressure boundary. Requirements, if any, for these materials shall be stated in the Design Specification.

CC-2120 CERTIFICATION OF MATERIALS

Copies of Certified Material Test Reports, certified reports of tests made by the fabricator or a qualified testing laboratory, or Certificates of Compliance as required by ANSI/AISC N690 shall be furnished to the Owner or designee for all supports provided under these requirements.

ARTICLE CC-3000 DESIGN

CC-3100 DESIGN REQUIREMENTS

CC-3110 GENERAL DESIGN REQUIREMENTS

The design requirements that shall be satisfied in the elastic analysis for any Design and Level A through D Service Loadings stated in the Design Specification are those given in Table Q1.5.7.1 of ANSI/AISC N690 and the additional requirements of [CC-3120](#), [CC-3130](#), and [CC-3140](#).

CC-3120 DESIGN LOAD CONSIDERATIONS

(a) For the design of supports, the stress caused by the restraint of free-end displacements of components and piping, such as thermal expansion and relative anchor displacements, shall be considered as a primary stress.

(b) The Normal, Severe, Extreme, Abnormal, Abnormal Severe, and Abnormal Extreme load categories of ANSI/AISC N690 shall be correlated to the appropriate Design and Service Loadings identified in Design Specification as shown in [Table CC-3120-1](#).

CC-3130 SPECIAL DESIGN CONSIDERATIONS

(a) The stress limit coefficients in [Table CC-3120-1](#) are not intended for control of deformation. When required by the Design Specification, deformation control shall be considered separately.

(b) As an alternative to design by analysis, Design by Load Rating as defined in NF-3380 and NF-3480 may be used.

(c) Plastic design per Part 2 of ANSI/AISC N690 shall not be used.

CC-3140 DESIGN LIMITS

CC-3141 Stress Limits

(a) The rules and stress limits that shall be satisfied for any Test Loading stated in the Design Specification shall be *Load Combination 1* (Table Q1.5.7.1 in ANSI/AISC N690), multiplied by a stress limit coefficient of 1.33.

(b) The Stress Limit Coefficients in ANSI/AISC N690 shall be modified as shown in [Table CC-3120-1](#).

(c) Thermal stresses within the support as defined by NF-3121.11 need not be evaluated.

(d) Shear stress limit shall not exceed $0.42S_u$, at temperature.

(e) To avoid column buckling, the allowable compressive stresses shall be limited to two-thirds of the critical buckling stress.

Table CC-3120-1
Correlation of Service Loadings and Stress
Limit Coefficients

Loading Category	Stress Limit Coefficient	Service Level
Normal	1.0	Design and Level A
Severe	1.33	Level B
Extreme	1.5	Level C
Abnormal	1.5	Level C
Abnormal severe	1.5	Level C
Abnormal extreme	1.7	Level D

ARTICLE CC-4000 FABRICATION

CC-4100 FABRICATION REQUIREMENTS

(a) The requirements for welding qualifications given in NF-4300 may be used for any portion of fabrication and installation in lieu of those specified in ANSI/AISC N690,

provided all such welding is performed by an N-Type Certificate Holder and the qualification is performed under the QA program applicable to the certificate.

(b) Thermal cutting is prohibited on quenched and tempered steels.

ARTICLE CC-5000 EXAMINATION

CC-5100 EXAMINATION REQUIREMENTS

CC-5110 REQUIRED EXAMINATION OF WELDS

CC-5111 Examination of Welds on Supports for Class 1 Piping

(a) All full penetration butt welded joints in Supports for Class 1 piping shall be nondestructively examined by radiographic or ultrasonic methods in accordance with ANSI/AISC N690.

(b) All other welded joints in Supports for Class 1 piping shall be nondestructively examined by liquid penetrant or magnetic particle methods in accordance with ANSI/AISC N690.

CC-5120 QUALIFICATION AND CERTIFICATION OF NONDESTRUCTIVE EXAMINATION PERSONNEL

CC-5121 NDE Personnel Requirements

All NDE personnel shall be qualified to the requirements of ANSI/AISC N690, and all nondestructive examinations shall be supervised or performed by an AWS Certified Welding Inspector.

CC-5122 Alternative Rules for Nondestructive Examination Personnel

As an alternative to [CC-5121](#), N-Type Certificate Holders may use NF-5500 for personnel qualification, provided the qualification is performed under the QA program applicable to the certificate.

ARTICLE CC-8000 NAMEPLATES, STAMPING WITH CERTIFICATION MARK, AND DATA REPORTS

CC-8100 GENERAL REQUIREMENTS

Nameplates, stamping with Certification Mark, and Data Reports are not required for Linear Piping Supports designed and constructed to the requirements of this Appendix.

(13)

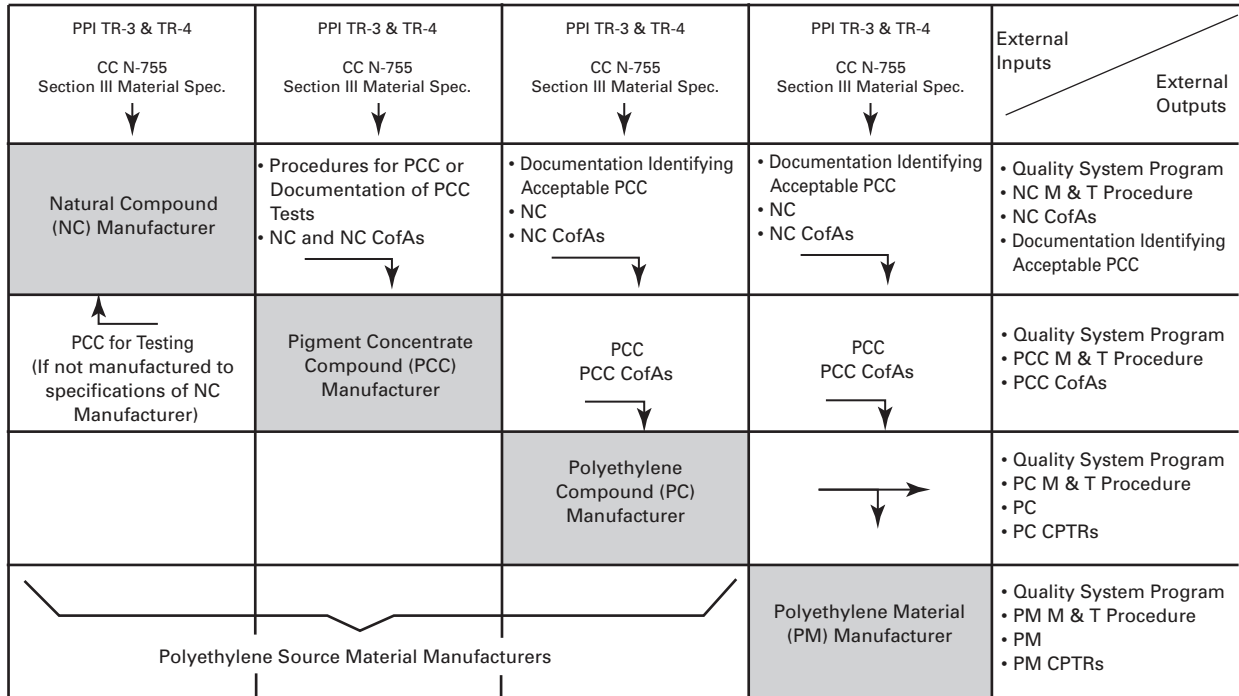
NONMANDATORY APPENDIX DD POLYETHYLENE MATERIAL ORGANIZATION RESPONSIBILITIES DIAGRAM

ARTICLE DD-1000 INTRODUCTION

DD-1100 SCOPE

This Nonmandatory Appendix contains [Figure DD-1100-1](#) depicting inputs and outputs that govern activities of Polyethylene Material Organization.

Figure DD-1100-1
Polyethylene Material Organization Responsibilities Per NCA-3970



GENERAL NOTES:

- (a) This figure depicts the following:
- (1) external inputs (top row) that govern activities of Polyethylene Material Organizations
 - (2) outputs from one Polyethylene Material Organization that are as follows:
 - (a) either inputs to other Polyethylene Material Organizations
 - (b) or are external outputs in the form of products or quality documentation
- (b) The definitions are as follows:
- (1) CofA = Certificate of Analysis
 - (2) CPTR = Certified Polyethylene Test Report
 - (3) M & T = Manufacturing & Testing
 - (4) Polyethylene Source Material Manufacturer = Natural Compound Manufacturer, Pigment Concentrate Compound Manufacturer, or Polyethylene Compound Manufacturer
- (c) Polyethylene material supplier and polyethylene service suppliers are not shown.

(13)

NONMANDATORY APPENDIX EE STRAIN-BASED ACCEPTANCE CRITERIA DEFINITIONS AND BACKGROUND INFORMATION

ARTICLE EE-1000 STRAIN INFORMATION

EE-1100 DEFINITIONS

EE-1110 EQUIVALENT (TRUE) PLASTIC STRAIN

The equivalent (true) plastic strain is analogous to the equivalent stress ($\bar{\sigma}$), which is the von Mises stress, where the superscript prime indicates the deviatoric stress tensor (σ'_{ij}) with ij reflecting tensor notation

$$\bar{\sigma} = \left(\frac{3}{2} \sigma'_{ij} \sigma'_{ij} \right)^{1/2}$$

Similarly, the equivalent plastic strain rate ($\dot{\epsilon}_{eq}^p$) is defined in terms of the plastic strain rate tensor ($\dot{\epsilon}_{ij}^p$)

$$\dot{\epsilon}_{eq}^p = \left(\frac{2}{3} \dot{\epsilon}_{ij}^p \dot{\epsilon}_{ij}^p \right)^{1/2}$$

The equivalent plastic strain (ϵ_{eq}^p) is the integral of the equivalent plastic strain rate over the time interval t

$$\epsilon_{eq}^p = \int_0^t \dot{\epsilon}_{eq}^p dt$$

Combining the above two equations results in the following equation for the equivalent plastic strain:

$$\epsilon_{eq}^p = \int_0^t \left(\frac{2}{3} \dot{\epsilon}_{ij}^p \dot{\epsilon}_{ij}^p \right)^{1/2} dt$$

Nonmandatory Appendix FF uses the above definition⁴¹ of equivalent plastic strain. Equivalent plastic strain is a common variable calculated in nonlinear finite element software codes, and details on its derivation are typically available in the documentation associated with those codes as well as other engineering treatises on strain and plasticity.

The equivalent plastic strain is a cumulative, positive scalar quantity, nondecreasing strain measure that takes into account the entire deformation history. Since the

driving mechanism for plastic distortion is the transformation of externally supplied energy (e.g., kinetic energy in the case of impacts and drops) into plastic work, the equivalent plastic strain is intrinsically a better indication of the material condition than any instantaneous stress determination.

In summary, the equivalent plastic strain cumulatively combines the strain state/history into a meaningful scalar value for comparative purposes. This value is especially useful in reducing the voluminous strain output created by finite element and other computer-solution methods commonly in use today. The fact that the equivalent plastic strain does not indicate whether tension or compression has caused the strain is appropriate and consistent with the triaxiality factor usage herein, which also conservatively ignores any strengthening effects due to compression.

EE-1120 QUASI-STATIC TENSILE TESTING

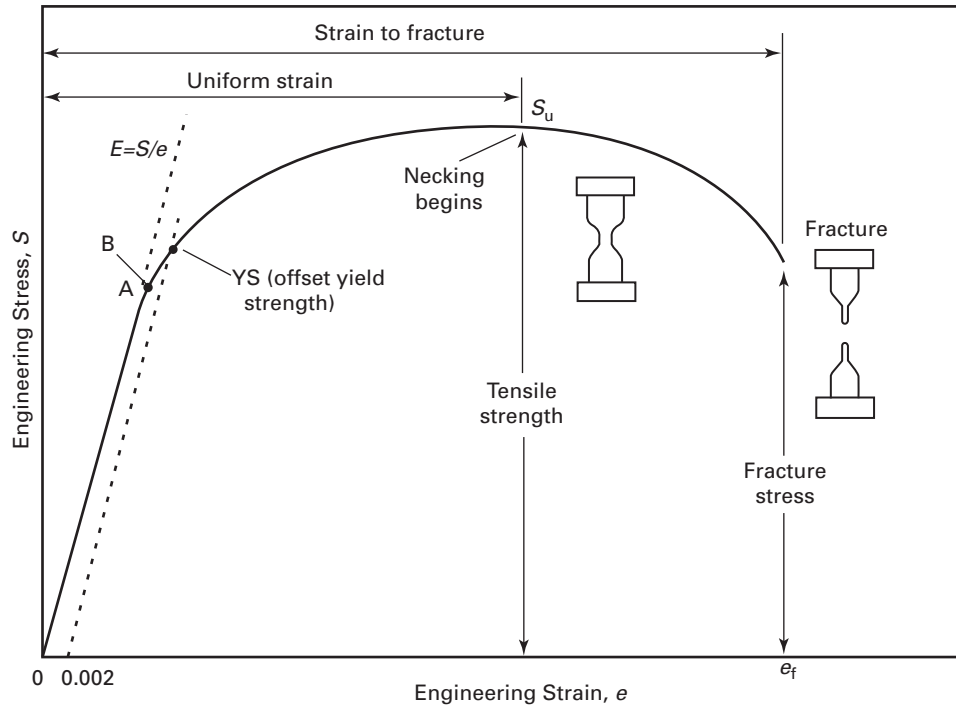
Stress-strain curves are usually presented as either

(a) engineering stress-strain curves, in which the original specimen cross-sectional area is used to determine stress, and the change in length divided by the original length determines strain, or

(b) true stress-strain curves, where the instantaneous cross-sectional area of the specimen is used to determine the stress and the strain.

To document a quasi-static tensile test, an engineering stress-strain curve is developed from the load-displacement measurements made during the test on the test specimen (Figure EE-1120-1, typical for ductile material). The engineering stress, S , plotted on this curve is the average longitudinal stress in the tensile specimen obtained by dividing the load, P , by the original specimen cross-sectional area, A_o . The engineering strain, e , plotted on the curve is the average linear strain obtained by dividing the change in gauge length, ΔL , of the specimen by the original length, L_o .

Figure EE-1120-1
Typical Engineering Tensile Stress–Strain Curve [1]



$$S = P/A_0$$

$$e = \Delta L/L_0$$

The elastic limit, shown as point B in Figure EE-1120-1, is the greatest stress the material can withstand without measurable permanent strain remaining after complete release of load. The yield strength, shown as point YS in Figure EE-1120-1, is the stress required to produce a small, specified amount of inelastic deformation. The usual definition of this property is the offset yield strength determined by the stress corresponding to the intersection of the linear elastic segment of the stress–strain curve offset by a specified strain of 0.2% ($e = 0.002$). The tensile strength (or ultimate strength), S_u , is the corresponding stress where the maximum load that the material can withstand occurs. This also corresponds to the point where the specimen becomes unstable (onset of necking) and necks down during the remaining course of the tensile test. Necking is the point of rapid, localized reduction of cross-sectional area of a specimen under tensile loading. It is disregarded in calculating engineering stress but is taken into account in determining true stress. Complete fracture (failure point) of the specimen follows necking. In order to accurately determine the complete stress–

strain curve, the total axial displacement of the test specimen must be accurately measured, using a large displacement extensometer or other appropriate means.

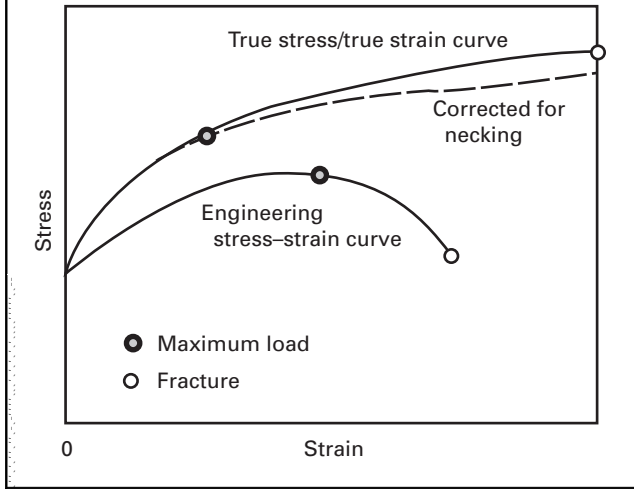
The engineering stress–strain curve does not give the most accurate indication of the deformation characteristics of a material because it is based on the original specimen dimensions that actually change continuously during the test. Also, at the point of ultimate load, necking begins and the cross-sectional area of the specimen decreases rapidly, and the load required to continue deformation lessens, as implied in Figure EE-1120-1. The average stress based on the original area likewise decreases, and produces the downturn in the engineering stress–strain curve beyond the point of maximum load. In reality, the material continues to strain-harden to fracture, so that the stress required to produce further deformation should also increase. If the true stress, based on the actual cross-sectional area of the specimen is used, the stress–strain curve increases continuously to fracture. If the strain measurement is also based on instantaneous measurement, the curve obtained is the true stress–strain curve as illustrated in Figure EE-1120-2.

Up to the point of necking, the true stress, σ_t , may be expressed in terms of engineering stress by

$$\sigma_t = S(e + 1)$$

Up to the onset of necking, the true strain, ϵ_t , may be determined from the engineering strain, by

Figure EE-1120-2
Comparison of Engineering and True
Stress-Strain Curves (Ref. [1])



$$\varepsilon_t = \ln(e + 1)$$

where “ln” is the natural log.

Beyond the point of maximum load (necking region), the true strain is based on the actual current area, A , and is expressed as follows:

$$\varepsilon_t = \ln(A_0/A)$$

and the true stress is based on the load and actual current area

$$\sigma_t = P/A$$

At the point of fracture, the true strain (ε_t) and true stress (σ_t) are thus expressed:

$$\varepsilon_t = \ln(A_0/A_f) \text{ and } \sigma_t = P_f/A_f$$

where

A_f = the area at fracture

P_f = the load at fracture

The true stress-strain curve beyond the point of onset of necking (the maximum load or uniform strain limit) is further complicated by the development of radial and hoop stresses in the necking region. The average axial or nominal stress given by $\sigma_t = P/A$ is not the true equivalent uniaxial stress because the hoop and radial stresses are not zero. Beyond the onset of necking, the nominal stress is often corrected to get the true equivalent uniaxial stress using a Bridgman Correction factor [2], which is dependent upon specimen geometry (corrected curve in Figure EE-1120-2).

EE-1130 TRUE UNIFORM STRAIN LIMIT

$\varepsilon_{uniform}$: true uniform strain limit or the true strain just prior to the onset of necking in a uniaxial tensile test [the true strain at the maximum load (tensile strength)] at the coincident average through-wall temperature of the base or weld material

In a uniaxial tensile test, a specimen experiences uniform straining along the entire gauge (or reduced area) length until the maximum load is reached (at the tensile strength).

EE-1140 TRUE FRACTURE STRAIN LIMIT

$\varepsilon_{fracture}$: true strain at fracture in a uniaxial tensile test at the coincident average through-wall temperature of the base or weld material

Beyond the uniform strain limit, further deformation occurs in a relatively small volume of material as the specimen local cross-sectional area reduces (“necks”). Note that even though a typical stainless steel true stress-strain curve shows a large area under the curve from the onset of necking to fracture, the volume of material associated with that necked region is smaller than the volume of material in the specimen’s gauge length — resulting in only a small additional amount of energy absorbed (compared to the energy absorbed in the entire gauge length volume up to the uniform strain limit) prior to the specimen breaking. Engineering fracture strain is the strain at fracture, denoted as e_f in Figure EE-1120-1.

Failure in stress-based acceptance criteria may be based on, for example, exceeding the specified minimum material yield strength. The acceptance criteria would then limit the stresses to a level below the material yield strength to provide a safety margin against yielding. In these strain-based acceptance criteria, large plastic deformations are expected and allowed in the structure. The goal of the acceptance criteria is to maintain the allowable leakage rate identified in the Design Specification. Therefore, failure would be defined as plastic strain levels that cause breach of the structure or through-wall crack formation.

The materials to which these strain-based acceptance criteria are limited, 304, 304L, 316, or 316L (or dual-marked 304/304L or 316/316L), are ductile austenitic stainless steels. The potential for leakage of these stainless steels would not be expected until the through-wall strains reach a level of significant material necking near the true fracture limit.

EE-1150 USE OF TRIAXIALITY FACTOR

Triaxiality Factor (TF) as used in the strain-based acceptance criteria of FF-1140 is defined as:

$$TF = \frac{(\sigma_1 + \sigma_2 + \sigma_3)}{\sqrt{\frac{1}{2}[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]}}$$

where σ_1 , σ_2 , and σ_3 are the principal stresses at the location under evaluation.

Many theories and formulations have been proposed to account for material damage under multi-axial stress and plastic strain conditions. The details of these theories and formulations will not be discussed herein. However, the chosen methodology employed in these criteria, as discussed in EPRI Report NP-1921 (ref. [3]), uses the triaxiality factor in a simple formulation.

The equivalent plastic strain correctly calculates the strain condition on the Von Mises yield surface in the absence of damage (crack initiation or flaw propagation). However, real materials experience damage under plastic deformation, which is accelerated when multi-axial tensile stress conditions exist. The concept of a stress triaxiality factor was first proposed by Davis and Connelly (ref. [4]), and has been widely discussed since (e.g., refs. [3] and [5]). As discussed above, the stress triaxiality factor is based on the principal stresses, and is the sum of the three principal stresses (first stress invariant) divided by the effective (Von Mises) stress at a location. The strain at failure in a general case is related to the uniaxial tension failure strain by:

$$\epsilon_{\text{general failure case}} = \epsilon_{\text{uniaxial tension failure}} / \text{TF}$$

Therefore, determining the allowable strain limits by accounting for stress triaxiality effects will promote damage prevention and the achievement of specified leakage rates.

A triaxiality factor of 1.0 represents uniaxial tension, a factor of 2.0 represents biaxial tension, and greater than 2.0 indicates a triaxial tension state. Triaxiality factors of less than 1.0 are due to compressive principal stresses in one or more directions. Examples of triaxiality factors for varying normalized principal stresses are shown in Table EE-1150-1.

Table EE-1150-1
Examples of Triaxiality Factor Calculations

Normalized Principal Stresses			Calculated	Description
σ_1	σ_2	σ_3	TF	
1	0	0	1	Uniaxial tension
1	1	0	2	Biaxial tension
1	1	$\frac{1}{4}$	3	Triaxial tension
1	$\frac{1}{2}$	$\frac{1}{2}$	4	Triaxial tension
1	1	$\frac{1}{2}$	5	Triaxial tension
1	1	1	∞	Triaxial tension
1	-1	0	0	Tension/compression
1	$-\frac{1}{2}$	0	0.378	Tension/compression
1	1	-1	0.5	Biaxial tension/ compression
1	-1	-1	-0.5	Tension/compression
-1	-1	-1	$-\infty$	Triaxial compression

Note that the triaxiality factor does not indicate that plastic straining is occurring — it merely indicates the associated stress state. Therefore, only the triaxiality factor calculated while plastic straining is occurring is applicable in the strain-based acceptance criteria. When plastic straining has stopped, the triaxiality factor simply indicates the elastic stress state.

The strain-based acceptance criteria indicate that a minimum triaxiality factor of 1.0 (positive value) must be used. Using a minimum factor of 1.0 conservatively ignores the potential strengthening/damage inhibiting effects of compressive stresses.

EE-1200 BACKGROUND INFORMATION

EE-1210 ACHIEVING DESIRED LEAKAGE RATES

Drop testing research has been performed (refs. [6], [7]) on 18 in. (457 mm) diameter, $\frac{3}{8}$ in. (9.53 mm) wall thickness and 24 in. (610 mm) diameter, $\frac{1}{2}$ in. (12.7 mm) wall thickness canisters, 10 ft and 15 ft (3 m and 4.5 m) in length, weighing about 6,000 lb to 10,000 lb (26.7 kN to 44.5 kN). The canisters were made of SA-312 and SA-240 austenitic stainless steel (316L or 316/316L). These canisters were dropped from 30 ft (9 m) onto a rigid, flat surface, impacting at a variety of orientations or from lower drop distances onto plate edges or ends of round bars. After drop testing, the worst-strained canisters of each test sequence were helium leak tested and found to have leakage rates of less than 10^{-7} std cc/sec, which is considered leaktight (ref. [8]). Certain canister test specimens had computer predicted maximum equivalent plastic strains in excess of the average through-wall thickness strain limits of FF-1140. This gives a clear indication that the average through-wall allowable equivalent plastic strain of 67% of the true uniform strain limit does indeed maintain desired leakage rates, up to and including leaktight conditions. Other 24 in. (610 mm) diameter, $\frac{1}{2}$ in. (12.7 mm) wall thickness canisters (Multi-Canister Overpacks) made from SA-312 and SA-240 austenitic stainless steel (304L or 304/304L) and weighing approximately 18,000 lb (80 kN) were drop tested and had similar results (ref. [7]).

During this canister drop testing effort, one canister of the 18 in. (457 mm) diameter geometry was dropped from 30 ft (9 m) with an integral skirt experiencing significant deformation. The analytically predicted peak equivalent plastic strain was 107% on the outside surface, far beyond the maximum strain limit established in FF-1140. Nondestructive examination (liquid penetrant) was performed on this surface of the test canister and no indications were identified, demonstrating that no cracks had initiated, even at these high strain levels.

Finally, over 500 impact tests have been performed during research efforts to quantify strain rate effects for austenitic stainless steel materials (refs. [9], [10]) including both 304/304L and 316/316L materials. This testing

included base and weld materials at -20°F (-29°C), room, 300°F (149°C), and 600°F (316°C) temperature conditions. One hundred seventy-one of those impact tests strained the test specimens to levels between 66% and 100% of the established uniform strain limit. These test specimens, with strains above the average (through-wall) equivalent plastic strain limits established by the strain-based acceptance criteria, had no visible cracks in the surface of the material. Forty-nine of these specimens were strained to 90% of the uniform strain limit or higher. Hence, the results of this testing have demonstrated that the 304/304L and 316/316L austenitic stainless steels are sufficiently ductile to experience strains exceeding the limits established in [FF-1140](#) without developing crack initiation concerns. No crack formation means these materials can achieve the desired leakage rates. The limits imposed by the strain-based acceptance criteria provide additional margins of safety.

These test results demonstrate that austenitic stainless steels have an ability to experience high strains without crack formation and still achieve desired leakage rates down to 10^{-7} std cc/sec. This is true under uniaxial loading and also multiaxial loading conditions because of the incorporation of the triaxiality factor into the strain-based acceptance criteria.

EE-1220 MATERIAL PROPERTIES FOR INELASTIC EVALUATIONS

Appropriate consideration of the actual inelastic response of the containment materials is vital to the accurate prediction of strains and the proper implementation of the strain-based acceptance criteria. Material properties in an aged condition (potential material degradation throughout the design life) must be considered. Temperature effects on material properties must also be incorporated into any inelastic evaluations using the strain-based acceptance criteria. Determination of containment strains at multiple temperature conditions (not just at the maximum or minimum temperature) is required to assure the resulting maximum product of the equivalent plastic strain and the associated triaxiality factor (maximum strain response) has been determined because an energy-limited event may occur at various temperature conditions.

Different heats of the same material specification can have wide ranges of material properties. This mixture of material properties can complicate the determination of the maximum strain response of the containment. Therefore, these varying material properties must be properly defined in the complete containment analysis model at the specific locations of use, reflecting an as-built condition. When calculating the resulting strains using inelastic analyses, the material properties are varied at locations for various components fabricated from different material heats to assure that the maximum energy is input into the containment boundary.

In order to properly implement the strain-based acceptance criteria, the necessary material property data [beyond those available in Section II or provided on the Certified Material Test Reports (CMTRs⁴²)] are defined as the true stress-strain curve (for computer model input), the true uniform strain limit, and the true fracture strain limit, for both base and weld materials at the temperatures necessary to define the coincident conditions of the dynamic energy-limited event.

Two options for determining appropriate material properties used in the inelastic analyses are provided in [EE-1221](#) and [EE-1222](#). In either option, the Code user can assume material properties for preliminary analytical evaluations as long as the final Design Report adequately reflects the temperature-dependent base and welded material properties actually used in fabrication of the containment and their specific locations of use. In one extreme, the assumptions can reflect a wide range of material properties and material use locations via numerous analytical evaluations so that the procurement of fabrication material is simplified and the analytical work does not have to be regenerated when the properties of the actual fabrication material become known. The other extreme is to assume material properties that reflect a very limited range of values (minimizing the analytical evaluations) but the material procurement process must then impose additional requirements in order to obtain material that properly reflects that limited range of material properties and material use locations. Regardless of the option chosen, evaluation of the material heats actually used in fabrication of the containment on a location-by-location basis considering temperature must be made for the final Design Report.

For explanation purposes, it is assumed that the user performs preliminary analyses prior to the actual initiation of fabrication. It then becomes incumbent on the Certificate Holder responsible for the final Design Report to justify the material properties used or to reanalyze the containment as necessary. This justification process is discussed below.

EE-1221 ASME-Specified Material (Base and Weld) Strength Properties

At this time, the ASME-specified true stress-strain curves (reflecting minimum yield and ultimate tensile strength values) and the true uniform and fracture strain limits (reflecting a 98% exceedance probability) associated with the strain-based acceptance criteria of [Nonmandatory Appendix FF](#) are still under development, so this option does not currently exist. Until these data become available, the user must develop the necessary material data based on tensile testing (see [EE-1222](#)) and their use justified in the final Design Report.

EE-1222 Actual Material (Base and Weld) Strength Properties

The second option permits the use of tensile test data reflecting the specific material properties from the actual material heats used in the containment fabrication, provided the necessary material properties (true stress-strain curves, true uniform strain limits, and true fracture strain limits) are correctly obtained. There are nine basic steps associated with this option.

Step 1. Perform preliminary analyses. If necessary, perform preliminary analyses using either limited analyses (requiring assurance that the materials used are within the acceptable range of tolerance for the material property data used in the inelastic analyses) or numerous analyses (sufficient to address both a wide range of potential material property data and location uses). If material is not available when the preliminary inelastic analyses are performed, assumed material properties may be used.

Step 2. Procure material and obtain appropriate CMTRs.

Step 3. Perform material testing. Perform the necessary base and welded material testing at the various temperatures of interest. These tensile test data (true stress-strain curves and appropriate strain limits) can be based on the mean (average) of three tests for each unique material heat or weld, at the coincident temperature conditions of the event, performed in compliance with the requirements of SA-370. Weld material test specimens shall consist of weld material through the entire volume of the reduced section of the test specimen.

Step 4. Check validity of material testing. The appropriate CMTR data should then be compared to the actual corresponding mean material test data and evaluated for accuracy ($\pm 10\%$ tolerance). This check assures that the tensile testing was properly performed. If the test data are beyond this tolerance, the three tensile tests must be repeated. If the mean values of the repeated tests match the CMTR data within the established tolerance, then the repeated test data should be used. If the repeated test data matches the first set of test data within the $\pm 10\%$ tolerance, then the set of data whose mean values of yield and tensile strengths more closely match the CMTR data should be used.

Step 5. Develop true stress-strain curves. Using the appropriate mean test data, develop the true stress-strain curves and determine the true uniform and true fracture strain limits at each temperature for each material heat.

Step 6. Compare material properties. The heat-specific true stress-strain curves for the actual material being used in fabrication should be compared to the material input of the inelastic analysis performed, on a location-by-location basis. If the true stress-strain curves vary from the corresponding values used in any existing inelastic analysis by more than 15% based on location use, the complete containment design must be reanalyzed and reevaluated.

Step 7. Perform final analyses. Perform new inelastic analyses as needed or redo any inelastic analyses that do not satisfy the material property data tolerance check. These analytical evaluations must adequately reflect the heat-specific material data at the appropriate locations within each containment model for the various appropriate temperature conditions. Resulting maximum strain responses can then be determined as the highest values from all of the pertinent analyses performed. This reanalysis effort is required because stronger materials (material properties greater than the ASME provided minimums) may alter the strain response of the containment. For the final inelastic analyses for the final Design Report, the assumed properties must eventually be reconciled with the actual material used in the fabrication.

Step 8. Establish strain limits for the strain-based acceptance criteria. Using the best mean material test data as established in Step 5, for each unique material heat and temperature, determine the 98% exceedance probability true uniform strain limits and true fracture strain limits for use in the strain-based acceptance criteria (98% exceedance probability is defined as the mean value minus two standard deviations).

Step 9. Compare predicted strains with strain criteria. Compare the predicted maximum strain responses to the strain-based acceptance criteria in [FF-1140](#). The tested true uniform and true fracture strain limits (adjusted to a 98% exceedance probability) must be used to establish the strain-based acceptance criteria limits in [FF-1140](#).

EE-1230 BASE VERSUS WELDED MATERIAL RESPONSES

Prior drop testing experience of full-scale Department of Energy spent nuclear fuel canisters (refs. [\[6\]](#), [\[7\]](#)) indicated there was no significant variation or discontinuity in the deformation responses of the canister wall when impact occurred directly onto a canister weld. A limited number of strain rate impact tests performed at the Idaho National Laboratory (ref. [\[11\]](#)) of welded material at elevated temperatures were performed prior to the completion of that material's quasi-static tensile testing. These first strain rate impact tests used the same drop weight and drop height as was used for the corresponding base material. The resulting strain rate responses of the welded material test specimens appeared to be very similar to the base material. However, some of the welded material test specimens necked or broke where the base material test specimens had not. The conclusion reached was that the welded material had a lower uniform strain limit and fracture strain than the base material. After completion of all the quasi-static tensile testing, this indeed turned out to be the case.

[Figures EE-1230-1](#) and [EE-1230-2](#) show, for 304/304L and 316/316L respectively, quasi-static tensile test results of base and welded materials at 300°F (149°C) temperature conditions. These representative engineering stress-strain plots illustrate that the uniform strain and

fracture strain limits for the welded material are lower than the associated base material. Figures EE-1230-3 and EE-1230-4 are comparative strain history plots of 304/304L and 316/316L base and welded material impact tests performed at -20°F (-29°C). These impact tests used the same drop weight, drop height, and test specimen geometry. These plots illustrate how similar the base and welded material were in terms of strain rate response. The plots also show that the welded material absorbs the impact energy with a lower maximum strain than the base material (i.e., welded material is stronger than base material). However, as discussed above, the welds fail at lower strain levels than the base material (i.e., welded material is less ductile than the base material).

Based on the strain rate range achieved, the welded material test specimens responded very similar to the base material test specimens. Therefore, one would expect the strain rate factors to be similar (ref. [11]). This permits the structural analyst to use the same strain rate factor data for both base and weld materials when incorporating strain rate effects into finite element models. However, the structural analyst must be fully aware that the welds can have lower uniform strain and fracture strain limits.

EE-1240 PROPER FINITE ELEMENTS MODELS

In order to properly implement the strain-based acceptance criteria, it is imperative that accurate strains be calculated. In turn, this means accurate analysis models of the containment are needed. These strain-based

acceptance criteria should be applicable only to strain results from Quality Models. A Quality Model is a model that adheres to the guidance set forth in the ASME Computational Modeling Guidance Document for Explicit Dynamics Software (currently being developed by the ASME BPV III Special Working Group on Computational Modeling For Explicit Dynamics), or using a model with suitable convergence and sensitivity studies already completed.

Issues that need to be properly addressed in a Quality Model (or equivalent) include

- (a) acceptable element types
- (b) proper element aspect ratios
- (c) adequate element transitioning
- (d) appropriate finite element meshing
- (e) acceptable modeling of welded and bolted joints
- (f) correct material property input
- (g) proper consideration of contact points, friction, gaps, and boundary conditions
- (h) realistic application of loading
- (i) correct solution technique
- (j) proper calculation of the Triaxiality Factor as defined in FF-1140
- (k) useful and correct strain output

EE-1250 STRAIN RATE EFFECTS

Strain rate effects refer to a material's ability to absorb increased amounts of strain energy during dynamic events, beyond that determined during quasi-static tensile testing (EE-1120). Resulting strain predictions within a

Figure EE-1230-1
Quasi-static Tensile Test Results for 304/304L Base and Welded Material at 300°F (149°C)

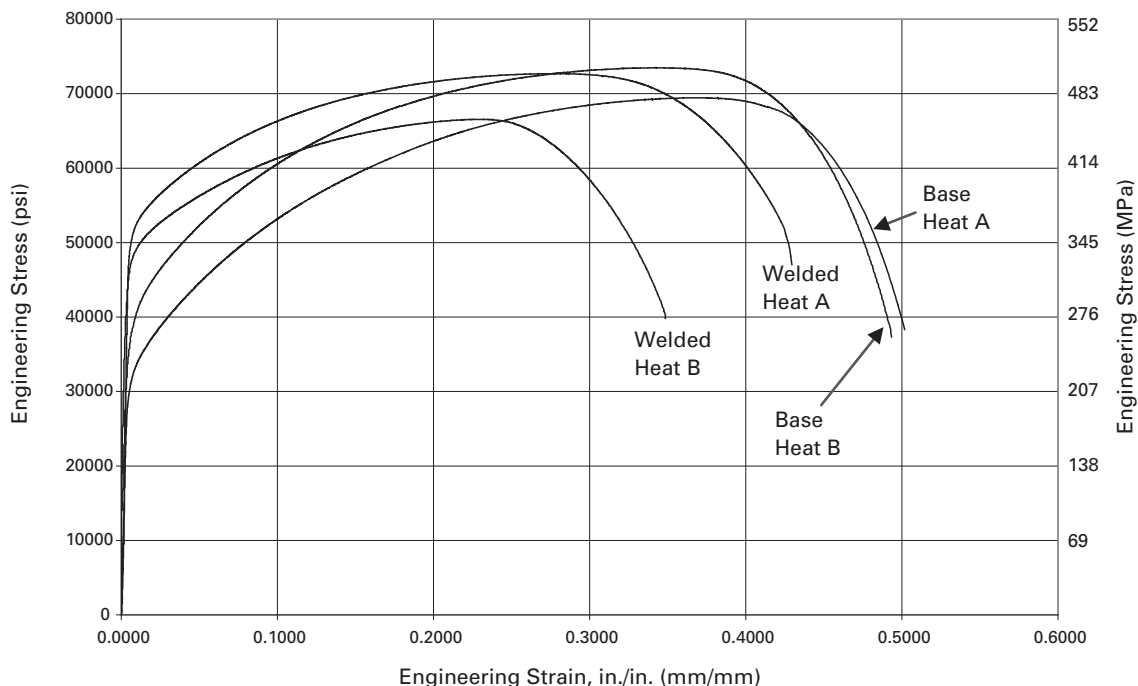


Figure EE-1230-2
Quasi-static Tensile Test Results for 316/316L Base and Welded Material at 300°F (149°C)

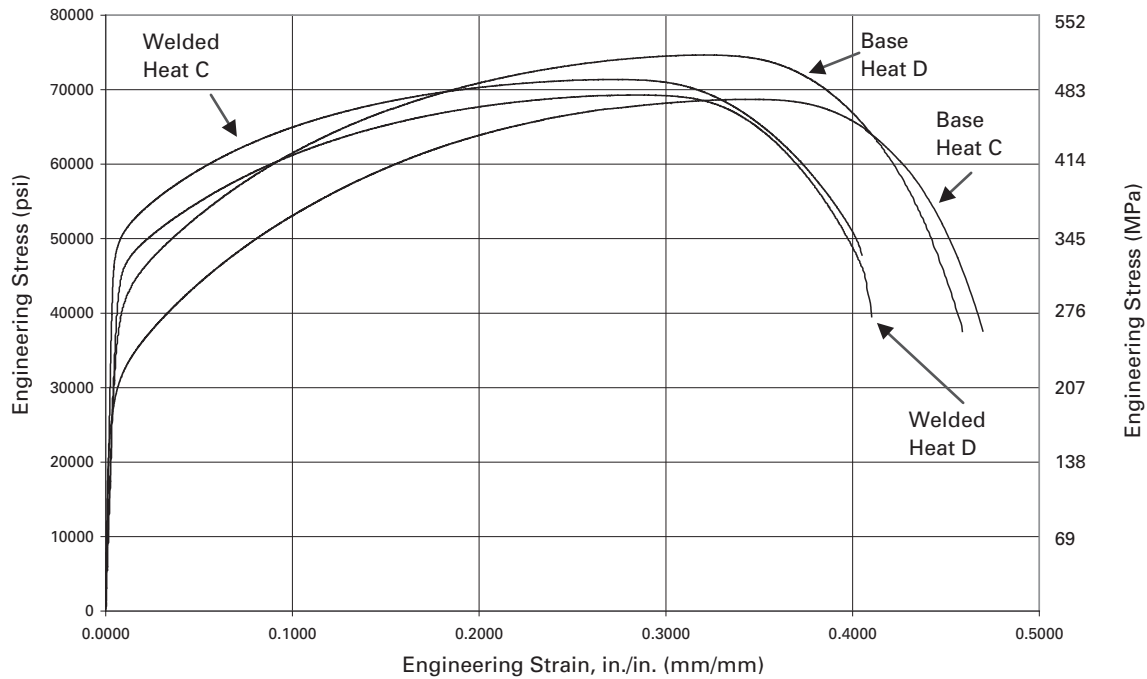


Figure EE-1230-3
Comparison of Base and Welded 304/304L Material to Identical Impact Tests at -20°F (-29°C)

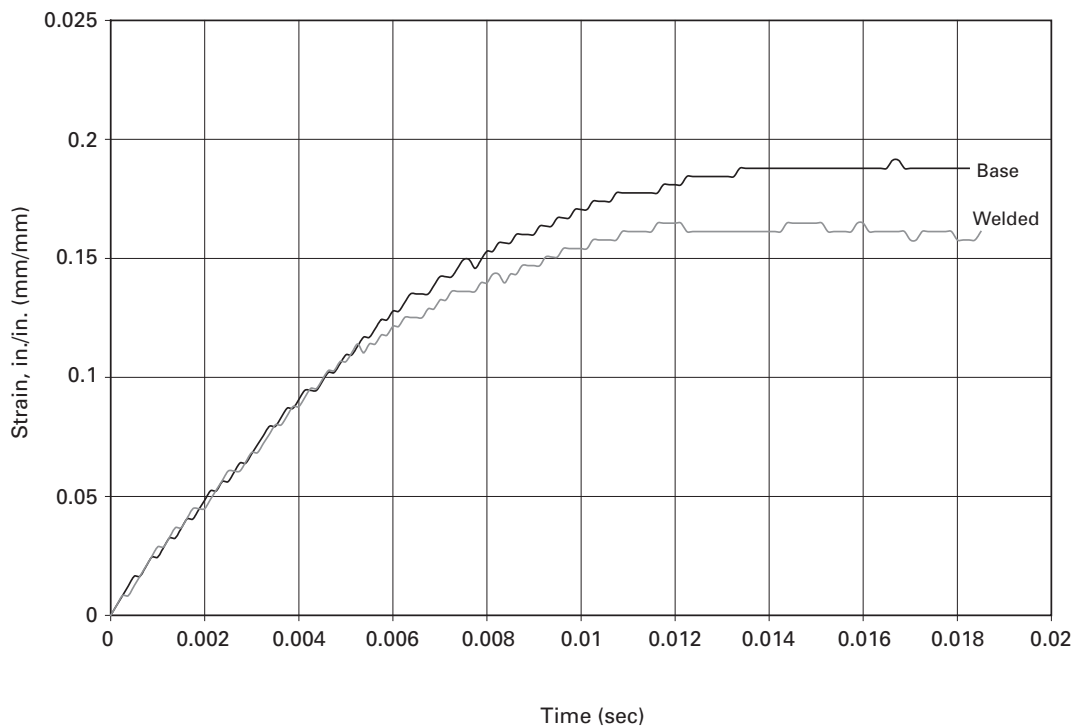
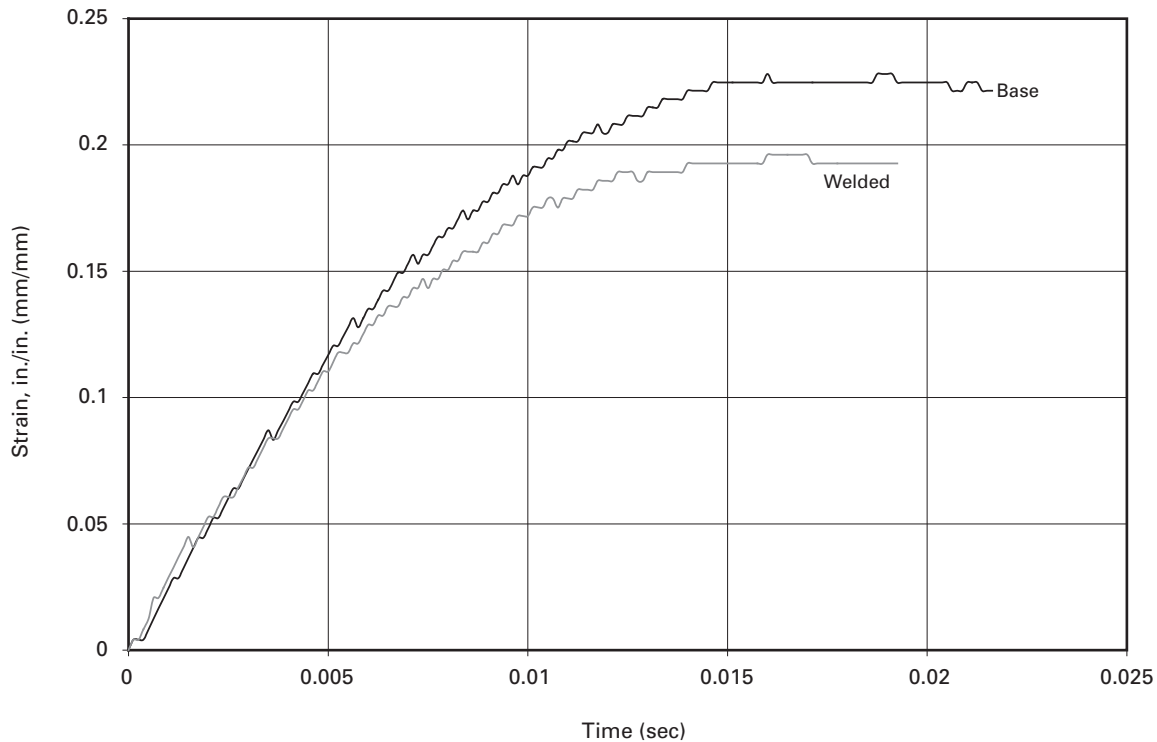


Figure EE-1230-4
Comparison of Base and Welded 316/316L Material to Identical Impact Tests at -20°F (-29°C)



containment design may change when strain rate effects are properly considered. In order to address this behavior, additional inelastic analyses need to be completed in order to assure that the resulting maximum strain responses have been determined.

The strain rate testing mentioned above (refs. [9], [10]) has provided data to support the development of strain rate elevated true stress-strain curves for both 304/304L and 316/316L austenitic stainless steel base and weld materials that account for strain rate strengthening up to a strain rate of nearly 40 in./in./sec (1 000 mm/mm/s), depending upon temperature [highest strain rates achieved at 600°F (316°C) with existing test equipment were in the lower 20s in./in./sec (500s mm/mm/s)]. Table EE-1250-1 provides a summary of the impact testing results and illustrates the magnitude of the increased energy absorption capacity that can exist in these ductile stainless steel materials at varying temperatures. The strain rate factor is applied to all of the stress values of a true stress-strain curve (developed from data obtained using SA-370 methods) to reflect the increased energy absorption capacity.

The results of the impact testing demonstrated that the effects of strain rate decreased with increasing temperature. Base and weld materials appeared to behave similarly during impact testing (same factors), but welded materials were not able to achieve strains as high as their

Table EE-1250-1
Factors for Specified Strain Rates

Strain Rate in./ in./sec (mm/ mm/s)	-20°F (-29°C)	Room Temperature	300°F (149°C)	600°F (316°C)
304/304L Stainless Steel				
5	1.333	1.235	1.166	1.043
10	1.361	1.278	1.210	1.094
22	1.428	1.381	1.316	1.217
316/316L Stainless Steel				
5	1.275	1.265	1.162	1.040
10	1.296	1.281	1.187	1.070
22	1.346	1.321	1.247	1.140

associated base material (demonstrated for both quasi-static and dynamic impact loadings). The uniform strain limits for both weld material and base material did not appear to change from the values established during quasi-static tensile testing for the strain rate range considered (up to 22 in./in./sec (550 mm/mm/s)).

Using the strain rate data developed as material input into analytical models of the impact tensile tests performed resulted in analytical predictions that showed marked improvements when compared to material input reflecting quasi-static tensile test results. Hence, considering the range of factors in [Table EE-1250-1](#), the methodology of [FF-1145](#) reflects the trend indicated by the data.

EE-1260 REFERENCES

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- [2] P. W. Bridgman, *The Stress Distribution at the Neck of a Tension Specimen*, Transactions of the A. S. M., Twenty-fifth Annual Convention, Chicago, Illinois, October 18-22, 1943.
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- [11] D. K. Morton and R. K. Blandford, "Impact Tensile Testing of Stainless Steels at Various Temperatures", EDF-NSNF-082, Idaho National Laboratory, March 2008.

(13)

NONMANDATORY APPENDIX FF STRAIN-BASED ACCEPTANCE CRITERIA FOR ENERGY-LIMITED EVENTS

ARTICLE FF-1000 ACCEPTANCE CRITERIA

FF-1100 STRAIN-BASED ACCEPTANCE CRITERIA

The strain-based acceptance criteria presented in this Nonmandatory Appendix utilize a variety of terminology that must be clearly understood. In addition, background information that explains various strain concepts, material property variations, and strain responses is useful in order to properly implement these strain-based acceptance criteria. Therefore, [Nonmandatory Appendix EE](#) has been established to provide this information to the Code user.

FF-1110 GOAL OF STRAIN CRITERIA

The goal of the strain-based acceptance criteria is to establish plastic strain limits that are capable of maintaining the allowable leakage rate identified in the Design Specification, during and after energy-limited events. To achieve this goal, only a limited number of proven ductile materials are allowed for which the strain limits have been established with sufficient margins of safety. When calculating the resulting strains using inelastic analyses, the material properties are varied at locations for various components fabricated from different material heats to assure that the maximum energy is input into the containment boundary (see [EE-1220](#)). The strain-based acceptance criteria have been established to prevent through-wall crack formation, thereby maintaining the helium leakage rate specified in the Design Specification, down to 10^{-7} std cc/sec ([EE-1210](#)) or lower. These strain-based acceptance criteria are not allowed to be used in instances where the resulting deformations could result in a breach of the containment.

FF-1120 CRITERIA LIMITATIONS

The strain-based acceptance criteria specified in this Nonmandatory Appendix were developed for the evaluation of containments subjected to energy-limited dynamic events that have been identified in the Design Specification as having to satisfy Level D Service Limits.

A number of additional limitations are imposed that shall be satisfied prior to determining if these strain-based acceptance criteria are appropriate to use. These limitations are specified in the paragraphs below.

FF-1121 Limited Loadings

Only energy-limited loadings shall be evaluated using the strain-based acceptance criteria. Such loadings are limited to one-time events per location on the containment where the strain-based acceptance criteria are implemented. Cyclic and repeated incremental strain responses are not allowed. The loadings include accidental drops and impacts of nonsharp, blunt objects [e.g., 6 in. (15 cm) diameter post with rounded edges, aircraft engine shafts, etc.].

FF-1122 Limited Materials

The strain-based acceptance criteria shall only be applied to 304, 304L, 304/304L, 316, 316L, and 316/316L material. The permitted ASME material specifications (as restricted by Table 2A of Section II, Part D, Subpart 1) are listed in [Table FF-1122-1](#). Materials used shall have sufficient material properties determined (see [Nonmandatory Appendix EE](#)) in order to properly implement the strain-based acceptance criteria and these properties and their implementation in the analysis shall be justified in the final Design Report.

If certain products from these material specifications have been subjected to excessive cold working or lack a proper heat treatment and the result is a significant loss of ductility (true fracture strain approaching the true uniform strain limit), these products shall not have the strain-based acceptance criteria applied to them. [FF-1140](#) provides additional requirements that assure this concern is addressed.

FF-1123 Limited Temperatures

The applicable material temperature range shall be limited from -40°F to 800°F (-40°C to 425°C).

Table FF-1122-1
Permitted Material Specifications and
Products

Material Specification	Type/Grade	Product
SA-182	F304L, F304, F316L, F316	Forgings only
SA-240	304L, 304, 316L, 316	Plate only (sheet and strip excluded)
SA-312 (Excluding HCW pipe)	TP304L, TP304, TP316L, TP316	Seamless and welded pipe
SA-965	F304L, F304, F316L, F316	Forgings
SA-376	TP304, TP316	Seamless pipe
SA-479 (Excluding strain-hardened material)	304L, 304, 316L, 316	Bars and shapes

GENERAL NOTE: Single or dual-marked materials (see Section II, Part D, Mandatory Appendix 7) are acceptable.

FF-1124 Limited Welded Joints

The applicability of these strain-based acceptance criteria to welds is limited to full-penetration butt welded joints (Categories A and B) only. Autogenous seam welds on SA-312 welded pipe shall be considered the same as the base material, able to implement the strain-based acceptance criteria. Other categories of welds and their heat-affected zones shall not use the strain-based acceptance criteria but the base material adjacent to these other types of welded joints, and heat-affected zones may still use the strain-based acceptance criteria. Note that the uniform strain limit for the weld material and adjacent heat-affected zone may be different than that of the base material.

FF-1125 Limited Fabrication Strains

Any process may be used to hot or cold form or bend containment boundary material including weld material provided that the following requirements are met:

(a) Fabrication-induced strains less than or equal to 5% do not need to be addressed in the strain-based acceptance criteria, nor are additional heat treatments required to reduce these strains. The fabrication strain (in percentage) shall be established as follows:

(1) for cylinders,

$$\% \text{ strain} = (50t/R_f) [1 - (R_f/R_o)]$$

(2) for spherical or dished surfaces,

$$\% \text{ strain} = (75t/R_f) [1 - (R_f/R_o)]$$

where

R_f = final radius to centerline of the curved surface

R_o = original radius (equal to infinity for a flat surface)
 t = nominal thickness

(3) for other shapes, fabrication-induced strains shall be established by appropriate means (e.g., measurement or analytical methods) and documented in the final Design Report

(b) The effects of fabrication strains that exceed 5% shall be minimized by means of post-fabrication heat treatment (per requirements specified for all grades in Section II, Part A, Specification SA-312, Table 2, "Annealing Requirements," or an equivalent heat treatment that reduces fabrication strain levels below 5%) or the fabrication strain⁴³ shall be deducted from the material's true uniform and fracture strain limit values. Fabrication strains that exceed 10% are not allowed.

(c) Residual stresses and the associated material strains in, or adjacent to, welds resulting from the welding process alone are not to be considered in the determination of fabrication strains.

FF-1126 Exclusions

The strain-based acceptance criteria shall not be applied to the following locations:

(a) regions of the containment where strain deformations are detrimental to maintaining the desired leakage rate (e.g., the sealing region of a bolted closure)

(b) structural or nonstructural attachments to the containment

(c) containment boundary fillet or partial penetration welds and their heat-affected zones, including such welds of attachments to the containment boundary

(d) threaded connections to the containment, even if seal welded

FF-1130 ACCURATE STRAIN DETERMINATION

The strain-based acceptance criteria shall be implemented using strains calculated from Quality Models. A Quality Model is a finite element model of the complete containment model that adheres to the guidance set forth in [EE-1240](#). Alternately, a model with suitable convergence and sensitivity studies completed that demonstrate the accuracy capability of that containment model may also be used. The explicit dynamics solution technique shall be employed for the analyses when using these acceptable finite element models.

FF-1140 STRAIN-BASED ACCEPTANCE CRITERIA

The highest calculated product of the equivalent plastic strain and associated triaxiality factor (from each unique combination of location and material heat) determined from the inelastic analyses using the material property approaches described in [Nonmandatory Appendix EE](#) shall be evaluated to satisfy the strain-based acceptance criteria identified below. Appropriate material data ($\epsilon_{\text{uniform}}$ and $\epsilon_{\text{fracture}}$) for base material and, if applicable, weld material (including heat affected zone interfaces) are needed to

implement the strain-based acceptance criteria. If weld repairs per WB/WC-2500 exist in the containment material and strain-based acceptance criteria are used, appropriate material properties of the repair weld shall be reflected in the inelastic analyses. However, if the depth of the repair cavity does not exceed the lesser of $\frac{3}{8}$ in. (10 mm) or 10% of the section thickness, the inelastic analysis implementing the strain-based acceptance criteria can treat these locations as the adjacent base material.

All containment materials implementing the strain-based acceptance criteria must have sufficient ductility. Sufficient ductility of the base material shall be demonstrated by satisfying either of the following requirements:

(a) the true fracture strain at room temperature shall be at least two times the elongation value specified on the Certified Material Test Report (CMTR),⁴⁴ or

(b) when temperature-dependent test data are available, the true fracture strain limit shall be at least two times the true uniform strain limit at all temperatures under consideration

In order to assure proper containment material responses throughout the specified design life, material properties at the beginning of the design life as well as at the end of the design life (aged condition) shall be evaluated when implementing the strain-based acceptance criteria. Quantification of any potential material degradation shall be justified in the Design Specification.

The required strain criteria evaluations are described in the paragraphs below. Strain parameters (ϵ_{eq}^p , $\epsilon_{uniform}$, and $\epsilon_{fracture}$) are defined in [Nonmandatory Appendix EE](#) and the Triaxiality Factor (TF) is defined in [FF-1143](#).

FF-1141 Criteria for Locations Away From a Gross or Local Structural Discontinuity

For material greater than $3t_n$ (where t_n is the adjacent nominal containment wall thickness) away from a gross or local structural discontinuity, the following shall be satisfied:

(a) The products of the equivalent plastic strain (ϵ_{eq}^p) and the associated TF value at each evaluation location through the section shall be calculated for each calculated time interval. The average of these products through the section, $[(TF)(\epsilon_{eq}^p)]_{avg}$, at any time shall be

$$[(TF)(\epsilon_{eq}^p)]_{avg} \leq (0.67 \epsilon_{uniform})$$

(b) The maximum product of the equivalent plastic strain (ϵ_{eq}^p) and the associated TF value, $[(TF)(\epsilon_{eq}^p)]_{max}$, at any time at any containment location⁴⁵ shall be

$$[(TF)(\epsilon_{eq}^p)]_{max} \leq [\epsilon_{uniform} + 0.25 (\epsilon_{fracture} - \epsilon_{uniform})]$$

FF-1142 Criteria for Locations at a Gross or Local Structural Discontinuity

At a gross or local structural discontinuity or within $3t_n$ of a gross or local structural discontinuity, the following shall be satisfied:

(a) The products of the equivalent plastic strain (ϵ_{eq}^p) and the associated TF value at each evaluation location through the section shall be calculated for each calculated time interval. The average of these products through the section, $[(TF)(\epsilon_{eq}^p)]_{avg}$, at any time shall be

$$[(TF)(\epsilon_{eq}^p)]_{avg} \leq (0.85 \epsilon_{uniform})$$

(b) The maximum product of the equivalent plastic strain (ϵ_{eq}^p) and the associated TF value, $[(TF)(\epsilon_{eq}^p)]_{max}$, at any time at any containment location⁴⁵ shall be

$$[(TF)(\epsilon_{eq}^p)]_{max} \leq [\epsilon_{uniform} + 0.25 (\epsilon_{fracture} - \epsilon_{uniform})]$$

FF-1143 Triaxiality Factor

The Triaxiality Factor (TF) is a time-dependent parameter and is defined as

$$TF = \frac{(\sigma_1 + \sigma_2 + \sigma_3)}{\sqrt{\frac{1}{2}[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]}}$$

where σ_1 , σ_2 , and σ_3 are the principal stresses at the location under evaluation. Only those TF values while the plastic straining is occurring need be considered.

The TF values to be used for the [FF-1141](#) and [FF-1142](#) evaluations shall be either

(a) the peak TF value at a location (time independent), where any TF value less than 1.0 is set to 1.0

(b) the instantaneous TF value at a location, where any TF value less than 1.0 is set to 1.0

FF-1144 Special Strain Limits

When the average (through the containment wall thickness) equivalent plastic strain (ϵ_{eq}^p)_{avg} is due to pure shear, the criteria in [FF-1141](#) and [FF-1142](#) shall be satisfied, but the Triaxiality Factor used shall equal 3.

FF-1145 Strain Rate Effects

In order to address strain rate effects ([EE-1250](#)), additional inelastic analyses shall be completed in order to assure that the maximum resulting product of the equivalent plastic strain and associated triaxiality factor have been determined. All inelastic analyses performed for the entire containment model shall be repeated two additional times,

increasing the true stress-strain curves by 20% (in the stress direction only) each successive time. As an alternative, experimentally determined strain rate data may be used in the structural evaluations for the final Design Report, provided the strain rate data address the proper range of strain rates experienced, and the use of the data is justified in the final Design Report.

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ENDNOTES

- 1 Throughout the remainder of this Article, wherever the words *component* or *components* are used, they shall be understood to include portions thereof, and also appurtenances and portions thereof.
- 2 [Tables XI-3221.1-1](#) and [XI-3221.1-2](#) give a list of many commonly used gasket materials and contact facings, with suggested values of m , b , and y that have proved satisfactory in actual service. These values are suggested only and are not mandatory. Values that are too low may result in leakage at the joint, without affecting the safety of the design. The primary proof that the values are adequate is the hydrostatic test.
- 3 The ratings in these standards are based on the hub dimensions given or on the minimum specified thickness of flanged fittings of integral construction. Flanges fabricated from rings may be used in place of the hub flanges in these standards provided that their strength, calculated by the rules in this Appendix, is not less than that calculated for the corresponding size of hub flange.
- 4 Where additional safety against abuse is desired, or where it is necessary that the flange be suitable to withstand the full available bolt load $A_b \times S_a$, the flange may be designed on the basis of this latter quantity.
- 5 When internal pressure occurs only during the required pressure test, the design may be based on external pressure and auxiliary devices such as clamps may be used during the application of the required test pressure.
- 6 The combined force of external pressure and bolt loading may plastically deform certain gaskets to result in loss of gasket contact pressure when the connection is depressurized. To maintain a tight joint when the unit is repressurized, consideration should be given to gasket and facing details, so that excessive deformation of the gasket will not occur. joints subject to pressure reversals, such as in heat exchanger floating heads, are in this type of service.
- 7 This definition of stress intensity is not related to the definition of stress intensity applied in the field of Fracture Mechanics.
- 8 Equivalent linear stress is defined as the linear stress distribution which has the same net bending moment as the actual stress distribution.
- 9 See [Table XIII-1130-1](#) and Note (4) of [Figure XIII-1141-1](#).
- 10 Subdivision of secondary stresses into membrane and bending components is not required because the same stress limits apply to both components.
- 11 The tests on which the design curves are based did not include tests at temperatures in the creep range or in the presence of unusually corrosive environments, either of which might accelerate fatigue failure. Therefore, these curves are not applicable at service temperatures for which creep is a significant factor. In addition, the designer shall evaluate separately any effects on fatigue life which might result from an unusually corrosive environment.
- 12 It is permissible to use $1.5S_m$ whenever it is greater than S_y .
- 13 Knowing the rated capacity of a pressure relief valve which is stamped on the valve, it is possible to determine the overall value of KA in either of the following equations in cases where the value of these individual terms is not known:
 (a) *Rated Steam Capacity*
 For pressures up to 1,500 psig (10.3 MPa)

$$KA = W_s / C_N P$$

For pressures over 1,500 psig (10.3 MPa) to 3200 psig (22 MPa)

$$KA = \frac{W_s}{F_N C_N P}$$

(b) *Rated Air Capacity*

$$KA = (W_a / CP) \sqrt{T / M}$$

This value for KA is then substituted in the above equation to determine the capacity of the safety valve in terms of the new gas or vapor.

- 14 Rated capacity (lb/hr air @ 60°F @ 14.7 psia)/0.0766/60 = rated capacity (scfm air) [Rated capacity (kg/h air @ 20°C @ 101 kPa_{abs})/1.204 = rated capacity (m³/h air)].
- 15 This conversion is not valid for liquid flashing valve operating conditions.
- 16 Knowing the rated capacity of pressure relief valve stamped with a liquid capacity, it is possible to determine the overall value of KA in the following equation where the value of the individual terms is not known:

$$KA = \frac{W_W \times 500}{2,407 \sqrt{(P - 14.7)(62.3)}}$$

- 17 Illustrative samples of statements are shown in Guide A.
- 18 Express metric values in exponential form.
- 19 Cozzone, F. P. Bending Strength in the Plastic Range. *Journal of Aeronautical Sciences*, May 1943.
- 20 Bruhn, E. F. *Analysis and Design of Flight Vehicle Structures*. Tri-State Offset Company, 1965, Chap. C3.
- 21 Gavalis, R. Bending Strength in the Plastic Range. *Machine Design*, July 1964.
- 22 Applicable operability requirements are contained in the Subarticles designated 200 in this Appendix, such as [B-2200](#), B-3200, etc.
- 23 Applicable regulatory requirements are contained in the Subarticles designated 300 in this Appendix, such as [B-2300](#), B-3300, etc.
- 24 Express metric values in exponential form.
- 25 [Tables E-1210-1](#) and [E-1210-2](#) give a list of many commonly used gasket materials and contact facings with suggested values of m , b , and y that have proved satisfactory in actual service. These values are suggested only and are not mandatory. Values that are too low may result in leakage at the joint, without affecting the safety of the design. The primary proof that the values are adequate is the hydrostatic test.
- 26 In addition to Section II, Part D, Subpart 1, Table U, S_u values are also available in Code Cases covering new or additional materials for components and their supports.
- 27 The *stress intensity factor* as used in fracture mechanics has no relation to and must not be confused with the *stress intensity* used in Section III, Division 1. Furthermore, stresses referred to in this Appendix are calculated normal tensile stresses not stress intensities in a defect-free stress model at the surface nearest the location of the assumed defect.
- 28 WRCB 175 (Welding Research Council Bulletin 175) "PVRC Recommendations on Toughness Requirements for Ferritic Materials" provides procedures in Paragraph 5c(2) for considering maximum postulated defects smaller than those described.
- 29 The coolant temperature is the reactor coolant inlet temperature.
- 30 The vessel metal temperature is the temperature at a distance one fourth of the vessel section thickness from the inside wetted surface in the vessel beltline region. RT_{NDT} is the highest adjusted reference temperature (for weld or base metal in the beltline region) at a distance one fourth of the vessel section thickness from the vessel wetted inner surface as determined by Regulatory Guide 1.99, Rev. 2.
- 31 $C_3 = C_4 = 0$ when $F_1' = 0$.
- 32 A Class FF flange bolted to a rigid foundation may be analyzed as a Group 1 assembly by substituting $2l$ for l in [eq. L-3242\(18\)](#).

- 33 Where the flanges are identical dimensionally and have the same elastic modulus E , but have different allowable stresses S_f , the assembly may be analyzed as a Group 1 assembly provided the calculated stresses are evaluated against the lower allowable stress.
- 34 See [L-3243\(a\)\(2\)](#).
- 35 The symbols for the various stresses in the case of a Group 3 assembly also carry the subscript I or II. For example, S_{H1} represents the longitudinal hub stress in Flange I of the Group 3 assembly.
- 36 An individual time history response may be considered to have dominant frequency if one-half or more of its total response can be identified with a single frequency.
- 37 "Technical Position on Piping Installation Tolerances," Welding Research Council Bulletin 316, July 1986.
- 38 "Guidelines for Piping System Reconciliation" (NCIG-05, Revision 1), Electric Power Research Institute, EPRI-5639, May 1988.
- 39 Whenever NC/ND is used for reference, the reference is NC for Class 2 and ND for Class 3.
- 40 Stops do not include snubbers (NF-3412.4).
- 41 Other equivalent formats of this equation using different nomenclature are acceptable.
- 42 For establishing material properties using CMTRs, the CMTR data must accurately reflect the material property data of the final product being used in the fabrication of the containment. If this is not the case, then the material testing procedure of [EE-1222](#) should be used to establish existing material property data.
- 43 Engineering and true strains at these low strain values (less than 10%) are essentially the same numerical value, and it is conservative to consider engineering strain as equivalent to true strain in this specific use.
- 44 This requires the material procurement effort for containment materials to require that reduction of area values be specified on the CMTRs along with the elongation and other typical tensile test data. It is recognized that the CMTR data are generated at room temperature conditions, which is acceptable for this effort of determining adequate material ductility. [FF-1140\(a\)](#) is valid only if the CMTR data accurately reflect the material properties of the final product being used in the fabrication of the containment.
- 45 Not applicable to points of numerical singularity in the finite element model as justified in the final Design Report.

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