**FLEXIBLE PIPING JOINTS
FOR
LARGE-SCALE BREEDER REACTOR
PRIMARY LOOP APPLICATIONS**

**Phase I
Technical Summary**

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CONTENTS

	Page
I. Introduction and Summary	11
A. Technical Scope of Phase I Studies	11
B. Summary of Technical Evaluations	12
II. Considerations for Flexible Joint Applications	17
A. Flexible Joint Types (Using Bellows)	17
1. Articulating Flexible Joint	19
2. Axial Motion Flexible Joint	19
3. Free Motion Flexible Joint	20
4. Universal Motion Flexible Joint	20
B. Screening of Flexible Joint Piping Configurations	21
1. Configuration	21
2. Number of Joints Between Components	21
3. Minimum Number of Bellows Between Components	21
4. Joint Type Employed	24
5. Joint Subtype(s) Employed	24
6. Motion Capability	24
7. Vertical Piping Run Requirements	24
8. Pipe Pressure Loading	25
9. Guard Vessel and Elevated Piping Requirements	25
10. Joint Accessibility	26
11. Pump and IHX Shielding	27
12. Joint Drainability	27
13. Viable Configurations	28
C. Operating Requirements and Their Effects on Joint Type - Configuration Selection	28
1. Large-Scale Breeder Reactor Design Conditions	30
2. Flexible Joint Design Requirements	31
3. Selection of Articulating Configurations	39
4. Use of Cold Leg Pump	39
D. Additional Design Considerations	40
1. Number of Bellows Plies	41
2. Redundant Designs	44

CONTENTS

	Page
3. Leak and Incipient Failure Detection	45
4. Flow Liners	45
5. Motion Limiters	46
6. Articulation Unit Lubricants	46
III. Background – Bellows Analytic and Design Techniques	49
A. Bellows Screening Rules and Failure Modes	50
B. Bellows Element Response Behavior and Analytic Prediction Capabilities	56
1. Static Stress Response and Prediction	56
2. Fatigue Response and Fatigue Life Prediction	78
3. Stiffness Prediction	82
4. Dynamic Loadings and Response Predictions	83
5. Instability Phenomena and Prediction	85
6. Pressure Drop Prediction	86
7. Interply Coupling and its Prediction in Multi-Ply Bellows	86
8. Effects of Added Reinforcement and Prediction Methods	90
IV. Flexible Joint Bellows Materials	97
A. General Considerations	97
B. Quantitative Material Comparisons	98
V. Bellows Fabrication Capabilities and Limitations	111
A. Bellows Fabrication Methods	111
1. Formed Bellows	111
2. Welded Disk Bellows	115
3. Other Welded Bellows	117
4. Machined Bellows	117
5. Deposited Bellows	118
6. Selected Large-Scale Breeder Reactor Bellows Element Fabrication Method	118
B. Current Vendor Capabilities	118
1. Vendor A	119
2. Vendor B	120

CONTENTS

	Page
3. Vendor C	121
4. Vendor D	122
5. Vendor E	123
6. Vendor F	124
7. Vendor G	125
VI. Bellows Design – Phase I Effort	127
A. Computer-Generated Scoping for Preconceptual Design	128
1. Bellows Element Design Equations	128
2. Bellows Design Loadings	135
3. Range of Geometries Investigated in Computer- Generated Scoping Studies	135
B. Reduced-Scope Screening for Preconceptual Design Selections	135
1. Preconceptual Screening Rules	137
2. Reduced Set of Design Geometries Based on Considerations of Time-Independent Strength Requirements	147
3. Fatigue and Elevated-Temperature Evaluations	160
C. Consideration of Double-Series Articulating Joint Bellows Elements	170
D. Feasibility Summary of Preconceptual Bellows Design	171
1. Pressure-Balanced Axial Motion Joints	171
2. Articulating Joints	173
VII. Preconceptual Flexible Joint Designs	175
A. Scope of Preconceptual Joint Design Effort	175
1. Bellows Design	175
2. Articulation Unit Design	176
3. Flow Liner Design	176
4. Redundancy - Leak Detection Implementation	176
5. Motion Limiter Design	177
B. Summary of Preconceptual Joint Designs	177
References	183

CONTENTS

Page

Appendices

A.	Selected Optimum Bellows Geometries Per Time-Independent Screening Rules 1a, 2a, 3, and 4	A-1
B.	Computational and Design Tables for Application of Time-Dependent and Fatigue Screening Rules 1b, c, d, 2b, c, d, 5, and 6	B-1
C.	Design and Computation Sheets for Alternate Flexible Joint Designs	C-1

TABLES

II-1.	Summary of Piping Configuration Features (see Figure II-1)	22
II-2.	Considerations for Configuration Disqualification	29
II-3.	Prototypic Large-Scale Breeder Reactor Design Parameters and Steady-State Operating Conditions	30
II-4.	Normal Operating Cycles – Hot Leg	31
II-5.	Normal Operating Hold Periods – Hot Leg	31
II-6.	Steady-State Joint Temperature and Pressure Requirements	32
II-7.	Cyclic Temperature and Pressure Loadings – 42-in. Joint	33
II-8.	Cyclic Temperature and Pressure Loadings – 56-in. Joint	33
II-9.	Sustained Loading Hold Periods – 42-in. Joint	34
II-10.	Sustained Loading Hold Periods – 56-in. Joint	34
II-11.	Effective Relative Horizontal and Vertical Component Displacements for Large-Scale Breeder Reactor Primary Piping Loop	36
II-12.	Design Motion Requirements for Flexible Joints	38
IV-1.	Comparison of Candidate Flexible Piping Joint Bellows Element Materials	99
VI-1.	Range of Geometries Considered in Initial Design Efforts	136
VI-2.	Selected Preliminary Preconceptual Unreinforced Single Bellows Geometries	157

TABLES

	Page
VI-3. Material Yield Point Data	158
VI-4. Selected Geometry Evaluation for Time-Independent Screening Rules (1)-(4) Using the Four Phase I Materials	159
VI-5. Usage Fractions for Time-Dependent, Load-Controlled Stresses in Selected Type 316 SS Bellows	162
VI-6. Comparison of Flexible Joint Material Creep Rupture Strengths	163
VI-7. Comparison of Time-Dependent Rupture Characteristics	163
VI-8. No-Hold Time Fatigue Damage Fractions for Selected Type 316 Stainless Steel Bellows	166
VI-9. Hold-Time Fatigue Damage Fractions for Selected Type 316 Stainless Steel Bellows	167
VI-10. Comparison of Allowable Strain Ranges at 1200°F and 12,000 Cycles (No Hold Time)	169
VI-11. Summary of Double-Series Articulating Joint Bellows Evaluation	172
VII-1. Summary of Preconceptual Joint Designs	178
A-1. Reduced Set of Optimum Geometries Per Screening Rules 1a, 2a, 3 and 4	A-2
A-2. Geometries Selected from Table A-1 Which: (1) Minimize $S_{O,r}$; (2) Represent Commonly Manufactured Shapes; and (3) Represent Thickness Effects On Convolution Height and $S_{O,r}$	A-11
B-1. Calculation of Primary Stress Usage Factors	B-3
B-2. Hold Period Definitions	B-4
B-3. Hold Period Stress Scale Factors	B-4
B-4. Significant Design Loading Cycles	B-5
B-5. No Hold-Time Fatigue Damage Fraction Calculations	B-6
B-6. Hold Time Fatigue Damage Fraction Calculations	B-7
B-7. Creep Damage Fraction Calculations	B-9
C-1. Double Series Articulating Bellows Scale Factors for Fatigue Evaluation	C-2
C-2. No Hold-Time Fatigue Damage Fraction Calculations for 42-in. Double-Series Articulating Bellows Designs BB and CC	C-2

TABLES

		Page
C-3.	Hold-Time Fatigue Damage Fraction Calculations for 42-in. Double Series Articulating Bellows Designs BB and CC.	C-3
C-4.	Stress and Stability Calculations for 42-in. Six-Ply Double-Series Articulating Bellows RR With Reinforcing Rings (See Subsection III-B for Nomenclature)	C-4
C-5.	Yield Point and Time-Dependent Screening Rules Applied to 42-in. Six-Ply Double-Series Articulating Bellows RR With Reinforcing Rings	C-6

FIGURES

I-1.	42-in. Gimbaleed Flexible Joint	15
I-2.	42-in. Hinged Flexible Joint	16
II-1.	Flexible Joint Types	18
II-2.	Piping Configurations for Flexible Joints	23
II-3.	Large-Scale Breeder Reactor Primary Loop Arrangement — Flexible Joint Incorporation	37
III-1.	Typical Flow-Induced Bellows Fatigue Failure	51
III-2.	Typical Motion-Induced Bellows Fatigue Failure	52
III-3.	Outer-Ply Overpressure Failure in Multi-Ply Bellows (Gas Service).	53
III-4.	Typical Squirmed Bellows	55
III-5.	Typical Bellows Element Motions	59
III-6.	Pressure Effects on Angulating Bellows Element.	61
III-7.	Salzman's Square Wave Bellows	63
III-8.	Salzman's Force and Deflection Compatibility Assumptions	64
III-9.	Salzman's "Curved Ended" Bellows	65
III-10.	Turner and Ford's Toroidal Bellows	73
III-11.	C_p for Convoluteed Bellows	94
III-12.	C_d for Convoluteed Bellows	95
III-13.	C_f for Convoluteed Bellows	96

FIGURES

	Page
IV-1. Plot of Larson-Miller Parameter P vs Stress for 1% Creep Strain, Comparing Alloys Inconel 718, Hastelloy X, Hastelloy C and Type 316 SS	100
IV-2. Plot of Larson-Miller Parameter P vs Stress for 0.5% Creep Strain, Comparing Alloys Inconel 718, Hastelloy X, Hastelloy C, and Type 316 SS	100
IV-3. Plot of Larson-Miller Parameter P vs Stress for Rupture Comparing Alloys Inconel 718, Hastelloy X, Hastelloy C, and Type 316 SS	101
IV-4. Maximum Allowable Stress vs Temperature Approved by B&PVC, Section VIII, App. Q, QA-600	102
IV-5. Design Fatigue Allowables at Zero Mean Stress ⁽⁴⁰⁾	103
IV-6. Design Fatigue Allowables With Mean Stress Correction ⁽⁴⁰⁾	105
IV-7. Comparison of Estimated No Hold Time Fatigue Curves of Inconel 718, Hastelloy X and Type 316 SS (zero Mean Stress)	106
IV-8. Comparison of Estimated No Hold Time Fatigue Curves with the Maximum Correction for Mean Stress (Inconel 718, Hastelloy X, and Type 316 SS)	107
IV-9. Type 321 Stainless Steel Bellows Fatigue Curves ⁽⁴¹⁾	108
IV-10. Nickel Base Alloy Bellows Curves ⁽⁴¹⁾	109
V-1. Rubber Bulging Method of Forming Convolutions	113
V-2. Mechanical Method of Forming Convolutions	113
V-3. Rolling Method of Forming Convolutions	114
V-4. Hydraulic Method of Forming Convolutions	116
VI-1. Definition of Stress Equation Nomenclature	129
VI-2. Deflection Due to Internal Pressure of Angulating Bellows ⁽⁴²⁾	130
VI-3. Bending Stress Shape Factor ⁽²⁾	131
VI-4. Spring Rate Shape Factor ⁽²⁾	132
VI-5. Behavior Curves for 42-in. Angulating Bellows from Selection Bases 1, 4a, 4b	151
VI-6. Behavior Curves 42-in. Angulating Bellows from Section Basis 3	152
VI-7. Behavior Curves for 56-in. Angulating Bellows from Selection Bases 1, 4a, 4b	153

FIGURES

	Page
VI-8. Behavior Curves for 56-in. Angulating Bellows from Selection Basis 3	154
VI-9. S_t - Type 316 SS	161
VI-10. Type 316 Stainless Steel Bellows "Fatigue Curve"	165
VI-11. Stress to Rupture Curve, Type 316 Stainless Steel	168
VII-1. 56-in. Hinged Joint	179
VII-2. 56-in. Gimbale Joint	180
VII-3. 42-in. Hinged Joint	181
VII-4. 42-in. Gimbale Joint	182

I. INTRODUCTION AND SUMMARY

Atomics International has conducted an evaluation of flexible piping joints for large-scale breeder reactor primary loop applications. The activities are being performed for the Energy Research and Development Administration within Atomics International's LMFBR Technology Program, Contract AT(04-3)824. This document is a summary of the technical studies performed during Phase I of the development program (April, May, and June, GFY 1975).

A. TECHNICAL SCOPE OF PHASE I STUDIES

Reactor manufacturer's studies have shown that there are significant economic incentives for employment of flexible piping joints (bellows) in large-scale breeder reactor primary loops. Incorporation of flexible joints in Class 1 piping systems will require development of analytic, manufacturing, installation, testing, and quality control techniques, and in addition will require ASME Boiler and Pressure Vessel Code approval of such incorporation. In order to identify the required scope of the technical development efforts, Atomics International has evaluated state-of-the-art capabilities and probable joint design configurations during the Phase I portion of the flexible piping joint program. Toward these ends, several major technical areas were defined for the Phase I studies:

- Selection of the preferred joint type and primary piping configuration(s)
- Identification of current bellows analytic capabilities
- Investigation of candidate bellows materials
- Evaluation of current bellows fabrication methods
- Initiation of the preconceptual bellows design efforts
- Initiation of the preconceptual joint design effort.

Each of these topics is presented in detail in Sections II through VII of this document. The following subsection (I-B) summarizes the important findings of the Phase I studies.

B. SUMMARY OF TECHNICAL EVALUATIONS

The initial investigations were performed to define the preferred flexible joint type(s) and preferred primary piping system configuration(s). Four general types of joints were evaluated: articulating motion, axial motion, universal motion (with spool piece), and free motion. These joints were employed in 14 different piping system configurations. Based on considerations of motion capabilities, numbers of bellows, joint reaction thrusts, guard vessel design, joint accessibility, and reactor shielding, the articulating joints were selected as the preferred type. These joints can employ either hinge or gimbal assemblies, depending on the detailed out-of-plane motion requirements of the piping system. In addition, the pressure-balanced axial motion joint was selected as a possible alternate to the articulating joint (for further evaluation in Phase I), although it is a larger, more complex unit. The normal operating temperatures, pressures, motions, and cyclic loadings for these selected joints were then defined based on a conservative prototypical large-scale breeder reactor design which used a hot leg primary pump.

In parallel with the joint - configuration selection, a literature search for bellows analysis and design techniques was conducted. Based partly on considerations of various bellows failure modes and the mechanisms which control these modes, and partly on functional requirements, the following response prediction capabilities were specifically addressed: static stresses, fatigue life, stiffness, dynamic response, instability pressure, internal pressure drop, interply coupling (in multi-ply bellows), and root reinforcement effects. Both state-of-the-art and recommended techniques were examined to provide a broad analytic base from which to initiate the technical development activities.

Four candidate bellows materials were evaluated during the Phase I studies: Type 316 stainless steel, Inconel 718, Hastelloy C, and Hastelloy X. These range from a currently ASME-approved material to unapproved higher-strength materials, and encompass the range of strengths which would be required for various bellows designs. The materials were compared qualitatively and quantitatively (where possible), leading to the assessment that Inconel 718 probably represents the preferred material. Future ASME codability considerations and

the effect of mean stress on fatigue properties, and long term stability may, however, result in preference being switched to an alternate material, possibly Hastelloy C. No preferred material was selected during Phase I. Available and estimated data for the candidate materials was prepared, so that each could be further evaluated in the bellows design efforts.

Current bellows fabrication literature and vendors were surveyed during Phase I to assist in selecting the preferred manufacturing process and to determine the limitations on this process. The formed, welded, machined, and deposited processes were compared, resulting in the choice of forming for the size and geometry range needed for large-scale breeder reactors. Based on personal visits to bellows manufacturers, information on fabrication capabilities and possible future extension of these capabilities were used to define fabricability of different bellows geometry ranges. Varying levels of fabricability were represented by imposing geometric constraints on the preconceptual geometries.

The operating requirements, design techniques, materials data, and fabrication limitations were then used to scope and screen an initial computer-generated matrix of over 33,000 articulating and axial-motion bellows geometries. Screening rules were developed from consideration of time-independent and time-dependent failure modes and their mechanisms. Based on the time-independent rules, with no geometric fabricability restrictions, the original matrix was reduced to 373 designs. These represented the "optimum" geometries for each bellows length, number of plies, and ply thickness. From this reduced set, 56 preconceptual geometries were selected which best represented: (1) no imposed geometric limits; (2) imposed geometric limits based on current fabrication capabilities; (3) definition of the optimum shape for ease of fabrication; and (4) consideration of the range of geometric parameters. Due to proportionately higher stresses and instability in comparison to articulating joints, axial joints were eliminated from further considerations at this time. Eleven of the geometries were selected for application at the time-dependent screening rules.

These final evaluations indicated that both Type 316 stainless steel and the higher strength alloys have good potential capability to satisfy the normal operating and design requirements for the high-temperature, low-pressure flexible joints in the pump suction line, and the low-temperature, high-pressure joints in the reactor inlet line. However, stainless steel was shown to have a low potential capability for satisfying the requirements of the high-temperature, high-pressure pump discharge line joints, indicating that the higher-strength alloys would probably be required in this piping location. It is important to note that the high-temperature, high-pressure piping environment does not exist in a primary loop which employs a cold leg pump. This implies that early large scale breeder reactors which use cold leg pumps could possibly incorporate stainless steel flexible joints in all primary loop lines. This is an important fact when one considers the long times which may be required to obtain Code approval of a higher-strength material for Class 1 elevated-temperature service.

In order to further evaluate the feasibility of using Type 316 stainless steel in the pump discharge line, three additional geometries were investigated using double-series-bellows articulating joints. Such joint designs use two bellows, and thus reduce the motion requirement per bellows to less than one-half of that for the geometries previously evaluated. In addition to this motion reduction, one of the bellows incorporated six plies with root reinforcing rings. Only the reinforced bellows passed all the screening rules; the other two bellows could not pass screening rules for which uncertainties exist concerning the categorization of stresses. With certain categorizations, these bellows could also pass all the screening rules. However, all these joints present fabrication difficulties which may possibly prevent their use.

Based on the bellows size ranges determined from the previously discussed studies, four preconceptual designs were generated. Two of these were 56-in. joints for the pump suction line, and the remaining two were 42-in. joints for the pump discharge and reactor inlet lines. Figures I-1 and I-2 are artists interpretations of a 42-in. gimbaled design and a 42-in. hinged design, respectively. During formulation of these designs, consideration was given to various features including: hinge versus gimbal, bellows redundancy, leak-incipient failure detection, flow liners, motion limiters, and bearing lubricants. The four preconceptual designs include arbitrary combinations of these feature variations.

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Figure I-1. 42-in. Gimbaled Flexible Joint

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Figure I-2. 42-in. Hinged Flexible Joint

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II. CONSIDERATIONS FOR FLEXIBLE JOINT APPLICATIONS

The incorporation of flexible piping joints in the primary heat transport system of a large-scale breeder reactor requires careful consideration of several interacting design, maintenance, and safety criteria. The following subsections discuss the applicable types of flexible joints and the available piping configurations for these joints. Preferred type - configuration combinations are selected, and their requirements are presented. Additional design considerations, such as bellows redundancy and leak-detection, are discussed in the last subsection.

A. FLEXIBLE JOINT TYPES (Using Bellows)

Four basic types of flexible piping joints were considered in the Phase I studies: articulating, axial motion, free motion, and universal motion. Several joint subtypes exist for each of these. The following paragraphs describe the key features of these units. Figure II-1 illustrates several of the types and subtypes of joints.

These four types of joints encompass the most widely used designs for piping systems. Articulating (hinged) flexible joints have accumulated several years of high-temperature sodium experience in the 10-in. loops of the Sodium Components Test Installation at the Liquid Metal Engineering Center. Axial motion bellows have seen much usage in elevated temperature sodium systems - the 44-in. expansion compensating bellows of the Hallam Nuclear Power Facility intermediate heat exchanger and the Space Nuclear Auxiliary Power system expansion compensators are only two examples of this usage. In the aerospace industry, articulating (hinged and gimbaled), axial motion, free motion, and universal motion flexible joints have been extensively used. This is especially true in rocket propulsion, where small size and weight dictate the use of flexing elements in minimum length piping systems. In contrast, commercial applications of flexible joints are also widespread. For example, the petrochemical industry uses articulating and axial motion joints in both processing plants and transport pipelines.

EXTERNAL HINGE
SINGLE BELLOWS

EXTERNAL HINGE
DOUBLE-SERIES
BELLOWS

EXTERNAL GIMBAL
SINGLE BELLOWS
CONCENTRIC REDUNDANT
DESIGN

INTERNAL GIMBAL
SINGLE BELLOWS

UNIVERSAL JOINT

FREE JOINT

PRESSURE-BALANCED
AXIAL MOTION JOINT

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Figure II-1. Flexible Joint Types

1. Articulating Flexible Joint

This type of joint is used to accommodate angular motion (rotation about an axis). The motion is controlled by a hinge or gimbal assembly (defined as the articulation unit), which can be mounted internal or external to the flexing element (bellows). Either single or double-series flexing element designs can be employed. There are thus eight basic subtypes of articulating flexible joints (2 articulation units x 2 internal-external x 2 single-double = 8).

Hinged articulation units are used when the angular motion can be restricted to a single plane. The hinge assembly absorbs the majority of the axial pressure thrust, and thus articulating joints can be used without external piping system reaction anchors. Gimballed articulation units are incorporated when out-of-plane piping motions are significant. They consist of a pair of hinges, 90° apart, and provide angular motion capability about any axis perpendicular to the flexing element longitudinal centerline. The gimbal unit (hinges plus ring) absorbs the axial pressure thrust of the joint, as for the hinged unit.

Location of the articulation unit inside the flexing element provides the most compact joint design (see Figure II-1); however, pressure losses and transient thermal stresses for such units can be excessive. The external design requires additional space, but has no flow restrictions and is inspectable and available for maintenance.

The most common articulating joints use a single bellows designed to absorb the total angular motion requirement. However, for bellows designs which are relatively long (to allow higher motions per unit motion stress), it is difficult to guarantee stability against squirm. In such cases, the double-series articulating joint can be used (see Figure II-1). This joint is equipped with a slightly more complex articulation unit, which equally divides the angular motion between two single bellows joined end-to-end (with a short straight section between them). Each bellows is thus required to absorb only one-half of the total joint angulation.

2. Axial Motion Flexible Joint

This type of joint is used to accommodate purely axial displacements. The motion is controlled by a telescoping assembly, which additionally helps to

prevent pressure instability (squirm). Two subtypes of axial motion joints are common: unbalanced and pressure-balanced units.

The unbalanced joint uses a single flexing element (bellows) which absorbs the total motion requirements of the joint. The outward thrust created by the internal pressure acting on the cross-sectional area of the bellows is in the same direction as the displacement of the joint. Because they are intentionally designed "soft" in the longitudinal direction, axial motion joints require external anchors to absorb the piping reaction loads and prevent excessive deformation of the bellows. This unfavorable characteristic can be eliminated by incorporating three interconnected bellows which create pockets of opposing pressure forces, resulting in internal balancing of axial thrusts. In addition to requiring three, rather than one, bellows per joint, the pressure-balanced units have the following disadvantages: (1) the outer balancing bellows must have a diameter $\sim\sqrt{2}$ times the primary bellows diameter; (2) the overall assembly will be approximately twice as long due to the addition of a balancing bellows element in series with the primary bellows; (3) the force transfer linkages will be massive, and may present thermal transient problems; and (4) the fabrication procedure may not allow complete examination of all bellows welds.

3. Free Motion Flexible Joint

This joint utilizes no linkages, and allows motions that are a combination of axial deflection, lateral offset, and angular rotation. These units have poor pressure stability (against squirm), and usually require load reaction anchors to absorb the pressure thrusts.

4. Universal Motion Flexible Joint

This joint is used to absorb any combination of axial, lateral, and angular motions, in any plane. It consists of two flexing elements (bellows) joined by a relatively long central connector (spool piece), see Figure II-1. No linkages are used to control the joint motion. It can absorb more lateral motion than a free joint equal in length to the two universal joint bellows. However, the overall length of the joint can be several times greater than for other joint types. Universal joints are unable to absorb internally generated pressure thrusts, and usually require piping system anchors to control reaction loads.

B. SCREENING OF FLEXIBLE JOINT PIPING CONFIGURATIONS

Several piping system configuration options are available for each type of flexible joint. Each must be considered from several viewpoints: the overall motion capabilities of the configuration must be evaluated (i.e., does the arrangement allow for both relative vertical and horizontal motion between components?); the interaction with the guard vessels must be assessed; the accessibility of each of the joints must be investigated; the interference with heat transport system component shielding must be determined; etc. Figure II-2 schematically illustrates the available piping configurations using the joint types presented in Subsection II-A. Table II-1 presents a capsule summary of the features of these configurations. The advantages and disadvantages of these systems are discussed in the following paragraphs, based on the features presented in Table II-1. The discussion refers to the table column structure, reading from left to right.

1. Configuration

This alphanumeric symbol refers to the piping configurations presented in Figure II-2. A lower case letter is used to indicate joint subtype, if required.

2. Number of Joints Between Components

The total number of flexible joints which are required between primary loop components is dependent on the piping configuration, and ranges from one to five for the arrangements studied. From both economic and loop size considerations, this number should be minimized. With the exception of F3, the configurations of Figure II-2 use reasonable numbers of joints. However, this factor is not sufficient for eliminating F3 from further consideration.

3. Minimum Number of Bellows Between Components

This number is based on using nonredundant joint designs (see Subsection II-D). This is a reasonable assumption because any redundancy requirements would double the number of bellows for most configurations. Articulating joints are assumed to be single bellows units, and pressure-balanced axial motion joints use three bellows per unit (see Figure II-2). The number of bellows in the configurations of Table II-1 range from one to six. These are reasonable numbers, although use of five or six bellows represents a more complex joint

TABLE II-1.
SUMMARY OF PIPING CONFIGURATION FEATURES
(see Figure II-1)

Config- uration	Number of Joints Between Components	Minimum Number of Bellows Between Components	Joint Type (s) Employed	Joint Subtype (s) Employed	Motion Capability	Vertical Piping Run Requirements	Pipe Pressure Loading	Guard Vessel and Elevated Piping Requirements	Joint Accessibility	Pump and IHX Shielding	Viable Configuration
AR1h	2	2	Articulating	Hinged	Horizontal only	Yes, long	No reaction thrust	Elevated piping and isolated guard vessels	One in guard vessel	Labyrinth	No
AR1g	2	2	Articulating	Gimbaled	Horizontal only	Yes, long	No reaction thrust	Elevated piping and isolated guard vessels	One in guard vessel	Labyrinth	No
AR2h	3	3	Articulating	Hinged	Horizontal and vertical	Yes, long	No reaction thrust	Elevated piping and isolated guard vessels	One in guard vessel	Labyrinth	No
AR2g	3	3	Articulating	Gimbaled	Horizontal and vertical	Yes, long	No reaction thrust	Elevated piping and isolated guard vessels	One in guard vessel	Labyrinth	No
AR3h	3	3	Articulating	Hinged	Horizontal and vertical	Yes, short	No reaction thrust	Elevated piping and isolated guard vessels	All above guard vessels	Labyrinth	Yes
AR3g	3	3	Articulating	Gimbaled	Horizontal and vertical	Yes, short	No reaction thrust	Elevated piping and isolated guard vessels	All above guard vessels	Labyrinth	Yes
AX1u	1	1	Axial	Unbalanced	Horizontal only	No	Thrust reaction at nozzle	No elevated piping and coupled guard vessels	Only joint in guard vessels	None	No
AX1b	1	3	Axial	Pressure-Balanced	Horizontal only	No	No thrust reaction	No elevated piping and coupled guard vessels	Only joint in guard vessels	None	No
AX2u	1	1	Axial	Unbalanced	Horizontal only	Yes, short	Thrust reaction causes bending moments	Elevated piping and isolated guard vessels	Only joint above guard vessel	Labyrinth	No
AX2b	1	3	Axial	Pressure-Balanced	Horizontal only	Yes, short	No thrust reaction	Elevated piping and isolated guard vessels	Only joint above guard vessel	Labyrinth	No
AX3u	2	2	Axial	Unbalanced	Horizontal and vertical	Yes, short	Thrust reaction causes bending moments	Elevated piping and isolated guard vessels	All joints above guard vessels	Labyrinth	No
AX3b	2	6	Axial	Pressure-Balanced	Horizontal and vertical	Yes, short	No thrust reaction	Elevated piping and isolated guard vessels	All joints above guard vessels	Labyrinth	Yes
F1	1	1	Free	—	Horizontal and vertical	No	Thrust reaction at nozzle	No elevated piping and coupled guard vessels	Only joint in guard vessels	None	No
F2	1	1	Free	—	Horizontal and vertical	Yes, short	Thrust reaction causes bending moments	Elevated piping and isolated guard vessels	Only joint above guard vessel	Labyrinth	No
F3	5	5	Free	—	Horizontal and vertical	Yes, short	Thrust reactions taken by piping anchors	Elevated piping but anchors penetrate guard vessels	Two joints in guard vessels	Labyrinth	No
U1	1	2	Universal	—	Horizontal and vertical	No	Thrust reaction at nozzles	No elevated piping and coupled guard vessels	Only joint in guard vessel	None	No
U2	1	2	Universal	—	Horizontal and vertical	Yes, short	Thrust reaction causes bending moments	Elevated piping and isolated guard vessels	Only joint above guard vessel	Labyrinth	No
C1hb	3	5	Articulating and Axial	Hinged and Pressure-Balanced	Horizontal and vertical	No	No reaction thrust	No elevated piping and coupled guard vessels	All joints in guard vessel	None	No
C2hb	3	5	Articulating and Axial	Hinged and Pressure-Balanced	Horizontal and vertical	Yes, short	No reaction thrust	Elevated piping and isolated guard vessels	All joints above guard vessels	Labyrinth	No
C3hb	3	5	Articulating and Axial	Hinged and Pressure-Balanced	Horizontal and vertical	Yes, long	No reaction thrust	Elevated piping and isolated guard vessels	One joint in guard vessel	Labyrinth	No

*Assuming no concentric redundant designs, single bellows used in articulating joints, three bellows per pressure-balanced axial motion joint

9112-28-

Figure II-2. Piping Configurations for Flexible Joints

and configuration design. These considerations do not warrant disqualification of any of the remaining piping configurations.

4. Joint Type Employed

The joint types are defined in Subsection II-A. Those configurations using more than one type of joint are at an economic penalty because they require two separate designs with differing fabrication requirements. These economic penalties are sufficient to eliminate the "combination" configurations (C1hb, C2hb, and C3hb), because these arrangements offer no advantages over other available configurations. C1hb, C2hb, and C3hb are eliminated from further consideration.

5. Joint Subtype(s) Employed

The joint subtypes are discussed in Subsection II-A.

6. Motion Capability

Certain of the flexible joint configurations can accommodate both relative horizontal motion and relative vertical motion between the component nozzles, while other configurations can accommodate only relative horizontal motion. The reactor, pump, and IHX are supported from different elevations and have different vertical thermal profiles. This results in unequal vertical thermal expansion profiles for these components. It is unrealistic to assume that primary heat transport systems could be designed to eliminate these vertical motion differences for both steady-state and transient conditions. Flexible joint configurations should be capable of accommodating both horizontal and vertical motions. Thus, those configurations (AR1, AX1, and AX2 in Figure II-2) which do not satisfy this criterion are here eliminated from further consideration.

7. Vertical Piping Run Requirements

Vertical runs are required in many of the piping configurations. As seen from Figure II-2, these runs can contain zero joints (AX2, F2, U2), one joint (AR3, AX3, C2), or two joints (AR2, C3). Runs containing two joints are longer than those containing one. Runs containing zero joints are shorter than those with one. Excessively long vertical runs will impact the containment building design, because the upper horizontal sections of the component-to-

component piping will be higher than the containment floor level. Primary loop shielding requirements would necessitate the construction of "doghouses" to contain these elevated runs. Such structures are not economic, and may interfere with above-floor design features. Where possible, configurations should be selected which do not have excessively long vertical piping runs. For articulating joints, configuration AX3 is thus preferred over AX2. Although they were previously eliminated, configuration C2 would be preferred over C3.

8. Pipe Pressure Loading

As discussed in Subsection II-A, the unbalanced axial, free, and universal motion joint designs usually require piping system anchors to control internal pressure-generated thrusts. In comparison, the articulating and pressure-balanced axial joints are designed to transmit no thrusts into the piping system, and thus do not require piping reaction anchors. However, inspection of the piping configurations of Figure II-2 and Table II-1 will show that only F3 is amenable to anchoring, while AX2u, AX3u, F2, and U2 also would require anchoring. Any attempts to control reaction forces in these latter arrangements would restrict the motion of one or more legs of the configuration, generating excessive stresses in the restricted section(s) during thermal motions. Configurations AX2u, AX3u, F2, and U2 are thus eliminated from further consideration.

9. Guard Vessel and Elevated Piping Requirements

Safety considerations for large-scale breeder reactors require the use of guard vessels surrounding each primary heat transport system component. Current design practices use elevated piping runs between components, with these runs passing up and over the guard vessels. The void volume between each component and its guard vessel is minimized (within inspectability restrictions) because the largest such void volume in the primary system determines the operating reactor sodium elevation above the minimum reactor sodium level (i.e., a leak into a guard vessel can only drain the reactor to the minimum safe sodium level). Guard vessels are isolated from one another so that only a single void volume is filled if a component leakage into the guard vessel should occur. In addition, a leak in the elevated piping runs between guard vessels cannot drain the reactor below the minimum safe sodium level. The sodium

level in this piping is above the minimum safe reactor level, and a leak in any pipe can only drain the reactor until the siphon is broken at the top of the reactor outlet nozzles (which are near the minimum safe reactor sodium level).

Because of these safety considerations, it is extremely important to evaluate each flexible piping joint configuration for its impact on the guard vessel and elevated piping requirements. From Figure II-2, it can be seen that configurations AX1, F1, U1, and C1 have piping runs which are not elevated and are totally surrounded by a guard pipe. This guard pipe interconnects the component guard vessels, and because this would couple all guard vessel void volumes together, a guard barrier internal to the guard pipe must be installed. This barrier must necessarily decouple the guard vessels by providing a positive seal between the sodium piping and guard piping. This introduces one additional design problem: the thermal expansion of the guard vessels and interconnecting guard pipes must be accommodated. This could be done by floating the IHX and pump guard vessels or by installing bellows in the guard pipe on either side of the guard barrier. Either of these schemes introduces additional design complexity into the guard vessel system. It is Atomics International's opinion that such complexity is unnecessary, because alternate flexible piping joint configurations are available which can meet the basic guard vessel and elevated piping requirements while retaining the original simplicity of guard vessel design. Configurations AX1, F1, U1, and C1 are thus eliminated from further consideration.

Configuration F3 impacts the guard vessel design in a different way. This piping arrangement requires thrust reaction anchors at each elbow, and two of these anchors must penetrate guard vessels (see Figure II-2). This, in turn, would anchor the guard vessel and would require special design considerations to accommodate thermal expansion of the guard vessel. This additional complexity is unwarranted because alternate designs (using fewer joints) are available. Configuration F3 is thus eliminated from further consideration.

10. Joint Accessibility

Flexible piping joints may require visual monitoring and should be accessible for inservice inspection, possible periodic lubricant replenishment (of the motion assembly), and ease in removal replacement (if required). Joints

located within a guard vessel are not as accessible as those located above a guard vessel (in the elevated piping). Lack of joint accessibility is not a sufficient reason for disqualifying a piping configuration. Those arrangements having joints within guard vessels (AR1, AR2, AX1, F1, F3, U1, C1, and C3) have, however, already been disqualified.

11. Pump and IHX Shielding

Current design practices for LMFBR's provide for full radiological shielding of the pump and intermediate heat exchanger from the reactor. Several of the flexible piping joint configurations provide for varying degrees of shielding, while others are extremely difficult to shield. The latter are those arrangements using a straight piping run between components (AX1, F1, U1, C1) which allows "direct shine" from the reactor. The remaining configurations employ elevated piping (inverted U), and provide labyrinth shielding against direct and single-scattered radiation (see Figure II-2). The nonshielded configurations (AX1, F1, U1, and C1) have been disqualified by previous screenings; however, this discussion provides secondary reasons for their disqualification.

12. Joint Drainability

These piping configurations which utilize horizontally mounted flexible joints may not allow complete draining of the bellows elements of these joints. Thus, sodium could be trapped in these bellows following piping draining, and would solidify there if the primary heat transport system piping were brought to room temperature. However, at the present time, it is not certain that drainability of all joints is necessary because the effects of sodium solidification on bellows convolutions has not been determined. In addition, the requirement for reducing the bellows temperature below the solidification temperature has not been established (i.e., the horizontal flexible joints could be maintained above this temperature even though the remainder of the heat transport system is at room temperature, or the joint heatup and cooldown could be administratively controlled to prevent solidification during motion).

Only three of the piping configurations considered (see Figure II-2) do not incorporate horizontally mounted joints (AR1, AR2, and C3). Of these, AR1 has been disqualified because it provides no vertical motion capability, while C3 has been disqualified because it incorporates two types of joints.

Thus, configuration AR2 represents the only configuration which could provide vertical mounting for all flexible joints. Note that this configuration incorporates articulating joints, and that no configurations using axial motion or free motion joints can be devised (because these joints must necessarily be installed parallel to the motion which they are to absorb).

Because of the uncertainties on drainability requirements for commercial plants, the drainage consideration was currently not assumed sufficient for disqualification of a piping configuration. The preferred configurations (AR3h, AR3g, and AX3b), thus, each incorporate horizontally mounted joints. If such joint mounting was disallowed (based on system operating requirements and results of solidification tests), configuration AR2 would necessarily be selected. This configuration has disadvantages associated with long vertical piping runs (see Subsection II-B-7), and joint accessibility (see Subsection II-B-10). However, these disadvantages would have to be ignored in light of a drainability requirement. The individual angulation requirements for the bellows employed in configuration AR2 would be equal to or less than those for configuration AR3. Thus, the analyses of angulating bellows as reported in this document would also apply to the flexible joints of configuration AR2.

13. Viable Configurations

AR3h, AR3g, and AX3b are viable based on the screening criteria adopted in this subsection. Table II-2 provides a summary of the primary and secondary considerations which were used to disqualify configurations. Further discussions on the selection between the three screened options will be presented in Subsection II-C.

C. OPERATING REQUIREMENTS AND THEIR EFFECTS ON JOINT TYPE - CONFIGURATION SELECTION

Based on the screening discussions of Subsection II-B, three joint type - configurations were selected: two articulating (AR3h, AR3g) and one pressure-balanced axial motion (AX3b). This subsection discusses the operating requirements for the joints in these configurations, based on a preliminary large-scale breeder reactor design. These requirements were used in the bellows design studies (Section VI), which resulted in selection of the articulating

TABLE II-2
CONSIDERATIONS FOR CONFIGURATION DISQUALIFICATION

Configuration	Primary Consideration	Secondary Consideration
AR 1h	Cannot accommodate vertical motion	Long vertical piping run
AR 1g	Cannot accommodate vertical motion	Long vertical piping run
AR2h	Long vertical piping run	Joint inaccessible
AR2g	Long vertical piping run	Joint inaccessible
AX 1u	Cannot accommodate vertical motion	No elevated piping – coupled guard vessels
AX 1b	Cannot accommodate vertical motion	No elevated piping – coupled guard vessels
AX2u	Cannot accommodate vertical motion	Thrust reaction to piping
AX2b	Cannot accommodate vertical motion	-
AX3u	Thrust reaction to piping	-
F 1	No elevated piping – coupled guard vessels	Joint inaccessible
F 2	Thrust reaction to piping	-
F 3	Piping anchor penetrates guard vessel	Joints inaccessible
U 1	No elevated piping – coupled guard vessels	Joint inaccessible
U 2	Thrust reaction to piping	-
C 1hb	Uses two types of joint	No elevated piping – coupled guard vessels
C 2hb	Uses two types of joint	-
C 3hb	Uses two types of joint	Long vertical piping run

joint arrangements as the preferred configurations. The qualitative features of this section are presented in this Subsection.

1. Large-Scale Breeder Reactor Design Conditions

A prototypic set of large-scale breeder reactor operating conditions was used in the Phase I flexible piping joint studies. The plant design features and steady-state operating conditions are presented in Table II-3.

TABLE II-3
PROTOTYPIC LARGE SCALE BREEDER REACTOR DESIGN PARAMETERS
AND STEADY-STATE OPERATING CONDITIONS

Electrical rating	1500 Mwe
Number of loops	3
Reactor outlet temperature	1050° F
Primary loop ΔT	300° F
Primary loop ΔP	160-190 psi
Primary pump location	Hot leg
Primary pump suction piping size	56 in.-OD
Primary pump discharge and reactor inlet piping size	42 in.-OD

These steady-state operating conditions are representative of a typical large-scale breeder reactor flexible piping joint environment. A hot leg rather than cold leg pump was used because this results in the high-pressure pump discharge piping also being at high (hot leg) temperature. This high-pressure, high-temperature environment presents the most difficult design conditions for flexible joints. Satisfaction of these conditions would demonstrate that both hot leg and cold leg pump flexible joint designs are feasible.

In addition to the steady-state conditions, there are cyclic loading conditions and hold periods which must be included in the design analyses. Table II-4 presents typical normal operating temperature and sodium flow

cycles for the hot leg (worst design condition). These cycles were based on the Clinch River Breeder Reactor primary pump umbrella transients,⁽¹⁾ extrapolated to large-scale breeder reactor conditions.

TABLE II-4
NORMAL OPERATING CYCLES – HOT LEG

Load Cycle	Temperature Cycle End Points (° F)	Flow Cycle End Points, % of Design Flow	Number of Cycles Over Plant Lifetime
1	50-1050-50	0-100-0	13
2	400-870-400	10-30-10	80
3	400-1050-400	10-100-10	60
4	600-1050-600	10-190-10	260
5	870-1050-870	40-100-40	12,000
6	990-1050-990	80-100-80	55,000

Normal operating hold periods (for sustained loading effects) have also been defined, based on Reference 1 extrapolated to commercial plant conditions. These hold periods are presented in Table II-5.

TABLE II-5
NORMAL OPERATING HOLD PERIODS – HOT LEG

Hold Period	Sustained Temperature (° F)	Sustained Flow, % of Design Flow	Duration (hr)
1	1050	100	1.18×10^5
2	990	80	1.18×10^5
3	870	40	2.63×10^4

2. Flexible Joint Design Requirements

The normal operating conditions considered in the flexible piping joint Phase I studies include: temperature, internal pressure, motion capability, load cycling, and sustained-loading hold periods. No off-normal operating

conditions were considered. The normal operating conditions vary for different piping configurations, joint types, joint subtypes, and loop locations. The following paragraphs discuss and identify these various requirements.

For the prototype large-scale LMFBR as defined in Subsection II-C-1, there are three segments per primary loop: pump suction, pump discharge, and reactor inlet. The pump suction line uses 56-in. piping, while the other lines use 42-in. piping. At least two joint designs will thus be required, and are here defined as the 56-in. joint and 42-in. joint. The design temperature for the 56-in. joint is equal to the reactor outlet temperature (1050°F). Although the maximum operating pressure for this joint is ~2 psig, the design pressure is set at -15 psig. This results from requiring vacuum testing capability for the piping system and/or joints. The 42-in. joints operate in both the pump discharge line (hot leg, 1050°F) and the reactor inlet line (cold leg, 750°F). The maximum operating pressure for both locations is similar (190 psig). It is convenient to design a single 42-in. joint for 1050°F and 190 psig which could be used in either location. Table II-6 summarizes the joint steady-state temperature and pressure requirements.

TABLE II-6
STEADY-STATE JOINT TEMPERATURE
AND PRESSURE REQUIREMENTS

Joint Size, Piping OD (in.)	Design Temperature (°F)	Design Pressure (psig)
42	1050	190
56	1050	-15

The cyclic temperature and pressure loadings for these designs were calculated from the steady-state operating values and the load cycle conditions of Table II-4. These are presented in Tables II-7 and II-8.

The sustained loading hold periods for the joint designs were calculated from the steady-state operating values and the hold periods of Table II-5. These are presented in Tables II-9 and II-10.

TABLE II-7
CYCLIC TEMPERATURE AND PRESSURE LOADINGS –
42-in. JOINT

Load Cycle	Temperature Cycle End Points (° F)	Pressure Cycle End Points (psig)	Number of Cycles
1	50-1050-50	0-190-0	13
2	400-870-400	2-30-2	80
3	400-1050-400	2-190-2	60
4	600-1050-600	2-190-2	260
5	870-1050-870	30-190-30	12,000
6	990-1050-990	122-190-122	55,000

TABLE II-8
CYCLIC TEMPERATURE AND PRESSURE LOADINGS –
56-in. JOINT

Load Cycle	Temperature Cycle End Points (° F)	Pressure Cycle End Points (psig)	Number of Cycles
1	50-1050-50	2-0-2	13
2	400-870-400	2-1-2	80
3	400-1050-400	2-0-2	60
4	600-1050-600	2-0-2	260
5	870-1050-870	1-0-1	12,000
6	990-1050-990	0.5-0-0.5	55,000

TABLE II-9
SUSTAINED LOADING HOLD PERIODS - 42-in. JOINT

Hold Period	Sustained Temperature (° F)	Sustained Pressure (psig)	Duration (hr)
1	1050	190	1.18×10^5
2	990	122	1.18×10^5
3	870	30	2.63×10^4

TABLE II-10
SUSTAINED LOADING HOLD PERIODS - 56-in. JOINT

Hold Period	Sustained Temperature (° F)	Sustained Pressure (psig)	Duration (hr)
1	1050	0.0	1.18×10^5
2	990	0.5	1.18×10^5
3	870	1.0	2.63×10^4

The actual design conditions provided to the analytical design effort are pressures, temperatures and motions. Pressures and temperatures can be taken directly from the steady-state and cyclic operating conditions in Tables II-6 through II-10. However, motions must be calculated based on temperatures and thermal expansion calculations, and the procedure is discussed in the following paragraphs.

The three piping configurations (AR3h, AR3g, AX3b) which passed the selective screening examinations of Subsection II-B include two types of joints: articulating and pressure-balanced axial motion. For the articulating type there are four subtypes: (1) externally hinged, single bellows articulating; (2) externally gimbaled, single bellows articulating; (3) externally hinged, double-series bellows articulating; and (4) externally gimbaled, double series bellows articulating. The motion requirements of these different types and subtypes are dependent upon the exact loop arrangement in which they are

incorporated. However, for all five arrangements, each component-to-component piping segment (inverted U) is designed to lie in a single plane. Thus, all piping motions would occur in that plane if the piping system was absolutely planar, and for articulating joints, all pivot axes were parallel.

For the articulating joints, this means that either hinged or gimbaled designs could be employed, and both would require the same in-plane motion capability. The hinged unit has no out-of-plane capability, and any out-of-plane motions that exist must be absorbed by the piping and hinge assembly. The gimbaled unit would be employed if predicted, worst-case, out-of-plane motions could not be safely absorbed by the piping configuration. It is reasonable to assume that the out-of-plane motion requirements, which are a function of the installed piping arrangement, would be small relative to the in-plane motion requirements. Because of this, hinged and gimbaled joints were considered equivalent for determination of the in-plane angulation requirements. The worst-case, out-of-plane motion requirements were not considered in the Phase I studies.

The motion requirement of the articulating joints are specified as positive and negative rotation amplitudes from the unangulated (null) position. The total required angular motion range is therefore, twice the specified rotation amplitude. The rotation amplitude is used, however, because each joint will be room temperature installed at one of its extreme angulated positions (i.e., the joint is installed in a preangulated condition). As the piping system approaches operating temperatures, each joint passes through the null position. At operating temperatures, each of the joints is at the opposite end of the total motion range from when it was installed at room temperature. This piping system design technique minimizes the design requirement of the bellows because the motion is evenly distributed between the positive and negative angulation. Therefore, the detrimental effect of angulation on critical buckling pressure is minimized, and flow liner clearances can be reduced.

The bellows motion requirement of the single bellows articulating joint is equal to the specified joint motion requirement. The bellows motion requirement of the double-series bellows articulating joint is equal to one-half of the specified joint motion requirement. This is because the articulating unit for the double-series bellows joint will be designed to equally divide the joint motion between the two bellows (see Subsection II-A).

The motion requirement of the pressure-balanced axial joints are specified as positive and negative motion amplitudes from the unloaded (null) position. This method of specification is equivalent to that for the articulating joints. The axial joints are thus also installed in a predeflected condition.

The determination of the motion requirements for each size, type, and subtype of joint was based upon the thermal growths of the "inverted-U" primary loop piping segment designs which are schematically illustrated in Figure II-3. These designs encompass the implementation of either the AR3h, AR3g, or AX3b flexible joint configurations. The use of three articulating joints or two axial motion joints per loop segment is shown in the figure. The articulating joints could be either the hinged or gimbaled subtype with either single or double-series bellows. The axial motion joints are the pressure-balanced design.

The relative vertical and horizontal displacements which occur in each segment were determined by calculating the thermal growths for the piping system and components (based on a conservatively assumed set of thermal profiles). The displacements vary slightly for piping systems which employ different types of flexible joints because the lengths of the different joint types varies. Table II-11 presents the relative vertical and horizontal displacements of the three piping loop segments (see Figure II-3) using a nominal length flexible joint which is representative of the various joint types and subtypes which could be employed. The relative displacements are prescribed at the center of the uppermost flexible joint in the segment (i.e., at the top of the "inverted-U").

TABLE II-11
EFFECTIVE RELATIVE HORIZONTAL AND VERTICAL COMPONENT
DISPLACEMENTS FOR LARGE-SCALE BREEDER
REACTOR PRIMARY PIPING LOOP

Relative Displacement of:	With Respect to:	Relative Displacement Occurring at Uppermost Flexible Joint Center	
		Horizontal (in.) (positive inward)	Vertical (in.) (positive upward)
Pump	Reactor	6.7	1.2
Pump	IHX	5.0	2.7
Reactor	IHX	4.8	1.3

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PLAN VIEW - SINGLE LOOP

ELEVATION VIEW - REACTOR TO PUMP

ELEVATION VIEW - PUMP TO IHX

ELEVATION VIEW - IHX TO REACTOR

9112-29

Figure II-3. Large Scale Breeder Reactor Primary Loop Arrangement Flexible Joint Incorporation

The joint motion requirements were determined by calculating the motion for each flexible joint in each piping segment, using the actual vertical and horizontal displacements for the joint and segment configuration being considered. The maximum motion requirement for each size, type, and subtype of joint was then selected and a safety factor of $\sim 10\%$ was applied. For example, there would be six 42-in. articulating joints in each primary loop: three for the pump discharge segment and three for the reactor inlet segment. The motions for each joint were calculated, and the maximum motion plus $\sim 10\%$ was specified as the motion for all of the joints. Table II-12 presents the flexible joint motion requirements for each size, type, and subtype of joint. As previously mentioned, the angular motion requirements for the single and double-series articulating joints are not the same because the joint lengths are different, and this impacts the piping loop segment design (which impacts the thermal growths and joint motions).

TABLE II-12
DESIGN MOTION REQUIREMENTS FOR FLEXIBLE JOINTS

Piping OD (in.)	Joint Motion For		
	Single-Bellows, External Hinge or Gimbal, Articulating	Double-Series Bellows, External Hinge or Gimbal, Articulating*	Pressure- Balanced Axial Motion
42	$\pm 2.0^\circ$	$\pm 1.4^\circ$	± 2.8 in.
56	$\pm 2.0^\circ$	$\pm 1.5^\circ$	± 3.7 in.

*For double-series unit, bellows motion is one-half of joint motion.

3. Selection of Articulating Configurations

Section VI details the bellows screening examinations based on the preceding joint design requirements. These examinations resulted in disqualification of the pressure-balanced axial motion joint due to pressure instability (squirm) problems. Qualitatively, this occurs because the bellows linear motion range for the axial designs are approximately three times greater than the effective axial motions (radius times sine of angle change) of the articulating designs. This results because the horizontally mounted axial joint (see Figure II-2, configuration AX3) must absorb the total horizontal thermal expansion motion between components.

In contrast, the three articulating joints (configuration AR3) divide the thermal motions among themselves, using the piping between as a spool piece, resulting in lower effective linear motions for the bellows elements of these joints. This larger motion requirement of the axial motion joints can be met with acceptable motion stress levels by either: increasing the bellows length, convolution height, or number of plies; or decreasing the total thickness or pitch. However, all of these changes (except increasing the length) will increase the pressure stresses. Furthermore, increasing the length lowers the critical buckling pressure. For the motion and pressure requirements of Tables II-6 through II-11, an axial motion joint geometry could not be found which would have reasonably low stresses and still remain stable. Additional details of the analysis and conclusions are presented in Section VI, leading to elimination of configuration AX3b based on present system requirements.

4. Use of Cold Leg Pump

Based on the use of a hot leg pump in the primary loop, the bellows design studies (see conclusions of Subsection VI-D) have preliminarily demonstrated

that non-optimized, 56-in. and 42-in. flexible piping joints could probably employ a currently Code-approved elevated-temperature material (Type 316 stainless steel). The 56-in. joint Type 316 stainless steel bellows design (pump suction hot leg piping) is probably fabricable with only slight improvements in present forming techniques. The 42-in. joint bellows were designed for the worst of the two normal operating environments: the pump discharge hot leg piping rather than the reactor inlet cold leg piping. The preconceptual evaluations revealed that a bellows designed for such conditions could possibly employ Type 316 stainless steel, but would require significant development in both fabrication and analytic capabilities. In comparison, a bellows designed specifically for the high-pressure, low-temperature environment of the 42-in., cold-leg, reactor inlet piping is probably fabricable with minor improvements in current techniques.

These facts lead to an important conclusion with regard to early large-scale breeder reactor incorporation of flexible joints designs using currently approved materials: the primary loop environment is better tailored to such designs when a cold leg pump is employed. This results because the high pressure piping is at cold leg temperatures in such a primary loop arrangement. The 56-in. joint designs would then be applicable to the reactor outlet (hot leg, low pressure) and pump suction (cold leg, low pressure) piping, while a 42-in. low temperature design could be employed in the pump discharge piping (cold leg, high pressure). Bellows for both designs would probably be fabricable using Type 316 stainless steel, with only minor development of current manufacturing processes. In retrospect, this would provide a relatively rapid means of employing flexible piping joints in large scale breeder reactor designs, because current material technologies could be employed.

D. ADDITIONAL DESIGN CONSIDERATIONS

There are several additional features which were considered during the Phase I flexible joint studies. These were: (1) number of bellows plies, (2) redundant designs, (3) leak and incipient failure detection, (4) flow liners, (5) motion limiters, and (6) articulation unit lubricants. The following subsections qualitatively discuss each of these design features.

1. Number of Bellows Plies

Two general groupings of bellows can be made with respect to the number of plies: single-ply and multi-ply. Single-ply bellows are defined by a continuous boundary thickness. The multi-ply bellows have a boundary thickness consisting of two or more tightly spaced layers. The gapping between the layers, the convolution geometry, the surface roughness, and the applied loadings all influence the amount of structural interaction between adjacent layers. Therefore, the multi-ply configuration is inherently a more sophisticated flexible element than the single-ply configuration. Additionally, the fabrication techniques for multi-ply bellows encompass those for single-ply bellows, while requiring additional consideration being given to the layered construction. These two factors are central to many objections raised against the use of multi-ply bellows. However, with respect to performance characteristics, they have definite advantages over single-ply bellows, and because of these advantages have had numerous applications in aerospace and in other industries.

a. Advantages of Multi-Ply Construction

One of the critical assumptions often made with multi-ply bellows analyses is the behavior between adjacent plies. From stiffness and strain gage tests, there appears a tendency for the plies to move relatively independently under motion loadings, while "locking" together to resist pressure loadings. This tendency is affected not only by geometric parameters (such as gapping between layers), but also on the magnitude of the loading itself. Regardless, the tendency is to lower the motion stresses per unit pressure stress for a multi-ply bellows versus a single-ply bellows (with the same total thickness). An alternative viewpoint is that the pressure stresses are directly proportional to the total thickness, while the motion stresses are inversely proportional to the individual ply thickness. Therefore, after designing the total thickness to withstand the pressure loading, the motion stresses can be minimized to acceptable levels by increasing the total number of plies. This results in the following advantages over single-ply designs:

- 1) For a given loading condition, multi-ply bellows can be made shorter than a single-ply bellows with the same convolution shape and total thickness (decreasing length increases the motion stresses);

- 2) For a given loading condition, multi-ply bellows may be made thinner than a single-ply bellows with the same convolution shape and overall length (total stress is often the governing design parameter);
- 3) For a given loading condition, multi-ply bellows may be made with more convolutions than single-ply bellows with the same total thickness and overall length (formability is related to individual ply thickness rather than total thickness);
- 4) For a given loading condition and material selection, a multi-ply bellows may be the only feasible design choice for a particular flexible joint (increasing overall length decreases stability coupled with item (1) above).

Other advantages of multi-ply over single-ply construction are:

- 1) There exists the potential to monitor the configuration for leakage between plies or for ply failure. This would then provide time to remove the loading on the bellows before loss of the pressure retaining boundary occurs. A discussion of this method is presented in Subsection II-D-3.
- 2) Inherent in the multi-ply design is a certain (usually undefinable) amount of friction damping which would be advantageous with respect to bellows vibratory response (flow-induced vibration, earthquakes, etc.).

b. Disadvantages of Multi-Ply Construction

As stated before, two factors central to many individual's objections to the use of multi-ply configurations are the structural interaction between layers, and the additional fabrication techniques required above those needed for single-ply designs.

The amount of structural interaction between layers is a function of geometric and loading parameters, which does not lend itself to a simple characterization. Therefore, incorporating this effect into an analysis is at best approximate and is difficult to substantiate. Most of the methods currently assume either full interaction or no interaction at the interface surfaces. A few methods

have assumed directionally dependent interactions with one of these methods modifying the amount of interaction based on stiffness tests. A non-linear method has also been outlined which will incrementally modify the location of interactions as deformations accumulate. However, all of the approaches are limited by the inability to adequately understand the actual behavior which is to be mathematically modeled. This inability is largely the result of present experimental deficiencies, and lack of good nondestructive dimensional examination.

Additionally, modification of the analytic efforts with experimental testing (as is done in some fatigue life predictions) can aid in minimizing the sophistication needed in the development effort. However, this particular problem is completely nonexistent in single-ply bellows, and the additional analytical effort and uncertainties are a disadvantage to the use of multi-ply bellows construction.

The key additional fabrication techniques needed for multi-ply construction are:

- 1) Techniques to assure close tolerancing of ply composite makeup to prevent large gaps between plies not accounted for in the analyses;
- 2) Special cleaning techniques to prevent contamination problems between plies, which may have detrimental effects on metallurgical properties or form vapor pressures which can "blow apart" the ply construction;
- 3) Special welding techniques at the ends which would prevent leakage behind the inner layers while maintaining a smooth transition to the next assembly; and
- 4) Techniques for working with thinner materials which are more sensitive to handling damage (scratches, nicks, etc.,) and weld degradation.

All of the above add to the difficulty in assuring good quality control and product performance, and are a disadvantage particular to multi-ply construction.

Other disadvantages are the possibility of friction damage due to interply rubbing; the inability to make thin-to-thick end transitions in the parent material; and the existence of built-in notches at the ply faying surfaces on end joints (which can act as stress risers). A final consideration, as so far identified, is the performance of the multi-ply configuration in an internal vacuum situation.

Using conventional welding techniques, the pressure between plies is near atmospheric. Subjecting the inner region of the bellows to a vacuum condition will result in atmospheric pressure (or higher at higher temperatures) being applied on the outer surface of the innermost ply. In some designs, with a low value of maximum internal pressure, the stresses induced by the vacuum situation could significantly exceed the allowable levels. Designing for this pressure situation would require individual ply thicknesses comparable to a single-ply construction designed to the same specifications. This thus removes any stress advantages in going to multi-ply construction. On the other hand, electron beam welding techniques result in a near vacuum between layers, which would then create a bearing transfer from outer plies to inner plies upon application of an internal vacuum loading. The advantages of a multi-ply construction could then be preserved.

2. Redundant Designs

The externally hinged or gimbaled articulating joint designs can incorporate redundant bellows elements. This is accomplished by completely enclosing the primary (sodium pressure boundary) bellows with a concentric secondary bellows. The primary and secondary bellows are not in contact, and the annulus between them may be evacuated or filled with an inert gas (between 0 and 15 psia). Since only the primary bellows is subjected to the sodium pressure loadings, and since the secondary bellows can be designed with a longer total lifetime than the primary bellows, a "true redundancy" can be realized in this "concentric bellows" joint design. This true redundancy is compared to the redundancy which, supposedly exists in a multi-ply bellows. This latter redundancy is, in fact, only an over designed number of plies or ply thickness, and the redundant nature of these units is essentially a misnomer.

The concentric bellows articulating joint designs will, of course, be larger, more complex, and more difficult to manufacture than their nonredundant counterparts. They do, however, offer significant safety features over the nonredundant design, and (as discussed in Subsection II-D-3) provide the capability of monitoring for failures of the primary or secondary bellows elements. One of the four Phase I preconceptual flexible joint designs incorporated the concentric bellows redundancy concept. Refer to Section VII for a discussion of this design.

3. Leak and Incipient Failure Detection

Two methods of leak detection have been employed in flexible joint designs. For multi-ply bellows, leak monitors can detect pressure changes between plies to warn of cracks developing in the bellows element. For the concentric bellows redundant joint design (see Subsection II-D-2), the annulus between the primary and secondary bellows can be monitored to detect failures. There are, however, some important considerations in the application of either of these methods.

For the multi-ply bellows, a through-crack occurring in a ply will cause the environment on either side of the ply to be the same. If the original environments on either side of the ply were different, a transducer which can detect this difference (e.g., the presence of sodium vs vacuum) may be used to provide a failure monitoring system. Note, however, that for multi-ply bellows with more than two plies, it may be impossible to establish an original variance of environment between inner plies. Therefore, a through failure in an inner ply will not be monitored. Additionally, all plies are subjected to similar loading conditions, and failure of one ply will place an additional load on the remaining plies. This could lead to cascading failures, and could provide a very limited time for shut-down of the system upon detection of the first ply failure.

Monitoring the annulus of the concentric redundant joint can provide indication of failure of the primary bellows or secondary bellows. The monitoring device must be capable of discriminating between the sodium, the annulus atmosphere, and the joint environment atmosphere. This requires that the annulus and joint environment atmospheres be different. If this monitoring sophistication is not required (i.e., the presence and not the location of the failure is only required), a simple pressure sensor can be employed with an evacuated redundant annulus.

4. Flow Liners

The Phase I studies indicate that flow liners should be incorporated in the flexible joint designs. Flow liners are internal sleeves which guide the flow through the joint as smoothly as possible. They additionally mitigate bellows temperature transients and reduce the potential for flow-induced vibration which sometimes occurs due to flow turbulence around unlined bellows convolutions. Integral flow liners are common in both articulating and axial motion joint designs.

The aerospace industry has developed and successfully used flow liners in articulating joints with angulation requirements up to five times those of a large-scale breeder reactor's joints. All four Phase I preconceptual joint designs incorporated flow liners. Refer to Section VII for a discussion of these designs.

5. Motion Limiters

Because flexible joints are designed for a limited motion capability, it is advisable to provide some intrinsic design feature to limit the motion range during operation. For articulating joints, limiters can be designed into the external hinge or gimbal assembly. In addition, "end-of-travel" devices should be provided to detect any joint which attempts to angulate beyond its design limit. The four Phase I preconceptual joint designs incorporated various types of motion limiters. Refer to Section VII for a discussion of these designs.

6. Articulation Unit Lubricants

The ability of the hinged or gimbaled flexible joint to function properly is dependent on adequate lubrication of the bearings. The environment for the LMFBFR flexible joints includes high temperatures and radiation. Under these conditions, the usual organic lubrications are unsatisfactory. In addition, there may be limited access to the joints, thus demanding a highly reliable lubricant which does not require periodic replenishment or maintenance. The Phase I studies included preliminary investigations of high temperature lubricants, and of a low temperature "cooled" bearing design which could utilize alternate lubricants.

The environmental and design conditions for the high-temperature ($>800^{\circ}\text{F}$) articulation unit bearings dictate that a highly reliable dry-film lubricant be used, which preferably would not require periodic maintenance or replenishment. Such lubricants have been developed in the aerospace industry; however, their demonstrated service lifetimes are well below the 30 years which would be required for large-scale breeder components. Dry film lubricants for high temperatures consist of coatings of inorganic compounds which serve to prevent metal contact and which have low shear strength, thus providing relatively low coefficients of friction. There are commercially available dry film lubricants

which might be used singly or in combination for the flexible pipe joint articulation unit bearings. These include ceramic-bonded graphite and lead oxide, bonded mixtures of graphite and molybdenum disulfide, and bonded calcium fluoride-barium fluoride eutectic. All of these films have been found to perform well under conditions of high load and high temperature when used on a wide variety of base materials. The general characteristics of the films are:

- The coefficient of friction decreases with increasing hardness of the bearing material.
- The coefficient of friction decreases with increasing load, up to the load-bearing limit.
- Wear-life decreases with increasing load.
- The parts must be recoated if the film is worn through
- A thermal curing cycle is used during the coating process.
- Prediction of wear life and performance reliability requires testing under conditions closely simulating service conditions.

The Phase I studies also considered an alternate hinge or gimbal lubrication system which would operate at reduced temperatures ($<800^{\circ}\text{F}$). The reduced temperature operation may be obtained by forced or natural convection cooling of the bearing assembly on the hot leg flexible joints (see Section VII). This lower temperature environment offers the possibility of utilizing a wider variety of lubricants, many of which are not of the dry film type. Certain of these lubricants could be replenished remotely during operation of the bearing, and would not require curing cycles. These lubricants offer the advantage of a lower coefficient of friction relative to the higher-temperature dry film lubricants.

III. BACKGROUND – BELLOWS DESIGN AND ANALYTIC TECHNIQUES

At present, the ASME Boiler and Pressure Vessel Code does not allow the use of flexible joints in Class 1 piping systems such as the primary loops for a large-scale LMFBR. Therefore, no formal guidance is presently available from the ASME to assist in this Phase I flexible piping joint feasibility study with respect to codability. However, the ASME has recently set up a Task Group to develop construction rules for Class 1 flexible piping joints.

It became obvious that AI should establish some form of interaction with this Task Group, not only to increase the reliability of codability predictions, but also to help expedite development of the construction rules. Towards these ends, individuals employed by AI were funded to attend ASME Code meetings and provide support for rule development. This support has included development of proposed rules, weekly meetings with Task Group members, and assimilation and release of information from both Rockwell and industry-wide flexible piping joint literature.

Recently, another ASME Task Group was set up to divide the responsibilities of rule development into two separate but interactive areas. An AI employee was selected to chair this new Task Group. Roughly, the responsibilities of the two Task Groups are as follows:

<u>Original Task Group</u>	<u>New Task Group</u>
Develop rules governing design for flexible piping joints in low- and elevated-temperature Class 1 service.	Develop rules governing materials, fabrication, examination, installation, and inspection for flexible joints in low- and elevated-temperature Class 1 service.

Through close interaction with the ASME committees, AI can continually improve its perspective of the capability of large-scale LMFBR flexible joints. In addition, AI can periodically modify the Phase II design and testing activities to expedite Code certification of the final design.

A. BELLOWS SCREENING RULES AND FAILURE MODES

Several analytic screening rules were developed and applied during the Phase I preconceptual joint selection studies (see Section VI). It was Atomics International's opinion that these screening rules should foreshadow the intent of any future flexible piping joint design rules which would be developed by the ASME Boiler and Pressure Vessel (B&PV) Committees for Section III, Class 1, elevated-temperature service. In general, the Code design rules provide a means for guarding against specific failure modes of a pressure-retaining boundary.

At elevated temperatures, the conventional failure modes considered are:

- Failure due to ductile rupture
- Collapse due to incremental distortion
- Collapse due to buckling instability
- Failure due to fatigue damage
- Creep rupture from long-term loading at elevated temperatures
- Gross distortion resulting from creep deformations and creep-enhanced ratcheting deformations
- Cumulative creep-fatigue damage
- Creep-enhanced buckling due to long-term loading.

The Phase I studies included identification of many common and uncommon bellows failure modes. A discussion of several important bellows failure modes is presented in the following paragraphs. This discussion is based on Rockwell International's bellows experience and on a survey of bellows performance literature.

One of the most common modes of failure is due to bellows fatigue damage. The failure mechanism is progression of a crack through the pressure-retaining boundary. It is caused by strain cycling due to a fluctuating load condition, such as operational motion, flow-induced vibration, mechanical vibration, or pressure cycling. Figure III-1 illustrates a flow-induced fatigue failure. No flow liner (see Subsection II-D-4) was used in this design, and the convolutions were excited in a mode with the crests basically stationary and the roots oscillatory in the

9112-30

Figure III-1. Typical Flow-Induced Bellows Fatigue Failure

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longitudinal bellows direction. The high stresses were thus meridional bending stresses at the crown. This failure mode is often evidenced by cracks running perpendicular to the initial circumferential cracks which were formed at the crests, resulting in loss of complete pieces of material.

Another example of a fatigue failure is shown in Figure III-2. It was caused by fluctuating operational loads, but was probably aggravated by score marks on the outside of the crests. These score marks resulted from the abrasive action of a braided "protection" cover on the bellows. The effects of the score marks are evident because it is more common for motion-induced fatigue failures (in U-shaped convolutions) to occur at the root area, where the meridional motion strain ranges are a maximum in the bellows.

Overpressure causes two common failure modes. The first mode is a longitudinally oriented tearing of one or more bellows convolutions. This results from circumferential membrane stresses which exceed the ultimate strength of the material. Figure III-3 shows the effect of overpressure on the outside ply of a multi-ply bellows (used for gas service). The inner bellows plies developed

Figure III-2. Typical Motion-Induced Bellows Fatigue Failure

9112-32

Figure III-3. Outer-Ply Overpressure Failure in
Multi-Ply Bellows (Gas Service)

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through-leaks, probably fatigue-induced, which loaded the outer ply to full operating pressure. Note the longitudinal tearing, the collapse of convolution roots, and the bulging of the crowns. This is an extreme example of overpressure failure, since the usual mode is a single longitudinal rupture line over one or two convolutions.

The second overpressure failure mode is instability or "squirm." A bellows under internal pressure tends to behave like a slender column subjected to an axial load. For bellows, this load consists of the hydrostatic end thrust (product of pressure and pitch area) and that part of the spring force range which appears in compression. The result of this instability is a "squirmed" bellows, such as the one illustrated in Figure III-4.

Other potential bellows failure modes identified during the Phase I studies were:

- Thermally induced failures due to large temperature gradients, particularly for thicker-walled geometries in liquid metal service
- Pressure bulging of multi-ply bellows due to contaminants trapped between plies, which create expansive gases upon heating
- Tearing, puncture, or degradation of the thin bellows material due to improper handling, weld splatter, etc.
- Corrosion related failures due to mechanisms like stress corrosion or excessive thinning from corrosion pitting
- Accumulated local distortion in a convolution due to an elastic followup mechanism
- Convolution damage due to solidification of fluid in the convolution roots and crowns; a potential problem, especially with horizontally installed bellows in liquid metal service
- Fretting failures due to frictional rubbing between plies in a multi-ply bellows, at reinforcing rings, or at the crown outer surface when an external cover is used
- Abrasive effects on unlined bellows from particles in the fluid stream, impinging on the convolutions.

9112-33

Figure III-4. Typical Squirmed Bellows

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B. BELLOWS ELEMENT RESPONSE BEHAVIOR AND ANALYTIC PREDICTION CAPABILITIES

The following analytical capabilities will be specifically addressed:

- Static stress prediction
- Fatigue life prediction
- Stiffness prediction
- Dynamic response prediction
- Instability prediction
- Pressure drop prediction
- Interply coupling and its prediction in multi-ply bellows
- Effects of added reinforcement and its prediction.

In most cases, summaries of the technical aspects and response behavior will be presented in addition to the current identified analytic state-of-the-art. Usually, the analytical method for predicting response in a single-ply bellows differs only slightly from the method used for a multi-ply configuration. Therefore, it will simplify the format by restricting the following discussions to single-ply construction. The additional analytical methods for treating interply coupling in multi-ply bellows are discussed in a separate section. The same is true for reinforced bellows versus unreinforced bellows, and a separate section includes discussion of the effects of reinforcement on both response and analytic prediction capabilities.

1. Static Stress Response and Prediction

a. Summary of Internal Pressure Stress Fields in Formed Bellows

(1) Meridional Direction

(a) Deep U-Shaped Convolutions

For this design, the maximum stresses occur near the roots and crowns and are predominantly bending stresses. The stresses at the root torus are theoretically (uniform thickness) about 10% greater than those at the crown torus. However, the greatest wall thinning usually occurs at the crowns, which tends to decrease this difference. The bending fields are such that the root torus tends to decrease in meridional radius while the crown torus tends to increase in

meridional radius (hence the name "bulging stress" is often used). Therefore, the tension field is always on the inside surface at these locations. In the root area there exists a small compressive membrane stress, making the outside surface at the root the theoretically highest stressed point in the bellows (excluding end effects). In the crown area, there is a slight positive membrane stress, making the inside surface more highly stressed than the outside surface.

An increase in the crown radius will tend to decrease the crown bending stress. Increasing the convolution depth tends to increase the maximum bending stress, and move its location away from the apex of the roots and crowns. Increasing the thickness tends to decrease the maximum stresses.

(b) Semi-Toroidal Shaped Convolutions

The maximum stresses in this design occur near the midsides of the convolutions and are primarily pure bending stresses. The bending fields are such that the tension field is on the outside surface at this location.

An increase in the meridional radius of the toroids tends to increase the maximum stresses. It also tends to introduce negative membrane stresses in the inner torus and positive membrane stresses in the outer torus. Increasing the thickness generally results in a decrease in the maximum stresses.

(2) Circumferential Direction

(a) Deep U-Shaped Convolutions

For this design, the stress field is predominantly bending with a maximum stress about one-quarter to three-quarters of those in the meridional direction. The maximum stress is compressive and usually occurs on the outside surface at the root apex. It is a combination of a bending stress (tension on inside surface) and a negative membrane stress. A positive membrane stress exists in the flats and a negative membrane stress exists in the crest area.

An increase in the convolution depth tends to increase the maximum stresses and the relative contribution of the membrane stresses. An increase in thickness tends to lower the magnitude of the maximum stresses.

(b) Semi-Toroidal Shaped Convolutions

The circumferential stress field for this design is predominantly membrane with a maximum stress about 25% greater than the maximum meridional stress.

The location of this maximum stress is on the outer surface near the midsides of the convolution. It is a combination of about three-quarters positive membrane stress and one-quarter bending stress (tension on outside surface).

An increase in meridional radius of the toroids tends to increase the maximum stresses. Increasing the thickness generally results in a decrease in the maximum stresses.

b. Summary of Axial Motion Stress Fields in Formed Bellows (Stress Discussion is for an Increase in Length).

(1) Meridional Direction

(a) Deep U-Shaped Convolutions

The stress field for this design is almost purely bending with a maximum stress on the outside surface in the root area. This is a tension stress, therefore, the pressure and motion stresses have maximums occurring in the same location and are additive for axial reduction. In the crown area the bending field gives positive stresses on the inside surface, making the pressure and motion stresses additive in this area for axial extension.

An increase in convolution crown radius will generally result in an increase in the maximum meridional stress, but will shift its location away from the apex of the root. The maximum stress tends to increase as the convolution depth is decreased. Increasing the convolution cone angle (angle between flats) also tends to increase the maximum stress.

(b) Semi-Toroidal Shaped Convolutions

For this design, the maximum stresses are positive, occur on the outside surface of the inner torus and are almost completely due to bending. The location varies from the root apex to the midside position, depending on toroidal radius and thickness.

An increase in the meridional radius of the toroids will tend to increase the maximum meridional stresses, as does increasing the thickness. Also associated with these increases in stress is a shift in the peak location towards the midside of the convolutions.

(2) Circumferential Direction

This stress field usually has a maximum value of one-quarter to three-quarters of the maximum value in the meridional direction. It is a mixture of bending and membrane regions, and has maximums located at varying locations with geometry. The only definite trends are that U-shaped convolutions exhibit more bending than semi-toroidal shaped convolutions and that membrane fields are positive on the root convolutions and negative on the convolutions for axial extension.

Overall, the deflection stresses are very sensitive to small changes in shape (such as introduced by modeling assumptions), while the pressure stresses are very sensitive to small changes in thickness. Therefore, to obtain accurate predictions of stresses in a given bellows, the bellows cross section must be carefully modeled and the actual thickness variation used.

c. Summary of Other Motion Stress Fields

Motions other than pure axial motion are associated with different flexible joint designs. For example, in the preconceptual designs examined, only the axial motion joint is restricted to purely axial deflections. The other designs introduce offset (lateral) deflections and/or angular deflections. Consideration must also be given to in-plane-rotations (twisting). These are shown in Figure III-5.

Figure III-5. Typical Bellows Element Motions

9112-34

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(1) Offset Deflection

This motion causes a double curvature in the bellows centerline and beam type meridional bending stresses in the upper and lower fibers, while inducing diminishing stresses near the mid-depth of the cross section. The stress fields at these upper and lower extremes are similar in character to those for pure axial motion. Therefore, it has been traditional practice to convert the offset displacements into equivalent axial motions with regard to those locations on the circumference which are most stressed. This then, ignores the benefits of a large low-stresses region near the mid-height of the bellows (simple trigonometry provides this conversion). Furthermore, there is an unequal distribution of "equivalent axial motion" along the length of the bellows, with the displacements increasing with the distance from the midlength position. The conversion process should then account for this unequal distribution and empirical correction curves from test data are sometimes used.

(2) Angular Deflection

This motion causes a uniform bending of the entire length of the bellows. Again, the stress fields at the circumferential location (corresponding to the extreme fibers in an equivalent beam) are the maxima and similar to those for axial motion. Therefore, a conversion to equivalent axial motion based on trigonometric relations is usually performed, and the whole bellows circumference assumed to be stressed at the maximum value.

A complication arises when the articulating bellows is operating under pressure. The bellows does not deflect into a circular arc intersecting both end point centers. It deflects instead into an arc with a smaller radius of curvature (see Figure III-6). The minimum radius of curvature decreases with increasing pressure and becomes significant as the operating pressure gets above one-tenth of the critical buckling pressure. Correction curves, based on beam column equations, are available to increase the effective axial displacement based on this pressure effect.

(3) In-Plane-Rotation of Gimballed Bellows

When a gimballed joint is rotated about both of its shearing pin axes simultaneously, a twisting (torsion) motion is imposed on the bellows. This rotation is zero when the joint is deflected about only one gimbal pin axis, and maximum

9112-35

Figure III-6. Pressure Effects on Angulating Bellows Element

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when deflected about both axes equally. The effect is to cause shear flow around the circumference of the bellows, uniform (ideally) along the meridional length. The effect of thinning would be to increase the shear stresses proportionately in the thinned locations. The shear stress also increases with mean bellows radius and decreases with overall length for a given angle of rotation.

d. Summary of Approaches to Static Stress Predictions

Several approaches to the solution of the static stresses in bellows have been used by designers. Prior to the development of the digital computer, the analyst was limited to using shell equations which could be solved in explicit form for the governing system of equations. This limited the available geometries to simple shapes, and bellows geometries were assumed to be combinations of these shapes. As the theoretical solutions for toroidal shells became more sophisticated, bellows technology adapted the more accurate explicit forms, yet the approach was still geometry limited.

Once computers became available, numerical solutions could be performed using a wide range of geometric variables. This led to the preparation of design charts which could extend the explicit representations. At about the same time, finite difference solutions became popular in shell analysis and the application to bellows was quick to follow. Then direct integration procedures were programmed so that a complex geometry could be automatically formulated as an assemblage of classical shapes. Both of these latter methods allowed varying thickness approximations and could give accurate descriptions of the three-dimensional state of stress. However, both had serious convergence problems, which required a very careful handling of program input.

The latest improvement in bellows static stress analysis has been the application of the finite element method. Boundary conditions, geometric modeling, and material properties could be made more representative of the actual bellows being analyzed. Convergence problems became manageable and additional techniques such as nonlinear analysis, time dependent properties and loadings, and dynamic solutions (which are available in many finite element method codes) became available to the analyst.

The following detailed discussion of these methods is divided into four areas:

- Explicit formulas

- Finite difference solutions
- Direct integration solutions
- Finite element solutions.

(1) Explicit Formulas

Several sets of explicit bellows stress equations were identified in the Phase I efforts. Most of these equation sets are based upon thin shell theory. The differences in the various formulations stem in a large part from the choice of:

- Numerical approximation of the classical shapes which are combined to give the total bellows geometry
- Numerical definition of the displacement fields which are assumed to occur in the bellows
- Method of approach to the solution of the governing shell equations and simplifying approximations.

(a) Salzman

The first application of shell theory to bellows analysis was in 1946 by F. Salzman.⁽²⁾ Since this approach forms the basis for the motion stress equation,⁽³⁾ which was used in the preconceptual design effort, it will be treated in detail.

Salzman first addressed the determination of the axial load deflections of a "square wave" bellows composed of cylinders and annular rings as shown in Figure III-7.

9112-36

Figure III-7. Salzman's Square Wave Bellows

He found it convenient to neglect that the inner and outer cylinder pieces had different radii by calculating the deflection of both with the mean bellows radius, $\bar{R}$. This then assumed that the crests and roots would undergo the same loadings and deformations. This is a reasonable assumption for a displacement study since the motions at both apexes are mutually compensated, and the overall bellows deflection will average out the differences at roots and crowns.

He also assumed that the hoop stresses in the annular plates could be neglected, allowing use of the simpler beam theory versus plate theory. This is done by considering the annular plates as radial beams. He modified the bending stiffness of these elemental beams, using plate bending theory, to more adequately account for the lateral (hoop) constraint to bending. He considered the beams rigid against axial deformation which provides a simple boundary condition on the "mean" cylinder.

He then imposed compatibility between the end rotation of the beams and the end rotation of the cylinder. He loaded the beams with the axial force P at their midlengths and the cylinder with end loads P and moments $Ph/2$ (h is the convolution height) as shown in Figure III-8.

9112-37

Figure III-8. Salzman's Force and Deflection Compatibility Assumptions

The strain energy terms were written for the short cylinders and inextensible beams. Using Castigliano's theorem, he determined the axial deflection, δ_o , under the load at the midpoint of the beams.

The second part of his work addressed the determination of axial deflections for a "curved ended" bellows composed of flat annular rings connected at the roots and crowns with semitoroidal sections (see Figure III-9).

9112-38

Figure III-9. Salzman's "Curved Ended" Bellows

This is more realistic of actual bellows. The result of this effort was a set of curves which related the geometric parameters of the "curved ended" bellows to a correction term, β , which would modify the simpler solution of the "square wave" bellows first analyzed.

Briefly, the determination of β follows. The slope change of the tangent line to the rounded end is assumed to behave as a sine series defined with three parametric constants. The strain energy expression for meridional bending of a curved annular differential strip is formed. For small a/r ratios (a = radius of toroidal cross section, r = centerline distance to centroid of differential strip), two terms could be considered negligible, and were dropped from the expression.

This reduced the expression to the strain energy of a conic section. Therefore, Salzman reasoned, analogous to plate bending theory, that the contribution to strain energy from the meridional component of membrane stress could be ignored, and the rounded ends were considered rigid in the meridional direction. This allowed the strain energy expression to be completely defined in terms of the trigonometric slope change parameter. Requiring continuity of slope change at the curved end-flat ring interface defined two of the three constants in the slope change expression. The third constant was selected so as to minimize the strain energy expression for meridional bending of the curved ends.

Three further simplifications were then introduced.

- Rather than integrate the strain energy expression for the curved crowns and roots over their respective radius ranges, a fixed value of radius equal to the average radius of the bellows was used. This assumed that the roots and crowns behave the same in bending and ignored any affect convolution height may have on bending characteristics of the curved ends.
- The flat annular ring was considered rigid in the radial direction.
- The hoop stresses in the flat annular ring were ignored.

Then, using Costigliano's first theorem, the deflection under an axial load at the mean radius of the bellows was computed. The deflections for numerous combinations of geometric parameters were computed, and correction factors to allow use of the "square wave" bellows solution were formed. This also allowed computations⁽⁴⁾ of maximum meridional bending stresses by:

- Using the maximum bending moment for the "square wave" bellows (at the corner)
- Expressing P in terms of the axial deflection, δ_0
- Converting the actual (curved end) bellows deflection, δ , to δ_0 by use of the correction factor, β .

The result is:

$$\sigma_{b,d}^m = \frac{3}{2} \frac{\delta E t}{\beta (1 - \nu^2) h^2 N_c}$$

where

δ = total axial displacement of bellows

E = modulus of elasticity

t = bellow's nominal thickness

β = correction factor for curved ended bellows from Salzman's graph

ν = Poisson's ratio

h = convolution height $\div 2$

N_c = number of convolutions in bellows.

Based on the assumptions used in the derivation, this meridional bending stress equation would be most applicable to bellows which:

- Had small $a/\bar{R}$ ratios
- Had negligible hoop stresses in the flat portions of the convolutions
- Had identical root and crown geometries which behaved similarly at both locations
- Had negligible meridional membrane stresses in the curved ends.

(b) Feely and Goryl

In 1950, Feely and Goryl did an experimental investigation⁽⁵⁾ of a dished flat-plate type of welded bellows and an omega bellows. Contained in the paper was some theoretical work contributed by Den Hartog to determine the stresses due to pressure and deflection. He used the simplifying assumption that the dished disks could be treated as fixed-fixed beams. This was reasonable since they were semirigidly held by welding at the inner and outer edges, and the convolution height to mean radius ratio was small (see remark by Timoshenko,⁽⁶⁾ p. 63). The resulting analysis was straightforward using beam equations, resulting in:

$$\sigma_{b,p}^m = \frac{Ph^2}{2t^2}$$

$$\sigma_{b,d}^m = \frac{3Et\delta}{h^2}$$

where

$\sigma_{b,p}^m$ = maximum meridional bending stress due to pressure

$\sigma_{b,d}^m$ = maximum meridional bending stress due to expansion

P = internal pressure

h = convolution height

t = bellow thickness

E = modulus of elasticity

δ = axial deflection of single convolution.

General expressions were also given for converting angular rotation in a hinged joint to equivalent axial expansion, and the required forces and moments to respectively expand and rotate a bellows. Also given is the optimized h/t ratio for obtaining the maximum deflection for a given set of allowable stress and pressures.

Comparison of theoretical predictions to experimental strain gage data was "surprisingly close."

(c) Clark and Anderson

Almost simultaneously, in 1950, Clark⁽⁷⁾ and Haringx⁽⁸⁾ vastly improved the existing thin shell theory for toroidal shells. They used asymptotic integrations of thin shell theory based on newly tabulated Hankel functions of order one-third. The exact second order complex differential equations for general loadings of shells of revolution were solved for various simple geometries. Later in 1950,⁽⁹⁾ Clark extended the solution to both convoluted and toroidal bellows. However, the convoluted bellows did not include any flat plate sections. For convoluted bellows, the following explicit formulae were given for certain peak stresses due to axial loading:

$$\sigma_{m,d}^c = 0.925 (1-\nu^2)^{1/3} \frac{(\bar{R}a)^{1/3} W}{t^3}$$

occurring at $\theta = \frac{\pi}{2} - \frac{1.89}{\mu}$

$$\sigma_{b,d}^m = 1.63 (1-\nu^2)^{-1/6} \frac{(\bar{R}a)^{1/3} W}{t^3}$$

occurring at $\theta = \frac{\pi}{2} - \frac{1.35}{\mu}$

where

$\sigma_{m,d}^c$ = maximum circumferential membrane stress

$\sigma_{b,d}^m$ = maximum meridional bending stress

ν = Poisson's ratio

$\bar{R}$ = mean radius of bellows

a = convolution radius

t = bellow's thickness

W = axial load per unit length of mean circumference

θ = angle around the convolution from the crest

$$\mu = \frac{6 \sqrt{12 (1-\nu^2)}}{\lambda} \quad \left(\lambda = \frac{t\bar{R}}{a^2} \right).$$

These equations are valid only for fairly large values of $a/\bar{R}$ and large values of μ , say greater than 10; which corresponds to small values of λ , say less than 0.2. Otherwise, the solution breaks down and predicts stresses and positions of peak stress which (from the symmetry of the problem) cannot be true.

Comparison with experimental data indicates fairly reasonable predictability of peak stresses in semitoroidal bellows.

Later improvement on Clark's work by Hetenyi and Timms⁽¹⁰⁾ in 1954 resulted in application to bellows configurations which are slightly arched. They also eliminated the restriction of application to geometries with large $a/\bar{R}$ ratios.

The only explicit design equation they give is for the maximum meridional bending stress occurring near the apex of the root with a magnitude:

$$\sigma_{b,d}^m = 0.59 \left(\frac{a}{R_c^2 t^5} \right) W$$

where the notation is similar to Clark's with,

R_c = distance of center of meridional "circle" from axes of rotation of shell.

They also state that the maximum circumferential stress (membrane + bending) is usually about 30% less than the σ_b^m value.

Further improvements in the asymptotic solution approach were made by Clark himself in 1958,⁽¹¹⁾ and 1963.⁽¹²⁾ In 1959, Tumarkin,⁽¹³⁾ independent of Clark's work, developed a similar asymptotic integration of the shell equations using higher order approximations. However, none of these methods resulted in any further explicit representations of stress equations.

Another application of asymptotic integrations to bellows analysis was by Anderson^(14, 15) in 1964. He extended the geometric applicability of the previous researcher's work to include bellows with flat sections between rounded ends (U-shaped convolutions).

To limit the order of permutations in the parameters to be used to generate design charts, several simplifying assumptions were made to narrow the general solution:

- The thickness of the bellows was assumed to be constant
- The convolution radius, a , was assumed identical for the crown and root locations
- The flat annular ring portions were treated as radial strips. However, compensatory modifications to the design curves were made which minimize the errors introduced by this simplification
- Bellows with curved or sloping central regions between the toroidal roots and crests were treated as U-shaped. Using Winborne's⁽¹⁶⁾ experimental study, minor corrections were made to account for this simplification.

Anderson's work resulted in a set of stress equations and supplementary charts which allowed prediction of maximum bellows stresses over a wide range of geometric parameters. The equations produced were:

$$\sigma_{m,p}^m = \frac{Ph}{2N_p t}$$

$$\sigma_{m,p}^c = \frac{P\bar{R}a}{N_p t (0.571 a + h/2)}$$

$$\sigma_{b,p}^m = \frac{PC}{2N_p} \left(\frac{h}{t}\right)^2 \left(\frac{\bar{R}}{R_i}\right)$$

$$\sigma_{b,d}^m = \frac{1.648 Et}{h^2 N_c C_D} \left(\frac{\bar{R}}{R_i}\right) \delta$$

where

$\sigma_{m,p}^m$ = maximum meridional membrane stress due to pressure

$\sigma_{m,p}^c$ = maximum hoop membrane stress due to pressure

$\sigma_{b,p}^m$ = maximum meridional bending stress due to pressure

$\sigma_{b,d}^m$ = maximum meridional bending stress due to expansion

P = internal pressure

h = convolution height

N_p = number of plies

t = bellow nominal thickness

$\bar{R}$ = mean radius of bellows

a = radius of convolution

R_i = root radius of bellows

E = modulus of elasticity

δ = total bellows axial deflection

C_p, C_D = parametric constants from applicable design charts.

Verification of the analytical solutions was done by comparing solution runs using other theoretical approaches and experimental data existing in the literature. The theoretical methods compared to are those of Salzman,⁽²⁾ Turner and Ford,⁽¹⁷⁾ Clark,⁽⁹⁾ and Dahl.⁽¹⁸⁾ The experimental data used is from the work of Turner and Ford.⁽¹⁷⁾

Overall, the agreement between Anderson's results and the other approaches is good over the range of geometries where the other approaches are valid. Agreement with experimental data is also good. The key advantage of Anderson's equations over the others is that his are not limited to specific geometric ranges, whereas the others are applicable only in limited ranges.

(d) Dahl

In 1953, Dahl⁽¹⁸⁾ presented a paper which studied the problem of a toroidal expansion joint under an axial load. He used a complimentary strain-energy solution which expressed the bending deformation as a four-term power series in the geometric variable, μ . His method of solution was valid only for small values of a/R , which made it complimentary to that of Clark.⁽⁹⁾ The only explicit stress equations given are:

$$\sigma_{m,d}^c = 2.15 (1-\nu^2)^{1/3} \left(\frac{\bar{R}a}{t^2}\right)^{1/3} W$$

$$\sigma_{b,d}^m = 2.99 (1-\nu^2)^{-1/6} \left(\frac{\bar{R}a}{t^2}\right)^{1/3} W$$

where the notation agrees with Clark's notation.

For the range μ less than 30, very good agreement to experimental stress values is shown. Since Clark's solution is valid for μ greater than 10, the overlapped region of applicability was compared. Very close agreement was shown between stress results (within 5%) of the two solutions and experimental data.

As in most of the other bellows analyses, the agreement between theory and experiment was better for stresses than for displacements. This trend is unique in continuum based theory with the one exception of beams-on-elastic foundation solutions.

(e) Turner and Ford

In 1956, Turner and Ford⁽¹⁷⁾ formed the solution to a toroidal bellows based on a Raleigh-Ritz energy procedure. It used a five-term series to express the shape of the meridional curve which could have reversed curve sections of 270° of arc, or 180° of arc (see Figure III-10).

Figure III-10. Turner and Ford's Toroidal Bellows

However, the bellows could not have any straight sections. They assumed that $a/\bar{R} \ll 1$ to simplify their analysis and neglected membrane strains in the meridional direction. No explicit formulation of stress equations were given but they cite good agreement with axial force tests done on four bellows.

(f) Laupa and Weil

In 1961, Laupa and Weil⁽¹⁹⁾ used a five-term series to express the bending deformation in analyzing toroidal sections of convoluted bellows, applicable to large μ . Convoluted bellows, with and without flat sections, subjected to both deflection and internal pressure are included. A sample problem is solved but neither design charts nor formulae are presented.

(g) Rockwell International Reports

The first Rockwell report on bellows design was the 1960 report by J. D. Wells.⁽²⁰⁾ It adapts the Feely-Goryl equation for $\sigma_{b,p}^m$ with extension to multi-ply bellows by considering the plys locked together:

$$\sigma_{b,p}^m = \frac{P}{2N_p} \left(\frac{h}{t}\right)^2$$

where N_p is the number of plys and the remaining notation is similar to that of Feely and Goryl.

The increase in stress due to thinning is considered by adjusting the above nominal value to reflect thinning measurements performed at Rockwell on hydro-formed bellows.

The $\sigma_{b,d}^m$ stress expression has no allowance for convolution shape, so it appears to be a specific application for particular bellow geometries of a theoretically based equation:

$$\sigma_{b,d}^m = \frac{1.56 Et\delta}{N_c h^2}$$

where the notation is similar to that of Clark and Anderson.

This equation is applied to multi-ply bellows by assuming each layer acts independently and by assuming that no thinning exists. The former assumption is potentially nonconservative, while the latter assumption is conservative.

The $\sigma_{m,p}^c$ equation is a modification of the thin cylinder expression, pr/t , for hoop stress:

$$\sigma_{m,p}^c = \frac{PR_{rms}}{N_p^t} \left(\frac{1}{0.57 + \frac{2h}{q}} \right)$$

where q is the bellows pitch, and the remaining notation is similar to that of Clark and Anderson. It again uses the potentially nonconservative assumption for multi-ply bellows that the plies are locked together. This stress is then increased for the effect of thinning by use of a modifying coefficient based on examination of typical hydroformed cross-sections. It is interesting to note that this equation is similar to Anderson's later work,^(14, 15) applied to U-shaped bellows, except for the use of root mean square radius, R_{rms} , instead of mean radius, $\bar{R}$.

Aside from the Anderson reports, the next Rockwell bellows report was published in 1964 by Winborne.⁽¹⁶⁾ It included strain gage data used in a comparison with Anderson's equations for static stress prediction. Later that year, Winborne prepared another report⁽²¹⁾ which used the bellows stress expressions of previous authors combined with correction terms to minimize various approximation errors.

The latest Rockwell design report⁽³⁾ was developed by the Rocketdyne Division for NASA. It is based on a Society of Aerospace Engineers specification for aerospace vehicle cryogenic duct systems.⁽⁴⁾ A computer program has been written based on this report and was used for the preconceptual design effort. Further discussions on the computer techniques appear in Subsection VI-A, where the preconceptual design analytic effort is discussed.

(h) Other Efforts

Other developments of explicit stress equations have been identified in the Phase I effort; several of these promise to expand the state-of-the-art as defined in the preceding text. Particularly promising are the efforts by R. A. Clark in 1970⁽²²⁾ which include a rigorous analysis of the errors involved in the asymptotic integration approach. Also encouraging are the recent efforts⁽²³⁻³¹⁾ of Hamada, Takezono and their Japanese associates to eliminate many of the previous approximations in the solution of stresses in U-shaped bellows. Particularly noteworthy are their shell theory approach and design curve generation for (unsymmetrical) bending of U-shaped bellows.⁽²⁸⁾ This type of effort has long been awaited by designers of articulating bellows.

(i) Summary

The common deficiencies in the explicit formulations examined in Phase I are the following:

- Inability to account for thinning effects. (However, some sources modify the equations using test data or an "effective" thickness which is rationally or conservatively selected.)
- Inability to account for differences in root and crown geometries.
- Inability to generate a complete description of the stress fields. Even at the locations of high stresses, some stress components can not be treated. This is a particular deficiency when stress intensities based on the maximum principle stress differences must be determined.
- The accuracy of the various solutions are geometry-sensitive and only a few place limitations on their applicability with restrictive geometric ranges on their design charts.
- The methods cannot directly treat nonlinear material behavior (plasticity or creep).
- The methods cannot directly address nonlinear geometry behavior (large deformations).

The main advantage of explicit methods is the ease of generating stress numbers, allowing a large set of geometries to be examined at low cost.

(2) Finite Difference Solutions

The finite-difference approach approximates the governing differential equations of shell theory by difference expressions. A mesh of points which overlays the bellows geometry to be modeled, is mathematically defined. This allows application to arbitrary shaped convolutions of varying thickness. The difference expressions are then written at each point and the resulting set of equations can be solved simultaneously once boundary conditions are imposed. Aside from the particular mesh orientation, the basic distinction between any two finite difference solutions is in:

- The order of the difference expressions used to approximate the governing differential equations.

- The method of solution of the resulting equations.

The advantages of this method are in its geometric applicability, its low computer cost, its ability to be extended to nonlinear material and geometric behavior, and its complete stress definition.

The disadvantages are the need for a high speed computer, the high cost in comparison to explicit solutions, the possibility of unconverged solutions due to an overly coarse mesh, and the scarcity of general purpose, finite difference programs.

(3) Direct Integration Solutions (Shell Programs)

The approach used here is to break the convolution meridian into segments, each of which is defined by a classical shell shape. Numerical expressions can then be used to define each segment length. Continuity conditions at the ends of each segment are combined with these resulting equations and, upon definition of boundary conditions at the extreme edges of the shell, a final set of matrix equations can be formed. This set of matrix equations can then be solved and the stresses determined by basic substitution into constitutive relationships. Sometimes this matrix can be ill-conditioned when self-equilibrating boundary loads are imposed on the shell ends. The results of the solution would then be meaningless. However, this problem can be avoided if the size of the end segments are sufficiently small in comparison to the decay length of the shell. Furthermore, internal checks are available which can assure a well behaved matrix.

The numerical integrations can incorporate smoothly varying thicknesses, and arbitrary meridional shapes can be accurately represented by the use of sufficient segments. This gives the method good geometric representation. Additional procedures can be used to model nonlinear material and geometric effects. Shell programs are also available that can treat anisotropic material behavior and nonsymmetric loading.

(4) Finite Element Solutions

The approach here is to approximate the bellows geometry by an assemblage of subdivisions (elements), where a variational principle, such as the principle of minimum potential energy, is used to define the mathematical relations for

each element. Then, using the stiffness approach, the system equations in terms of displacement degrees of freedom can be represented by symmetric matrices and their solution performed by conventional matrix algebra. Stresses are then obtained by back substitutions. The use of a sufficiently fine mesh can assure convergence to the exact solution; therefore, running the same problem with several mesh refinements is often employed to check for convergence. Some finite element programs also incorporate internal program checks which can test convergence in a single run.

A vast variety of element types exist which allow accurate modeling of very complex geometries. Nonlinear material and geometric capabilities are standard procedures for many programs and complete three-dimensional capabilities are available for nonsymmetrical problems. It is more directly applied than finite difference solutions, and can more adequately treat irregular geometries and boundary conditions than the shell programs. It can also treat complex thermal stress problems which few of the previous methods are capable of. Overall, it is potentially the most accurate of the various approaches; however, this must be traded off against higher computer costs and modeling times.

2. Fatigue Response and Fatigue Life Prediction

a. Summary of Fatigue Failures in Bellows

Fatigue failures in bellows can usually be attributed to one of three mechanisms, or a combination of them:

- Flow-induced vibration
- Mechanically induced vibration
- Functional deflections.

(1) Flow-Induced Vibration

Failures associated with this phenomenon are usually due to high-velocity flow impinging on the convolutions. If the natural frequency of the bellows is near the flow-induced exciting frequency, then a resonant condition may lead to low cycle fatigue failures. This failure mode is easily minimized by the addition of an internal liner which provides a smooth flow boundary. If liners cannot be used, the natural frequency of the bellows should be designed far enough away from any significant excitation ranges to assure that cyclic stresses are well below the

endurance limit of the material. This requires a dynamic analysis of the bellows, which will be treated in a later section.

(2) Mechanically Induced Vibration

The excitation may be from a pump or other machinery physically tied into the piping system. The excitation is usually low amplitude, high frequency pulses. Again, the natural frequency of the bellows should be intentionally shifted away from the significant excitation frequency ranges. Sharp radii or long, heavy, unsupported masses like universal joint spool pieces should be avoided. Supports, dynamic snubbers, or dampers can be efficiently placed to cut down the vibratory response.

(3) Functional Deflections

In a properly designed system, this will be the determining factor in the fatigue life of a bellows. The failures that have occurred are usually low cycle, high stress fatigue cracking. In elevated-temperature applications, creep interaction effects can also become significant. Many of the failures can be attributed to one or more of the following factors:

- Poorly defined functional requirements: The thermal piping motions or pressure surges are far in excess of the design values due to unknown factors in the system analysis or to improper specifications. Good system definition and conservative analyses can eliminate these problems.
- Improper Installation: Poor weldment leading to high stress concentrations often initiates fatigue cracking at the end connections. Joint preloading can occur if a misaligned pipe run is mechanically held in alignment during joint installation. These preloads must be avoided because they are not usually included in the design loads. Scratches, nicks or dents due to mishandling can lead to a premature initiation of cracks from the flawed area. Strict handling and welding specifications plus the use of strongback fixtures and external shielding during installation can reduce these types of failures.
- Poor quality control: Excessive thinning at the crown of formed bellows, or faulty diameter welds in welded bellows may lead to pre-

mature fatigue failures. Strict fabrication controls and inspections will minimize these effects.

- Corrosion: Proper material selection, cleaning, and passivation can eliminate early failure due to reduced thickness corrosive effects.

b. Summary of Fatigue Life Prediction Methods

To date, four distinct approaches have been identified by which bellows are designed for a particular fatigue life. The first three methods are analytical and differ in the allowable fatigue curves used for the fatigue life evaluations. The stresses may be calculated by any of the methods presented in Subsection III-B-1. The three types of allowable fatigue curves are determined from:

- Material fatigue data
- Coupon fatigue data
- Similar bellows fatigue data.

The fourth method is based solely on experimental methods. Fatigue life tests of several prototype bellows subjected to accelerated design fluctuations are used. This method is usually restricted to relatively simple design conditions and is straightforward enough to preclude discussion.

All approaches identified to date have been limited to low temperature, rapid cycling, and/or short life applications where creep effects are negligible. Discussion of the first three methods is presented in the following subsections.

(1) Material Fatigue Data

Most of the common engineering materials used in bellows have fatigue curves available in the technical literature. These curves are based on laboratory specimens machined from the various metals, and subjected to uniaxial strain cycling. Differences in the specimen metal and the metal as it exists in the bellows are due to such items as residual stresses and local strain hardening from forming, differences in temperature histories during fabrication, and the usual material property scatter between different production heats. Therefore, an undeterminable amount of error will result in any application of this type of data, and conservative safety factors must be applied to guarantee a good design. Other errors associated with using uniaxial test data also exist and are discussed in the following paragraph.

(2) Coupon Fatigue Data

To eliminate the previously mentioned material data problems, some bellows designers use fatigue data based on coupons taken from the critical locations in a representative bellows. This is often difficult and even impractical due to the extreme thinness of some designs. The major complications associated with using coupon uniaxial test data are not in obtaining representative coupons, but rather that the uniaxial test data is obtained under controlled and intentionally simplified conditions, while the material at critical locations in a bellows may be under complex conditions. This can lead to the following differences between the test and actual bellows conditions:

<u>Coupon Conditions</u>	<u>Actual Bellows Conditions</u>
Uniaxial stress field	Biaxial stress field
Zero mean stress	Possibly significant mean stress
Constant strain range	Possibly changing strain range
No strain concentrations	Strain concentrations
Failure criteria - rupture	Failure criteria - cracking

Various engineering techniques are available to help compensate for these differences. Using effective strain ranges based on a biaxial failure theory (like Tresca or distortion energy), rather than a maximum principle strain range, will better account for the biaxial effects. Adjusting the strain range versus number of cycle curves for the worst effect of mean stress will provide a conservative treatment of mean stress effects on fatigue life. Requiring the material adjacent to a peak stress location to "shakedown" to elastic action will cause the local behavior to be "elastically controlled" with less chance for changing strain ranges. Also, using elastic-plastic solutions to predict the local behavior will give a better comparison to constant strain range data. Many strain concentrations can be handled by use of theoretical strain concentration factors. Others can be treated by a sophisticated analysis like the finite element method. To reflect the difference in failure criteria (along with data scatter and other uncertainties), conservative safety factors such as 2 on strain range or 20 on number of cycles can be used.

(3) Similar Bellows Fatigue Data

When a bellows is subjected to fatigue tests at a peak stress level computed by analytical methods or strain gage data, the number of cycles to failure is often less than the number predicted from carefully polished fatigue specimen tests. This is due in part to stress concentrations, residual stress and local spots of strain hardening (or softening) not accounted for in the analysis or strain gage data. It may also be due to the inaccuracies in the uniaxial test data adjustments discussed in the previous section. Therefore, some sources incorporate bellows fatigue tests of representative bellows to modify (or verify) the analytic predictions. The scatter in this data may be either greater or less than the scatter in uniaxial tests. The reason for this is that bellows contain a sufficient number of flaws to more consistently fail due to a local concentration than polished specimens which contain only "random weak spots." However, manufacturing variations such as thinning, cold working, etc., can cause increases in the fatigue data scatter band. Therefore, if only a small number of bellows tests are performed, the scatter in uniaxial tests cannot be used to indicate a reasonable safety factor for scatter, and additional safety factors must be used for the uncertainties.

3. Stiffness Prediction

a. Need for Stiffness Prediction

The stiffness of a bellows element must be reliably predicted in order to determine the force transfers into the piping system. This then contributes to the stresses in the piping, the reaction loads at connecting nozzles, and the foundation loads of the system components.

b. Summary of Approaches to Stiffness Prediction

Explicit formulae are available in the open literature which, for many common convolution geometries, can theoretically predict the stiffness values. However, the accuracy of these solutions is often poor. This is primarily due to the fact that the spring rate is, in principle, the integration of all the stresses in the bellows. Most explicit formulae are based on a simplified definition of the stress

state and approximation of the true geometry. Frictional forces in the linkages for articulation joints can lead to stiffening of the spring rate. Additional effects from plastic straining of the material can also exist. It is thus clear why spring rates are one of the most difficult characteristics to explicitly predict with good accuracy.

Better success has been made using shell programs and finite element solutions which can better model the actual geometry and stress state. By imposing a certain deflection on one end of the model and observing the resultant forces at the opposite end, spring rate calculations can be directly obtained. Often this data is correlated with experimental spring rate tests to assess the accuracy of the stress solutions or to form empirical equations.

Since a spring rate test can be relatively easily performed, the preferred and most common method of predicting spring rates is by experiment. Often, the manufacturer provides this information as standard practice, or at least mean and variance data for a representative test lot.

One aspect of bellows spring rates must be recognized in order to assure a reasonable force transfer prediction: the spring rate is very rarely linear. Not only does the amount of stiffness vary with deflected distance, but it often depends on the direction of the loading. For axial motions, this latter effect is more significant than it is for rotational motions typical of articulating joints.

Therefore, if manufacturer's experimental spring rate data are to be used, care must be taken to note the distance displaced and the direction of displacement.

4. Dynamic Loadings and Response Predictions

a. Summary of Dynamic Loadings

Several dynamic loadings will exist in most piping applications of flexible joints. These include flow-induced vibrations, mechanically induced vibrations, pressure pulses and seismic motions. The first two have been discussed in the fatigue life prediction section and are mainly concerned with the response of the bellows element itself. The seismic loadings are more significant with respect to the manner in which the piping system, including the joint, responds. Therefore, two distinct prediction capabilities must be evaluated: one for the bellows and one for the piping system.

b. Summary of Approaches to Dynamic Response Prediction

(1) Bellows Dynamic Response

Two approaches are available. The first is to design the system using snubbers, dampers, flow liners, etc., such that the excitation input on the bellows will be diminished to negligible levels. For flow-induced vibrations, this appears quite feasible because flow liners can be employed. However, for mechanical vibrations, where the excitation is external to the component it may be difficult to isolate the joint. Thus, the second approach may be needed.

The second approach is to determine the motions in the bellows due to the excitations, and from these compute the resulting stresses. This rigorous analytic approach determines the natural frequencies and mode shapes of the component and, using a modal analysis, determines the deflections and stresses. This will require a detailed finite element solution or equally sophisticated analytical tool. A more appealing approach is to determine only the natural frequencies and compare these to the excitation frequencies. If there is a correspondence, then the bellows is redesigned and reanalyzed and the process repeated until the natural frequencies are remote from any excitation range. This allows the use of theoretical and empirical formulae that have been developed to approximate the more common bellows natural frequencies. For flow-induced vibrations, excitation frequency approximations are also available which consider the coupling between the fluid flow dynamics and the structural dynamics of common convolution geometries.

Supplementing (or replacing) this second approach are experimental techniques to determine the natural frequencies of the bellows. For systems with long horizontal bellows and low frequency excitations, the effect of fluid mass on frequency response must be considered. An experimental determination of natural frequencies may be the only realistic method for such systems. A complete experimental approach to determine the lowest frequencies would involve shaker tests for the axial modes and pressure pulse tests for the "bulging" modes.

(2) Piping System Dynamic Response

The analytic capabilities here are primarily in piping programs which can address a reduced stiffness section in a piping run (a standard capability with most programs). Therefore, the main concerns particular to the bellows design

would be with such items as nonlinear stiffness modeling and the effects of deflection limiters.

5. Instability Phenomena and Prediction

a. Summary of Instability Phenomena

A bellows can possess an elastic instability where gross deflections can initiate at elastic strain levels. With insufficient stiffness to restore the forces causing this gross deflection, plastic deformations can result, and, at elevated-temperatures, creep buckling is also a possibility. A bellows under internal pressure, but with its ends restrained against the pressure separating force, behaves as a column with a load equivalent to the pressure separating force. As with a simple column, there is a critical buckling load for this equivalent system, and this load is dependent on bellows deflections and shape imperfections.

Axial bellows deflections produce a change in critical pressure directly proportional to the resulting change in live length—extension destabilizes and compression stabilizes. Tests show that initial offset deflections have no effect on critical pressure. Initial angular deflections are a special case where the bellows centerline continuously moves laterally under increasing internal pressure as the centerline's radius of curvature decreases. If stress levels due to this deflection are high enough, permanent deformations can result without sudden buckling occurring. In some instances, gross convolution deformations can develop where convolution crowns and/or roots touch where the particular side of the bellows is compressed. A bellows may also have a torsional instability from application of a twisting moment about its axis such as might occur when mounted in a gimbal articulation unit. Initial shape imperfections may also have significant reduction effects on critical buckling pressures. This can be due to either local collapse, stress riser effects, or a combination of both.

b. Methods of Prediction

The "column" instability buckling is often predicted using Euler type buckling formulae with effective lateral bending stiffness. The lateral bending stiffness can be determined from strength of material equations, continuum based numerical methods (like shell-programs or finite element solutions), experimental tests, or a combination of these. Direct prediction of the critical buckling

pressure can also be made using large deflection - instability nonlinear analyses methods found in some computerized field stress methods. In many bellows, a purely theoretical prediction will result in large errors because geometric imperfections are ignored and/or the boundary conditions are idealized. Therefore, experimental pressurization tests should be used to modify or verify the theoretical predictions.

Analytical prediction of buckling for initially angulated (hinged) bellows is difficult since a continuous sine deflection under loading occurs. Therefore, bending theory rather than instability theory must be used and supplemented with a definition of what amount of deflection constitutes "buckling." The approach would require an accurate three-dimensional modeling of the geometric imperfections and a field stress solution approach (shell program or finite element analysis).

The torsion buckling can be analytically addressed using torsional moments calculated from torsional deflection and spring rate formulae. These are then used in torsional instability equations, or alternately, in nonlinear field stress analyses.

6. Pressure Drop Prediction

The bellows element pressure drop must be determined in order to evaluate piping system pumping requirements. Ideally, the drop should be small or negligible. For unlined and internally linked flexible joints, the flow path can be severely disturbed, resulting in very high pressure losses. However, for properly designed, externally linked joints with integral flow liners, the pressure losses are only slightly higher than those for an equivalent smooth section of pipe. Because verified pressure drop equations are available for piping, the pressure drop of these joints is reasonably calculable.

7. Interply Coupling and its Prediction in Multi-Ply Bellows

a. Summary of Interply Coupling Mechanisms

Two different interply coupling mechanisms which create a difference in behavior response between multi-ply and single ply bellows have been identified, to date. These mechanisms are due to coulomb (interply) friction between

adjacent plies and normal direction restraint of inner plies by outer plies. Each is presented separately, followed by a summary of their effects on response behavior.

(1) Interply Friction

Interply friction creates coupling forces between plies which can have pronounced nonlinear effects on multi-ply bellows response to motion loadings. The significance of this nonlinearity depends on the magnitude of the internal pressure acting when the motion loadings are imposed.

Under a motion loading, the plies initially move together as if they were one piece. This is due to an interply friction force between the plies which prevents them from sliding past each other. As the motion is increased, a point is reached where the friction force is overcome and the plies then move relatively independently of each other. An abrupt change in response behavior then occurs.

The amount of interply friction is increased as the internal pressure is increased, resulting in a longer initial motion range where the plies move in a coupled mode. The amount of interply friction is also affected by the surface roughness of contacting surfaces and the size and location of gaps between adjacent plies.

(2) Normal Direction Restaint

An internal pressure in a multi-ply bellows will cause the innermost ply to expand outward. At the contact surfaces between this ply and the adjacent ply, normal forces will be exerted on the inner ply which will restrain its deflection pattern. The normal restraining forces are due to the resisting stiffness of the adjacent ply.

In an ideal, perfectly manufactured multi-ply bellows, the entire facing surfaces of adjacent plies will be in contact and a uniform transfer of the pressure load will occur throughout the bellows. The pressure stresses will then be distributed through the total thickness in a manner analogous to concentric pressure vessel designs; i.e., as if the bellows was of continuous thickness. In actuality, gaps will exist between plies. If small, these may close up in some areas as pressure is increased, resulting in a nonlinear response behavior.

Large gapping can lead to open voids in some areas throughout the pressure load history. The presence of gapping will also cause an uneven transfer of pressure loading between plies resulting in (increased) localized load transfers. Therefore, the results of gapping will produce an overall nonlinear behavior mechanism which may be further complicated by localized behavior.

(3) Effects on Behavior Response

Interply friction tends to increase the stiffness of the bellows, resulting in increased motion stresses. Some natural frequencies are also increased and coulomb damping developed. Localized normal direction restraint due to non-uniform gapping can cause high localized pressure stresses. High bending stresses have been observed in inner plies near a gap and in outer plies at a localized contact area. Since stability, creep and fatigue behavior are dependent on the actual stress state, these responses will also be affected.

Furthermore, since both interply friction and gapping effects on normal direction restraint are discontinuous functions of motion and pressure respectively, the behavior becomes not only complex, but nonlinear. In addition, there is a coupling between the two mechanisms since interply friction is a function of normal restraining force.

b. Methods of Prediction

Although exact analysis of the multi-ply bellows stress states is beyond state-of-the-art capabilities, certain sophisticated approaches which have been proposed by researchers, will be capable of improving theoretical predictions of multi-ply bellows response. However, some of these more sophisticated methods, even when thoroughly developed, may still be hindered by a lack of precise definition of the as-fabricated bellows geometry. The following paragraphs consider various multi-ply techniques in order to illustrate the sophistication range in analytic methodology.

The simplest multi-ply approach has been to use existing explicit single-ply formulae, modified to account for ply interaction behavior. The most commonly employed assumption is independent ply action under deflection loadings, and uniform pressure load transfer from inner to outer plies under pressure loadings. This assumption implies that no friction exists between plies and that there are no interply gaps. This leads to the theoretically minimum set of stress states,

and is potentially nonconservative. In contrast, the assumption that the plies act together as a single ply under deflection loadings, and independently under pressure loadings leads to conservative predictions of stress states. However, this assumption implies an infinite friction coefficient with no contact between plies, and places a multi-ply bellows at a disadvantage in relation to a single-ply bellows of the same total thickness.

Another approach has been to computer model the bellows as a multi-layered shell with the gap between plies modeled as a thin, uniform orthotropic layer which is bonded to the adjacent plies. The material properties of this "gap layer" are usually chosen such that it is "rigid" in the normal direction and extremely "soft" in the meridional direction. As with the first-mentioned method, independent motion stresses with a fully coupled pressure resistance is assumed for the plies. The main advantages over the first method are that the stress state is treated in more detail, and the actual convolution geometry can be better approximated.

A refinement of the latter approach is to conduct experimental stiffness tests and modify the orthotropic material properties such that the bellows stiffness is accurately modeled. This in effect integrates the local behaviors to conform to overall behavior.

A more advanced proposed method utilizes a uniform orthotropic "discontinuity" layer at the ply interface which has a high compressive modulus of elasticity in the normal direction and a near zero tensile modulus. The strength properties in the meridional and circumferential directions are near zero. An iterative solution is used. The relative motion of adjacent plies in the first solution determines the material properties of the thin discontinuity layer for the next solution. This assumes a fixed initial gap with zero interply friction and the history of normal restraining interaction can then be approximately modeled.

One of the most advanced methods proposed to date uses an "exact model" of the undeformed bellows cross section based on representative sectioning tests. The loading is then applied in stages, and as contact areas between plies are indicated, a normal coupling between layers is imposed. This coupling element is capable of transferring and measuring the normal force between the two plies.

If the load is compressive (contact), an out-of-plane interply frictional force (scaled to this normal load) is subsequently applied. If the normal load is ever negative, the coupling element is removed. This method addresses all critical interaction mechanisms, but requires complex nonlinear programming.

8. Effects of Added Reinforcement and Prediction Methods

a. Effects of Added Reinforcement

Several bellows manufacturers commonly use reinforcing rings at the outside surface of the convolution roots in order to decrease the maximum stresses in the bellows which are caused by internal pressure. A portion of the bellows pressure loading is transferred into the reinforcing ring. The location of the maximum pressure stresses in the root area (of U-shaped convolutions) is then shifted away from the apex region to a point near the edge of the contact interface between the bellows and reinforcing ring. Unfortunately, the motion stresses are increased by the addition of the reinforcements; however, the rings allow the use of thinner (lower motion stress) bellows for the same pressure load. The stress field description for the deflection and pressure stresses is not easily defined, and most of the knowledge on this behavior has been experimentally determined.

Often, reinforcing rings are used to prevent root collapse rather than to lower operating stress levels. Increased pressure in bellows tends to collapse the root portions of the convolutions in meridional bending prior to rupture of the crowns due to excessive hoop stresses. When gross section distortion is defined as a failure, the use of reinforcing rings allows the higher pressure failure mode of rupture to govern the design (rather than root collapse). The use of reinforcing rings also leads to deformation patterns under internal pressure, which modify the shape of the convolution so that it is more capable of resisting pressure forces (through membrane stresses).

Axial stiffness is increased with increasing reinforcing ring contact area. Theoretically, the stiffness could be doubled if the flexibility of the root area were completely eliminated such that all motions must be absorbed by crown region deformations. In addition, the pressure stability as it relates to axial stiffness would be increased by the use of reinforcing rings.

The effect on fatigue life is a function of cyclic load definition, reinforcing ring quality control, and bellows assemblage tolerances. If the cyclic loads are basically deflection induced, then the increase in motion stresses due to the added reinforcement may actually reduce the fatigue life. A reduction in bellows fatigue life can also occur from local strain concentrations introduced by poorly finished reinforcing ring surfaces. Initial gaps in the reinforcing ring - bellows element contact region can also create strain risers. However, a reinforced bellows, when properly designed and fabricated, can use thinner material, thus lowering both the deflection stresses and pressure stresses, and theoretically increase the fatigue life.

The effect of added reinforcement on dynamic response stems basically from the increase in both mass and stiffness. Depending on the particular mode of vibration, the bellows natural frequencies may either increase or decrease. In particular, the flow-induced mode, will be primarily affected by the increase in mass, and a lowering in frequency is expected. However, in the lateral beam bending mode and "breathing" mode, the stiffness increase and mass increase will both affect the natural frequency.

b. Analytical Methods of Response Prediction

Very little information on response prediction of reinforced bellows is available in the open literature. This is a result of the complexity of the response mechanisms which prevents rigorous analytical modeling. Therefore, experimental testing has been used to modify basic theoretical approximations, and this testing has been limited to particular geometries. Most manufacturers have thus developed their own proprietary reinforced bellows analysis methods.

Two literature sources were identified in the Phase I effort which attempted to rationally create reinforced bellows analytic methods. The first source consisted of two internal AI reports^(13, 14) which used the following assumptions:

- The radii at roots and crowns are identical.
- The peak bending stress at the initial point of contact of the ring and the convolution root is equal to the maximum bending stress in the unreinforced crown.

- The slope at the initial point of contact between the reinforcing ring and convolution root can be computed as that which provides geometric continuity around the convolution.

This allowed computation of stress and stability design curves using an asymptotic integration technique and thin toroidal shell theory.

The second literature source provided reinforced bellows performance equations prepared by the Expansion Joint Manufacturers Association (EJMA).⁽³²⁾ They reference Anderson's report as the analytical basis, but state that they have modified the equations to better model test data. Comparison of the two methods indicates that the modifications were substantial.

The EJMA equations were selected for the Phase I studies and are as follows:

$$\sigma_H = \frac{H}{2A_b} \left(\frac{R}{R+1} \right) \quad (\text{pressure hoop stress, psi})$$

$$\sigma_B = \frac{P}{2n} \left(\frac{W - 0.3q}{t_p} \right)^2 C_p \quad (\text{pressure bending stress, psi})$$

$$\sigma_b = \frac{5E_b t_p^e}{3(W - 0.3q)^2 C_d} \quad (\text{motion bending stress, psi})$$

$$P_c = \left(\frac{3.4}{N^2 q} \right) \left(\frac{d_p E_b T^3 n}{(W - 0.3q)^3 C_f} \right) \quad (\text{critical pressure, psi})$$

where

$H = Pd_p q$, resultant total internal pressure force acting on the bellows and reinforcement, lb_f

P = internal pressure, psi

$d_p = d + w$, pitch diameter of bellows, in.

q = pitch of bellows, in.

$A_b = (0.57lq + 2w)t_p n$, cross-sectional area of one convolution, in.²

$t_p = t \left(\frac{d}{d_p} \right)^{1/2}$, bellows material thickness factor for one ply, to correct for thinning during forming, in.

n = number of plies

t = nominal material thickness of one ply, in.

d = outside diameter of cylindrical tangent, in.

$R = \frac{A_b}{A_r}$, ratio of internal pressure force resisted by bellows to resistance provided by reinforcement of the same material

A_r = area of reinforcing ring, in.²

E_b = modulus of elasticity of bellows material, psi

$e = \frac{\theta d}{2N}$, total equivalent axial movement, in.

θ = rotation angle, radians

N = number of convolutions in bellows

C_p = shape factor, Figure III-11

C_d = shape factor, Figure III-12

C_f = shape factor, Figure III-13

Figure III-11. C_p for Convoluted Bellows

9112-40

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Figure III-12. C_d for Convoluted Bellows

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Figure III-13. C_f for Convoluted Bellows

9112-42

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IV. FLEXIBLE JOINT BELLOWS MATERIALS

The Phase I technical studies for the flexible piping joint development program included preconceptual evaluations of joint designs using a variety of bellows material options. These options ranged from ASME, Section III elevated-temperature approved austenitic stainless steels (Type 300 series) to currently unapproved Section III elevated-temperature alloys (Inconel 718, Hastelloy C, and Hastelloy X). To support this effort, a materials survey was initiated which examined currently available data on these materials. In general, it was found that there were insufficient design properties data for the unapproved alloys to permit satisfactory definitions of Section III, elevated-temperature Code type criteria. There was, however, enough information to make quantitative estimates of properties for comparative evaluations. The major specific areas where data is lacking were fatigue properties (particularly mean stress effects) and creep-fatigue interaction characteristics.

A. GENERAL CONSIDERATIONS

The flexible joint bellows material will see service at temperatures ranging from 750° F to 1050° F. This service will involve cyclic motion stresses with superimposed pressure stresses over long hold periods. The material will be exposed to the liquid sodium reactor coolant, and also to the reactor radiation environment.

The major requirement for candidate materials is elevated-temperature strength. The design properties of greatest importance are creep, fatigue, and creep-fatigue interaction characteristics.^(33,34) Another important aspect of the metallurgical behavior of the bellows material is the response of its mechanical properties to long periods at service temperatures. This is of particular importance for materials which are age-hardened, such as Inconel 718 (one of the candidates surveyed). The concern here is that, at higher service temperatures, loss of strength may occur through overaging, and that design stresses may exceed the strength of the overaged material. It is also possible that, at intermediate service temperatures, the material may continue to age-harden to the detriment of toughness. Thus, solution-hardened materials (Hastelloy X and C) have also been surveyed during the Phase I studies.

The candidate materials are qualitatively compared to Type 300 series stainless steels in Table IV -1. These stainless steels were selected for the comparative basis because, at the present time, they (and certain other alloys of comparable strength) are the only materials for which accepted design properties for Section III, Class 1, elevated-temperature pressure vessel service exist. Table IV-1 illustrates that a multitude of metallurgical properties and fabricability considerations are important to candidate flexible joint bellows materials.

B. QUANTITATIVE MATERIAL COMPARISONS

Based on the Phase I material surveys, the following paragraphs quantitatively discuss the important materials properties of Inconel 718, Hastelloy C, and Hastelloy X relative to Type 300 series stainless steels.

The creep and stress-rupture properties of Inconel 718, Hastelloy C, Hastelloy X, and Type 316 stainless steel are shown on Larson-Miller plots in Figures IV-1, -2, and -3. It is seen that the creep-rupture strengths of Inconel 718 are approximately twice those of Hastelloy C, approximately 2.5 times those of Hastelloy X, and approximately 5 times those of Type 316 stainless steel. On this basis, Inconel 718 would be the preferred flexible joint bellows material.

Another source of comparison of the design properties of these materials is data from Section VIII of the ASME Boiler and Pressure Vessel Code. The Code-accepted maximum allowable stresses for Inconel 718, Hastelloy C, Hastelloy X, and Type 316 stainless steel for Section VIII applications are shown in Figure IV -4. It is again seen that Inconel 718 exhibits the highest value for this strength parameter.

The fatigue strength of the flexible joint material is an important design mechanical property. In determining the allowable number of cycles under design conditions, the ASME elevated-temperature Code recommended procedure requires that the allowable strain range values for fatigue be adjusted downward to account for mean stress effects and to account for creep-fatigue interaction (when an elastic analysis is used and cyclic loading is mingled with hold periods under stress at elevated-temperature). The estimated design curve⁽³³⁾ for Inconel 718 with zero mean stress is compared with that of Type 316 stainless steel in Figure IV-5. It is seen that the Inconel 718 maintains an appreciable

TABLE IV -1
COMPARISON OF CANDIDATE FLEXIBLE PIPING JOINT
BELLWS ELEMENT MATERIALS

Candidate Materials	Advantages	Disadvantages
Type 300 Series SS	Accepted ASME Code elevated-temperature properties Ease of fabrication Long-term stability Sodium system compatibility	Stress corrosion cracking susceptibility Intergranular cracking susceptibility Creep-fatigue interaction
Inconel 718	High creep strength; low creep-fatigue interaction ASME Code-accepted design properties for creep and short-term tensile Fabrication by brazing is feasible Experience in bellows applications	Low temperature age-hardening treatment Fabricability by welding questioned by ASME Code Fatigue properties not well defined Thermal stability questioned Heat treatment variation would require more extensive testing program Sodium system compatibility needs definition
Hastelloy X	High creep strength; low creep-fatigue interaction Fabrication by brazing is feasible	No high temperature properties accepted by Section III of ASME Code Fabricability by welding may be questioned Fabricability properties are not well defined Thermal stability questioned Sodium system compatibility needs definition
Hastelloy C	High creep strength Solution hardened; thermally stable Weldability shown	No Section III, ASME Code accepted properties Sodium system compatibility needs definition

Figure IV-1. Plot of Larson-Miller Parameter P vs Stress for 1% Creep Strain, Comparing Alloys Inconel 718, Hastelloy X, Hastelloy C and Type 316 SS

Figure IV-2. Plot of Larson-Miller Parameter P vs Stress for 0.5% Creep Strain, Comparing Alloys Inconel 718, Hastelloy X, Hastelloy C, and Type 316 SS

9112-16

Figure IV-3. Plot of Larson-Miller Parameter P vs Stress for Rupture Comparing Alloys Inconel 718, Hastelloy X, Hastelloy C and Type 316 SS

9112-19

Figure IV-4. Maximum Allowable Stress vs Temperature
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103

9112-20

Figure IV -5. Design Fatigue Allowables at Zero Mean Stress⁽⁴⁰⁾

advantage for zero mean stress, except in the low-cycle regime (<1000 cycles). However, when the maximum correction for mean stress effect⁽³⁴⁾ is considered (Figure IV-6), Inconel 718 is at a disadvantage. The large correction for mean stress on Inconel 718 fatigue strength is not corroborated by test data, but is derived from relationships between the fatigue, yield, and ultimate tensile strengths developed for materials with low yield-strength/ultimate-tensile-strength ratios. This relationship places a severe penalty on materials with high yield/tensile ratios, such as Inconel 718.

Figure IV-7 is a comparison of estimated no hold time fatigue curves of Hastelloy X, Inconel 718, and Type 316 stainless steel at 1200°F and zero mean stress.⁽³⁵⁾ Figure IV-8 is the same comparison at 1200°F with application of the maximum correction for mean stress. It is seen that Hastelloy X shows a significant advantage when the maximum correction for mean stress is applied to the design fatigue curve. There are insufficient data for a similar comparison with Hastelloy C. Based on room temperature fatigue strength and the yield/tensile ratios at temperature, it is expected that the design fatigue curves for Hastelloy C will be approximately the same as those for Hastelloy X. For the flexible piping joint applications, the design fatigue properties of Hastelloy C and Hastelloy X would be preferred over Inconel 718, based on the mean stress effect correction developed for materials with low yield-strength/ultimate-tensile-strength ratios.

In comparison, however, data on fatigue life of bellows operating with internal pressure (see Figures IV-9 and IV-10) show that both Inconel 718 and Hastelloy C are superior to Type 321 stainless steel (for this room temperature test data). These results indicate that Code criteria for the mean stress effects on fatigue of Type 300 series stainless steel may not be appropriate for Inconel 718. This is an area of investigation which will require suitable analytic insight and development for final justification of the correlations for mean-stress effect in higher-strength alloys.

There are virtually no pertinent creep-fatigue interaction test data for Hastelloy C and Hastelloy X. The small amount of data⁽³⁶⁾ on Inconel 718 indicate that there would be no creep-fatigue interaction effects at 1100°F , and only slight

9112-21

Figure IV-6. Design Fatigue Allowables With Mean Stress Correction⁽⁴⁰⁾

Figure IV-7. Comparison of Estimated No Hold Time Fatigue Curves of Inconel 718, Hastelloy X and Type 316 SS (Zero Mean Stress)

9112-43

9112-44

Figure IV-8. Comparison of Estimated No Hold Time Fatigue Curves with the Maximum Correction for Mean Stress (Inconel 718, Hastelloy X, and Type 316 SS)

SYMBOL	TYPE OF MOTION	INTERNAL PRESSURE (psig)	NUMBER OF SAMPLES	DEFLECTION BENDING STRESS PRESSURE BENDING STRESS
○	AXIAL	10	48	30 TO 173
△	ANGULAR	140	18	2.5 TO 6
□	AXIAL	10	27	23 TO 49
◇	ANGULAR	100	2	1.17
●	ANGULAR	100	4	2.06

9112-45

Figure IV-9. Type 321 Stainless Steel Bellows Fatigue Curves⁽⁴¹⁾

SYMBOL	MATERIAL	TYPE OF MOTION	INTERNAL PRESSURE (psi)	NUMBER OF SAMPLES	DEFLECTION BENDING STRESS
					PRESSURE BENDING STRESS
○	HASTELLOY-C	COMBINED	50	5	8.7 TO 24
□	INCONEL 718	AXIAL	10	18	50 TO ∞
△	INCONEL-X	AXIAL	10	21	1 TO ∞
⊙	INCONEL-X	AXIAL	125 EXTERNAL	2	2.9
◇	HASTELLOY-C	COMBINED	56	1	2.9
▽	HASTELLOY-C	COMBINED	126	1	2.4
◊	HASTELLOY-C	COMBINED	65	1	5.2

Figure IV-10. Nickel Base Alloy Bellows Fatigue Curves⁽⁴¹⁾

effects at 1200°F. Type 316 stainless steel suffers a severe reduction in both creep-rupture strength and fatigue life under intermittent steady and cyclic loads. The large difference in creep strength/fatigue strength ratios is responsible for the difference in behavior. Therefore, from the standpoint of creep-fatigue interaction, Inconel 718 would be the preferred material, with Hastelloy C as an alternate.

In summary, the important design properties (with the exception of effects of mean stress on fatigue life based on low yield/ultimate ratios) tend to indicate that the preferred flexible joint material would be Inconel 718, with Hastelloy C as an alternate choice. However, the preliminary Phase I materials study did not include full considerations of materials codability for Section III, Class 1, elevated-temperature service. Further studies will be required to determine the "true optimum" bellows material for large-scale breeder reactor flexible joints.

V. BELLOWS FABRICATION CAPABILITIES AND LIMITATIONS

There are four basic methods currently employed in the fabrication of bellows elements. They result in formed, welded, machined, or deposited bellows. These methods are discussed in Subsection V-A. The advantages and disadvantages of each method were evaluated and led to the selection of forming as the preferred fabrication process. A discussion of the evaluations follows each method description. Subsection V-A concludes with a summary of the key reasons for selecting the formed bellows process.

Based on this selection, preliminary contact with several bellows vendors was made during Phase I in order to determine the current manufacturing capabilities. The vendors were surveyed as to their general capabilities, especially in the areas of forming, welding, heat treating, and testing. In some cases, their methods of design, and their overall analytic capabilities were also discussed. Subsection V-B outlines a preliminary definition of bellows fabrication technology as defined by these Phase I bellows vendor surveys.

A. BELLOWS FABRICATION METHODS

1. Formed Bellows

Over three-quarters of the commercially available bellows are formed units. These units begin as either thin-walled seamless tubing or butt-welded sheet metal tubing which is cut to the required length for forming of the convolutes. Inherent in these material forms is a certain amount of anisotropy due to the grain flow during fabrication. The bellows design may benefit from this characteristic by aligning the strong material property direction along one axis of the bellows (to coincide with the direction of the maximum normal stress levels). For seamless tubing, this result is automatic; however, for welded tubing, deliberate efforts must be made to achieve this end. Additionally, planishing of the longitudinal seam weld also imparts preferred grain directions, and care must be taken to align these along the correct axis. Other potential problems with seam welded tubing are strain risers due to faulty welding, and longitudinal stiffness variation across the weld due to weld-base metal mismatch or heat treatment effects. Nevertheless, due to size limitations and better control of ply thickness, almost all large diameter tube stock is seam welded. Still, there

are length restrictions on currently available, large diameter, welded tubing which may limit the allowable bellows live length or number of convolutions unless circumferential joining of tubes is determined to be satisfactory.

Several methods are currently used to form the convolutions of formed bellows.

a. Rubber Bulge (Elastomeric) Method (See Figure V-1)

Bellows are formed one convolution at a time by radially expanding the tube at equally spaced intervals along its axis with simultaneous axial compression. This axial load is applied to an internal rubber plug which forces the tube wall outward into an external die, thereby forming a convolution. An alternate approach is to only form an initial bulge with the rubber plug then finish the convolution shape by axially compressing the bulge with external dies. Since the convolutions are formed one at a time, and the tubing is free to pull in as the convolution is pushed out, material thinning is minimized. However, either forming approach is not recommended for tight-pitched convolutions because the rubber is difficult to extrude into the convolution shape, which results in a limited rubber plug life.

b. Mechanical Method (See Figure V-2)

A segmented die is inserted into a tube against a solid mandrel which acts as a stop. The die is mechanically forced outward to form the convolution. This radial expansion is done in several steps with the tube rotated to minimize tool marks which sometimes pose a problem with this method. There is no external tooling used in this method, although final dimensions are sometimes obtained by roll finishing (see roll forming discussion which follows). Convolutions are formed one at a time. This method is normally used for small diameters and thin-wall tubing.

c. Roll Forming (See Figure V-3)

A tube is inserted over a set of rollers, one of which supports the tube while the other forms the male part of the convolution "die." Two external rollers centered about the internal "male" roller, are loaded inwardly. This forces the tube to assume the shape of the inside roller. The tube may rotate about fixed-shaft forming wheels, or the tube may be stationary with the wheels rotating about the

9112-5

Figure V-1. Rubber Bulging Method of Forming Convolutions

9112-6

Figure V-2. Mechanical Method of Forming Convolutions

NOTE : ONE CONVOLUTION ROLLED AT A TIME

9112-7

Figure V-3. Rolling Method of Forming Convolutions

tube circumference. Rolling is continued on a single convolution basis until the desired length is achieved. Thinning of 30-40% is not uncommon on roll-formed bellows, as is the possibility of creating surface defects and smearing metal over these defects in such a way that they are hidden. These detrimental effects are less significant when roll forming is used only as a final step for a partially pre-formed bellows (known as roll finishing).

d. Hydraulic Method (See Figure V-4)

The hydraulic method uses oil or water (under pressure) to force the tube out into ring dies with controlled longitudinal shortening. Convolutions may be partially formed one at a time or simultaneously; there is currently no agreement on which method provides better shape control. The final convolution shape may be made with or without the die rings in place depending on the shape desired. The "without dies" method is claimed to be advantageous since it requires less contact of the tube with the metallic tooling; however, shape control is more difficult. If reinforcing rings are used, they are sometimes utilized as part of the die, and left in place following forming. Sometimes the final shape is obtained using roll finishing (see preceding subsection). Often the forming operation is done in stages with process-annealing following each stage. A problem exists when the hydraulic fluid is oil. Possible embrittlement or carburization of the metal can occur if it is not completely cleaned prior to process-annealing or final heat treating. Residues from oil have also been known to cause pit-type corrosion.

2. Welded Disk Bellows

The process begins with blanking of donut-shaped disks from sheet material. Care must be taken to eliminate any burrs which can prevent a good fitup in subsequent operations. The disks are then shaped by either machining, stamping, or pressing. They may then be heat treated to relieve residual stresses due to the shaping operation, or to increase the strength. The shaped disks are then welded in pairs at the inner diameter followed by welding to the final configuration at the outer diameter. During this operation, a variety of welding processes are employed. A final heat treatment is used to compensate for the metallurgical degradation of the heat-affected regions. This is often different than the handbook recommended procedures because of the thinness of the material and chance

SEQUENCE OF OPERATION

- 1) CAREFULLY SELECT PROPER MATERIAL SHEET THICKNESS
- 2) ROLL SHEET AND AUTOMATICALLY WELD LONGITUDINAL SEAM
- 3) SLIP CYLINDER INTO FORMING MACHINE
- 4) INSTALL FORM-RINGS EQUALLY SPACED BY SPACERS
- 5) APPLY INTERNAL PRESSURE AND BULGE BETWEEN FORM-RINGS
- 6) REMOVE SPACERS, COMPLETE BULGING PROCESS BY APPLYING INTERNAL PRESSURE AND AXIAL LOAD
- 7) REMOVE FORM-RINGS AND COMPLETE SIZING LENGTH OF PART

AFTER BULGING

COMPLETED PART

DETAIL B

Figure V-4. Hydraulic Method of Forming Convolutions

9112-8

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of creeping under dead weight loads at the heat treatment temperatures. Some success with multi-ply welded bellows has been reported; however, this success has been limited to two-ply construction, and extreme care was used in the fabrication procedures.

The advantages of welded disk bellows over formed-bellows are: (1) ductility requirements are reduced, allowing a wider choice of materials; (2) more deflection can be absorbed per unit length, resulting in shorter flexible joint assemblies; and (3) a wider choice of performance characteristics exists because of the greater variety in convolution shapes. The major disadvantages of welded disk bellows are due to the welding process itself. Many heat-resistant alloys suffer from loss of ductility and strength in the weld heat-affected zone. These material problems will not be entirely eliminated regardless of the welding process or heat treatment process used. Despite the impressive welding techniques which have been developed in the bellows industry, fatigue characteristics in weld areas are always poorer than in the unwelded region, and are also difficult to establish. Therefore, with the large amount of welding required, fatigue failures are much less predictable in welded disk bellows than other bellow types. Also, some welded disk bellows have abrupt changes in meridional slope at the weld location which introduce geometric stress risers at the weld bead also lowering the bellows fatigue life. Finally, the cost per bellows for welded bellows is usually higher than that for formed bellows.

3. Other Welded Bellows

Sometimes a sheet is corrugated with one or more convolution shapes while in the flat condition. The corrugated sheet is then rolled into a tube and longitudinally seam welded. Often, particularly with single convolution forms, circumferential welding is used to build up the desired number of convolutions. When compared to formed bellows, most of the advantages and disadvantages for these welded bellows are the same as those for the welded disk bellows.

4. Machined Bellows

Machined bellows are either turned or ground from solid bar stock, tubing, or forged rings. Customized designs are often employed to suit specific design requirements. The advantages of this method are precise control of geometry, use of high-strength materials not available in sheet form, elimination of welds,

and high quality of the surface finish. The key disadvantages are high costs, unavailability of some metals in large enough sizes, and lengthy fabrication times.

5. Deposited Bellows

An aluminum mandrel is machined to the final surface contour of the bellows to be formed. Using either chemical deposition or electroplating, the bellows material is deposited on the mandrel. The mandrel is then dissolved leaving the finished bellows. The key advantages of this method are: (1) its ability to produce very thin walled bellows; (2) its ability to produce very small bellows; (3) its ability to produce special bellows shapes; and (4) the absence of the requirements for welding. The disadvantages of these units are the high machining costs of the mandrels, the limited diameters and thicknesses of the ply ($t_{\max} \approx 0.06$ in., $D_{\max} \approx 12$ in.), the limited types of metals which can be deposited, and the difficult control of bellows thickness.

6. Selected Commercial LMFBR Bellows Element Fabrication Method

Based on the above considerations, the forming method of fabrication was selected for the flexible piping joint bellows element. Specifically, the welded method results in too many uncertainties in performance prediction; the machined method is restrictive due to the difficulty in obtaining adequate stock sizes in some of the high-strength materials being considered; and the deposited method does not appear feasible in the sizes required (and is also limited to single-ply construction).

Because seamless tubing is not readily available in the sizes needed for a large-scale breeder reactor plant flexible joint applications, and because extensive fabrication development already exists in the areas of longitudinal seam welds for sheet stock, the convolutes for the formed bellows should probably be formed from seam-welded tubes. The hydraulic method is the preferred forming process, with possible roll finishing to final dimensions.

B. CURRENT VENDOR CAPABILITIES

During the Phase I efforts, a limited number of bellows manufacturers provided data on their fabrication capabilities. These are summarized in the following surveys.

1. Vendor A

a. General Information

This vendor has been a flexible pipe joint (bellows) fabricator for 19 years, and has been involved in the design of bellows for 15 years. Their fabrication capabilities include forming, welding, and testing of bellows from 2-in. to 35-ft diameters. They are presently limited (in the 40 to 60 in. size range) to total bellow thicknesses of approximately 0.14 in. using high strength materials and four-ply construction. They have current ASME Code Stamp authorization to the requirements of Sections III and VIII.

b. Forming

This vendor uses initial mechanical forming with final dimensions provided by roll finishing of the convolutions. For multi-ply assemblies, the concentric nested tubes are subjected to elevated-temperature treatments in a reduced atmosphere oven for several hours, to insure against inner-ply moisture entrapment. Vendor personnel claim that a maximum of 5 to 7% material thinning occurs at the centerline of each convolution, based on experience with their manufacturing process.

c. Welding

The bellows tube longitudinal seam weld (the tubes are formed from sheet) is performed semiautomatically with a gas tungsten arc process (GTA), utilizing a stake weld fixture. Post-weld examinations include visual, liquid penetrant, and radiographic. The seam welds are subjected to nondestructive examination following forming.

Subsequent to forming of multi-ply bellows, two overlapping resistance welds are performed at the end trim areas. The seam weld nuggets create a solid material boundary for trim and weld attachment of end rings, flanges, etc. End attachment welding is performed either manually or semiautomatically with the GTA process.

d. Heat Treatment

Available heat treatment facilities are limited to reduced atmospheric ovens at a maximum of 2200°F. Bellows of high-strength nickel alloys that require any type of heat treatment are sent to vendors with vacuum furnace facilities.

e. Testing

Proof-pressure, cyclic, and helium leak test facilities are maintained. Other tests, if required, are performed at an approved testing laboratory.

2. Vendor B

a. General Information

This vendor is an ASME Section III and VIII authorized fabricator, and has broadened the scope of its bellows manufacturing capability with recent acquisitions. They claim extensive experience with Inconel 718 and other high-strength alloys for fabrication of flexible piping joints for elevated-temperature high-pressure applications.

b. Forming

A pressure roll method for forming of large diameter convolutions is used. Specific information with regard to the forming capabilities (i.e., thickness, number of plies, dimensional stability, etc.) was not made available; however, individual designs would gladly be reviewed.

c. Welding

Resistance seam welding is used for the skirt end closure of multi-ply bellows and for attachment of stiffening rings to single-ply bellows. The GTA process (manual or machine), is used for end attachment. The welding department includes capability for automatic GTA pulse welding.

d. Heat Treatment

Post-weld heat treatment of materials not requiring vacuum or inert atmospheres is performed in a reduced atmosphere oven. Heat treatment of high-strength nickel alloys is performed in a 60-in. diameter by 60-in. depth vacuum furnace.

e. Testing

The vendor maintains complete nondestructive examination facilities and test equipment, plus vacuum, cyclic, and proof-pressure testing equipment.

3. Vendor C

a. General Information

This vendor has 10 years experience in the design and fabrication of flexible piping joints for the aerospace, power generation, and petrochemical industries. Standard bellows sizes range from 1-in. to 96-in. diameter. Diameters over 30 ft have been produced. In the 40- to 60-in.-diameter range, total bellows thickness is limited to 1/8 in., convolution height to about 3 in., and minimum pitch between convolutes to ~ 1 in. The optimum convolution shape for fabrication would use a minimum ply thickness, a convolution pitch of ≈ 1.75 in., and a convolution height of less than 3 in. Capabilities include forming, welding, and testing for various bellows convolution shapes and forms. They hold ASME Code Stamp authorization to Sections III and VIII.

b. Forming

All convolution forming is performed by the roll method. Each convolute is partially formed, followed by replacement of rollers and forming to the finished dimensions. Specific information concerning dimensional tolerance control was not obtained; however, they are of the opinion that the forming process can be adjusted and/or refined as necessary to meet the most critical tolerance requirements.

c. Welding

The bellows longitudinal seam welds are formed using a semiautomatic GTA process with standard stake fixtures. The weld current sources have low pulse capability. The as-welded seam is subjected to visual, radiographic, and liquid penetrant inspection. A second liquid penetrant inspection follows the forming operation.

Bellows skirt welding and end closure welding of multi-ply assemblies is performed with the GTA process, without filler metal or overlapping resistance seam welds. Welding is controlled by an internal specification similar to MIL-W-6868, in which test specimens representative of production lots are subjected to shear and peel tests plus metallographic examination. The bellows component is trimmed through the closure weld for subsequent end attachment welding. This welding is performed using a manual or semiautomatic GTA process.

d. Heat Treatment

Heat treatment uses reduced atmosphere ovens at a temperature of 2200°F maximum. Vendor facilities are available for heat treatment of nickel base alloys or other alloys requiring vacuum or inert atmospheres.

e. Testing

Proof pressure and helium leak test facilities are available. Limited cyclic testing is also available.

4. Vendor D

a. General Information

Vendor D has 61 years experience in the bellows industry. Present forming facilities permit bellows diameters somewhat larger than 48 in., although most experience is limited to sizes less than 48 in. Total bellows thickness is limited to 0.180 in. with a maximum of three plies. Convolution height for the larger bellows is limited to ~2.5 in. This vendor does not have ASME Code authorization. Current capabilities include forming, welding, and pressure testing.

b. Forming

Convolutions are formed by the rolling method, where a single internal roller expands the metal between two outer rollers. Vendor D claims that thinning at any location of the convolution can be maintained to a maximum of 5%.

c. Welding

Semiautomatic GTA welding is employed for the longitudinal seam weld through which the bellows tube is created from sheet stock. Resistance seam welding is employed for sealing the ends of multi-ply bellows or for attaching stiffening rings to single ply bellows. Conventional GTA techniques are used to weld end attachments to the bellows. Low pulsed current GTA welding is also available for this purpose.

d. Heat Treatment

Heat treatment facilities are not available at Vendor D. When required, intermediate or final heat treating is performed at a local vendor. A low-temperature oven is available for preheating multi-ply bellows tubes prior to assembly.

e. Testing

Product inspection is limited to visual, dimensional, and liquid penetrant examination. When required, radiographic inspection is performed at an independent laboratory. Proof pressure test facilities are available.

5. Vendor E

a. General Information

Vendor E has been in the bellows design and fabrication business since 1908. Their present capability for size ranges from diameters of 1/4 in. to more than 6 ft; however, the majority of their products are small diameter, thin wall units which are used in control devices. Bellows wall thickness is limited to a maximum of 0.050 in. Vendor E holds current ASME Code authorization to Section III.

b. Forming

This vendor would probably use roll forming to produce bellows in the size range required for large-scale breeder reactor service. Their experience with roll forming includes materials such as stainless steels, nickel-base alloys, and Alloy A286.

c. Welding

Vendor E maintains a weld shop which employs state-of-the-art processes including resistance seam welding, GTA welding, gas metal arc welding, and submerged arc welding. The resistance seam welding and GTA welding processes are used for most expansion joints. Semiautomatic GTA welding is used for the longitudinal seam weld (in which the tubes are formed from sheet). Complete weld inspection services are available.

d. Heat Treatment

A heat treatment furnace is available for conventional processing and annealing. There are no provisions for reduced-atmosphere or inert-atmosphere environments.

e. Testing

Facilities include proof pressure testing and helium leak testing.

6. Vendor F

a. General Information

Vendor F has been in the bellows fabrication industry since 1958. Their capabilities include forming, welding, and testing of bellows from 1/4-in. to 72-in. diameter. Multi-ply bellows are limited to a maximum of 0.125-in. thickness using two plies and a maximum diameter of 72 in. Vendor F has current ASME Code stamp authorization to the requirements of Sections III and VIII.

b. Forming

Bellows are formed using the hydraulic method, expandable die method, rolling method, or a combination of these processes. Convolution height tolerances are normally $\pm 1/8$ in. on large diameter bellows.

c. Welding

Precision rolled sheet stock is longitudinally welded using an automatic GTA process in conjunction with conventional stake welding fixtures. Most welded joints are subsequently planished. Post-weld examinations include visual, liquid penetrant, and radiographic. In-process controls are well established with quality control systems implemented which adhere to ASME Code requirements.

d. Heat Treatment

Heat treatment furnaces with reduced atmosphere, inert atmosphere, and vacuum are available at this vendor.

e. Testing

Proof pressure, cyclic, and helium leak test facilities are maintained at Vendor F.

7. Vendor G

a. General Information

Vendor G has been a bellows fabricator for 45 years. Current fabrication capabilities range from 3-in. to 50-ft diameter, with temperatures from cryogenic to 1800°F and pressures from high vacuum to 6000 psi. Total bellows

wall thickness is limited to 0.250 in., while the convolution height is limited to a maximum of 2.5 to 3.0 in. Vendor G holds current ASME Code stamp authorization to requirements of Sections III and VIII.

b. Forming

Bellows up to 74 in. in diameter are formed by the hydraulic process. Bellows greater than this diameter are roll formed. A tolerance of ± 0.125 in. can be expected on the convolution height of a ~50-in.-diameter bellows.

c. Welding

Rolled sheet stock is longitudinally seam welded into tube form using an automatic GTA process. Post-weld examinations include visual, liquid penetrant, and radiographic. No rework is permitted on the longitudinal seam weld of rolled sheet stock. Prior to forming multi-ply bellows, sequenced resistance or tungsten arc welds are performed at the end trim areas.

d. Heat Treatment

Heat treatment facilities are limited to reduced atmospheric ovens at temperatures of $\sim 2200^{\circ}\text{F}$. Vendor facilities are available for high temperature, vacuum heat treatment.

e. Testing

Proof pressure, cyclic, and helium leak test facilities are maintained at Vendor G.

VI. BELLOWS DESIGN – PHASE I EFFORT

The bellows design efforts performed during Phase I were aimed at demonstrating the feasibility of incorporating flexible joints in large-scale breeder reactor primary piping systems. Towards this end, numerous bellows were investigated and screened in order to select a reasonable number of feasible designs for further, more detailed evaluation. These studies encompassed three major activities: (1) initial computer-generated scoping, (2) reduced-scope screenings, and (3) alternate design considerations. These activities supported investigation of over 33,000 bellows designs during Phase I.

The computer-generated scoping activity was performed with a Rockwell computer analysis routine. A matrix of designs was evaluated based on design loadings and geometry ranges consistent with commercial plant operating requirements. By examining the performance trends of the early bellows designs, the matrix was extended and refined to include over 33,000 single-ply and multi-ply, unreinforced, U-shaped and semitoroidal shaped, formed bellows.

The reduced-scope screening activity paralleled the computer scoping effort. Initial "screening rules" were developed based on time-independent stress combinations and critical buckling pressures. These rules allowed selection of "optimum" design geometries based on minimum required material strengths and stability against squirm. Geometric restrictions were imposed based on vendor surveys of current and future fabrication capabilities. This initial screening reduced the overall bellows matrix to 373 geometries. Performance trends of these geometries were further evaluated, and 56 designs were selected which required minimum material strength and which were representative of the fabrication limitations. From these geometries, 12 were selected for evaluation, using the strength data collected during the Phase I material studies, and for application of the remaining screening rules. These remaining rules examined time-dependent and fatigue failure modes: quantitatively for bellows using Type 316 stainless steel (a Code-approved material for elevated-temperature service), and qualitatively for alternate materials.

An alternate design study activity investigated three bellows geometries which were not generated during the initial computer scoping effort. Multi-ply unreinforced and multi-ply root ring reinforced geometries for double-series bellows

articulating joints were examined. All screening rules were employed to evaluate the feasibility of using Type 316 stainless steel in these types of designs.

A summarization of the results of these activities appears in Subsection VI-D.

A. COMPUTER-GENERATED SCOPING FOR PRECONCEPTUAL DESIGN

A Rockwell bellows design computer program was used to generate performance predictions on over 33,000 separate bellows geometries. The characteristics which were evaluated are:

- Maximum meridional deflection bending stress
(convolution bending stress)
- Maximum meridional pressure bending stress
(convolution bulging stress)
- Maximum circumferential pressure membrane stress
(hoop stress)
- Angular spring rate
- Fundamental (beam mode) buckling pressure
(instability pressure)
- Minimum stacked height
- Minimum height required to absorb equivalent axial compression
- Dead weight.

1. Bellows Element Design Equations

A brief description of the computerized formulations used in the preconceptual design studies is presented. The nomenclature for the equations is presented in Figure VI-1. The derivation and/or limitations of the explicit equations for stress, stiffness, and stability are discussed in Section III of this report.

a. Stress Equations

The convolution bending stress, σ_b , was determined from total equivalent axial deflection, $\delta^{(2)}$,

D_i = INSIDE DIAMETER, in.

D_m = MEAN DIAMETER = $\frac{D_i + D_o}{2}$, in.

D_o = OUTSIDE DIAMETER, in.

E = MODULUS OF ELASTICITY, psi

F = RADIUS OF CURVATURE FACTOR FOR ANGULATING BELLOWS (SEE FIGURE VI-2)

h_r = REQUIRED STACK HEIGHT, in.

h_s = MINIMUM STACKED HEIGHT

H = CONVOLUTION HEIGHT = $D_m - N_p \cdot T$, in.

K_{ang} = ANGULAR SPRING RATE, $lb_f \cdot in./deg$

K_{ax} = AXIAL SPRING RATE, $lb_f/in.$

K_g = RATIO OF UNFORMED TUBE LENGTH TO FORMED BELLOWS LENGTH
 $= 0.571 + 2 \cdot N_c \cdot H/L$

L = LIVE LENGTH, in.

N_c = NUMBER OF CONVOLUTIONS

N_p = NUMBER OF PLIES

P = DIFFERENTIAL INTERNAL PRESSURE, psi

P_c = DIFFERENTIAL CRITICAL PRESSURE FOR BUCKLING, psi

R = MEAN ROOT OR CROWN RADIUS, in.

R_f = FORMING RADIUS, in.

DETAIL A

T = PLY THICKNESS, in.

W = DEAD WEIGHT, lb

X = $(T \cdot D_m/2)^{1/2} / [3(1 - \nu^2)]^{1/4}$, in.

β = BENDING STRESS SHAPE FACTOR (SEE FIGURE VI-3)

δ = EQUIVALENT AXIAL DEFLECTION, in.

δ_{ax} = AXIAL DEFLECTION, in.

Δ = LATERAL (OFFSET) DEFLECTION, in.

ζ_{IV} = SPRING RATE SHAPE FACTOR (SEE FIGURE VI-4)

σ_b = CONVOLUTION BENDING STRESS, psi

σ_B = CONVOLUTION BULGING STRESS, psi

σ_H = CONVOLUTION HOOP STRESS, psi

θ = BELLOWS ANGULAR ROTATION, radians

ρ = BELLOWS MATERIAL DENSITY, $lb_m/in.^3$

ν = POISSON'S RATIO

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Figure VI-1. Definition of Stress Equation Nomenclature

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Figure VI-2. Deflection Due to Internal Pressure of Angulating Bellows⁽⁴²⁾

Figure VI-3. Bending Stress Shape Factor⁽²⁾

Figure VI-4. Spring Rate Shape Factor⁽²⁾

$$\sigma_b = \frac{3E\delta T}{2(1-\nu^2)\beta N_c H^2} ;$$

where δ is given by:

$$\delta = \underset{\text{(Axial)}}{\delta_{ax}} + \underset{\text{(Angular)}}{\left(\frac{D_m \theta}{2}\right)} + \underset{\text{(Offset)}}{\left(\frac{3D_M \Delta}{L}\right)}$$

This equivalent axial deflection was modified for the articulating bellows to account for the effect of internal pressure on the radius of curvature. The deflection (δ) is multiplied by the factor F^{-1} ; F is derived from considerations of similar nonlinear curvature effects which are found in classical beam-column theory,⁽⁴²⁾ and is defined in Figure VI-2. The convolution bulging stress, σ_B , from internal pressure, P , was computed using:⁽⁵⁾

$$\sigma_B = \frac{PH^2}{2N_p T^2}$$

The hoop stress, σ_H , from internal pressure, P , was given by:⁽⁵⁾

$$\sigma_H = \frac{PD_o}{2N_p K_g T}$$

b. Stiffness Equations

The elastic axial spring rate, K_{ax} , which is the change in axial force required to extend or compress the live length some incremental distance, was computed by:⁽²⁾

$$K_{ax} = \frac{\pi D_m E N_p T^3}{6(1-\nu^2)\zeta_{IV} N_c H^3}$$

The angular spring rate, K_{ang} , was related to the axial spring rate through simplified force equilibrium, giving:

$$K_{ang} = \frac{\pi D_m^2 K_{ax}}{1440}$$

c. Instability Equation

The critical buckling pressure, p_c was given by:⁽⁴³⁾

$$p_c = \frac{2\pi K_{ax}}{L}$$

This models the column buckling mode with the ends rigidly restrained.

d. Minimum Stacked Height Equation

The minimum height of the bellows when fully compressed, h_s , was computed by:

$$h_s = N_c (2R - N_p T)$$

This assumes compression until the sides of the crowns and roots touch.

e. Required Stack Height Equation

The required stack height, h_r , was computed by:

$$h_r = L - \delta$$

f. Dead Weight Equation

The dead weight of the bellows, W , was computed from the geometrically calculated material volume and density:

$$W = \pi (0.571 + 2N_c H) N_p T D_i \rho$$

2. Bellows Design Loadings

The design loadings for the 42-in. and 56-in. flexible piping joint bellows were presented in Subsection II-C. Please refer to this discussion for the detailed operating requirements and their formulations.

3. Range of Geometries Investigated in Computer-Generated Scoping Studies

Table VI-1 presents the total matrix of geometries investigated during the Phase I scoping studies. These include bellows for the 42-in. and 56-in. single-bellows articulating joints and for the 42-in. and 56-in. pressure-balanced axial motion joints (main bellows only). The matrix includes over 33,000 separate geometries, for which the previously discussed design characteristics were evaluated. The further screening of these designs is detailed in Subsection VI-B. Screening analyses for the bellows of the double-series articulating joint are presented in Subsection VI-C.

The bellows internal diameters (D_i) were selected 3 in. larger than the piping diameter to guarantee room for motion clearances around a flow liner assembly.

The pitch of the bellows in Table VI-1 was varied to examine a full range of geometries. The lower limit on pitch was set by the minimum convolution radius, $R_{\min}$, as determined from considerations of common forming capabilities:

$$R_{\min} \geq 2T + \frac{N T}{2}$$

The upper limit on pitch was given by either: (1) the number of convolutes required to generate a semitoroidal shape (no flats); or (2) the theoretical limitations of the explicit equations based on convolution radius-to-height ratio.

B. REDUCED-SCOPE SCREENING FOR PRECONCEPTUAL DESIGN SELECTIONS

The designs generated in the scoping studies were reduced to 56 selected optimum geometries by application of stress and buckling screening rules.

TABLE VI-1
RANGE OF GEOMETRIES CONSIDERED IN INITIAL DESIGN EFFORTS

Live Length (in.)	No. of Plies	Ply Thickness (in.)		Convolution Height (in.)		
		Range	Increment	Range	Increment	
42-in. Articulating Joint Bellows ($D_i = 45$ in.)						
14	1	0.08-0.18	0.020	2	0.50-4.00	0.5
14	4	0.02-0.045	0.005		0.50-4.00	0.5
18	1	0.08-0.18	0.020		0.50-4.00	0.5
22	1	0.08-0.18	0.020		0.50-4.00	0.5
22	1	0.15-0.25	0.010		0.10-3.50	0.1
22	4	0.03-0.06	0.005		0.10-1.50	0.1
28	1	0.08-0.18	0.020		0.50-4.00	0.5
28	1	0.15-0.20	0.010		0.10-3.00	0.1
28	1	0.12-0.16	0.020		0.20-4.00	0.2
28	4	0.02-0.045	0.005		0.50-4.00	0.5
28	4	0.035-0.05	0.001		0.10-1.50	0.1
28	4	0.03-0.045	0.005		0.20-4.00	0.2
36	1	0.08-0.18	0.020		0.50-4.00	0.5
36	2	0.06-0.12	0.010		0.20-4.00	0.2
36	4	0.02-0.045	0.005		0.50-4.00	0.5
45	1	0.08-0.18	0.005		0.50-4.00	0.5
45	1	0.12-0.16	0.020		0.20-4.00	0.2
45	4	0.01-0.035	0.005		0.50-4.00	0.5
45	4	0.03-0.045	0.005		0.20-4.00	0.2
55	1	0.08-0.180	0.020		0.50-4.00	0.5
65	1	0.08-0.180	0.020		0.50-4.00	0.5
75	1	0.08-0.180	0.020		0.50-4.00	0.5
42-in. Axial Motion Joint Bellows ($D_i = 45$ in.)						
45	1	0.08-0.180	0.020		0.50-4.00	0.5
45	4	0.02-0.045	0.005		0.50-4.00	0.5
56-in. Articulating Joint Bellows ($D_i = 59$ in.)						
5	1	0.03-0.09	0.02		0.25-4.00	0.25
5	2	0.015-0.045	0.01		0.025-4.00	0.25
5	4	0.01-0.04	0.01		0.025-4.00	0.25
10	1	0.03-0.09	0.02		0.25-4.00	0.25
10	2	0.015-0.045	0.01		0.25-4.00	0.25
10	4	0.01-0.04	0.01		0.25-4.00	0.25
15	1	0.03-0.13	0.02		0.50-7.00	0.50
15	4	0.02-0.045	0.005		0.50-7.00	0.50
20	1	0.03-0.13	0.02		0.50-7.00	0.50
20	4	0.01-0.035	0.005		0.50-7.00	0.50
25	1	0.03-0.13	0.02		0.50-7.00	0.50
25	1	0.07-0.11	0.02		2.00-4.50	0.125
25	4	0.01-0.035	0.005		0.5-7.00	0.5
56-in. Axial Motion Joint Bellows ($D_i = 59$ in.)						
25	1	0.03-0.11	0.02		0.50-0.40	0.50
25	4	0.01-0.035	0.005		0.50-0.40	0.50

Discussions of the screening rules and the selection process are presented in this subsection. Additionally, 12 representative designs were selected and evaluated for fatigue and elevated-temperature characteristics by the application of additional screening rules. A description of these geometries and their evaluations complete this subsection.

1. Preconceptual Screening Rules

Fourteen screening rules, covering six failure mode groups, were developed to reduce the computer-generated designs. The six groups were:

- Circumferential yielding, rupture, or excessive deformation
- Root or crown plastic collapse, rupture, or excessive deformation
- Incremental distortion of convolutions
- Squirm
- Fatigue damage
- Creep-fatigue damage.

The development of the preconceptual screening rules was accomplished by consideration of the ASME Section III design philosophy, tempered with engineering judgement and simplifications. Since no present set of Code rules is applicable to the proposed Class 1 expansion joint designs, the selection of existing design rules from the Code documents is somewhat arbitrary. Furthermore, the basic philosophy behind some particular rules is not always obvious, and in some cases may not be directly applicable to bellows behavior. It was felt that a more rewarding approach would be to address specific bellows failure modes (see Subsection III-A) and formulate rules which are based on the fundamental mechanics of failure. In this way, the screening effort could be geared to that of a preconceptual design level (versus a preliminary or final design level), and a large number of alternate configurations could be examined. Nevertheless, enough overlap exists between the design philosophy used here and the intent of the current ASME Code philosophy, to allow the final reduced set of representative configurations to be qualitatively evaluated as to their feasibility for compliance with future Code rulings. However, it must be noted that these screening rule evaluations are limited by the technical level of

the preconceptual analytical efforts, as well as by the incomplete treatment of other characteristics such as: fabrication control, inspection, testing, etc.

The following subsections briefly discuss the basis for each of the generated screening rules, following a short discussion of the categorization of stress values. This preconceptual stress analysis addressed three bellows stress components:

σ_H - Pressure membrane stress in the circumferential direction

σ_B - Pressure bending stress in the meridional direction

σ_b - Deflection bending stress in the meridional direction.

The philosophy of the ASME Code is to provide allowable magnitudes for different types of stresses, depending on degrees of significance. There is little question that σ_H must be treated as a primary stress and appropriate stress limits used. However, σ_B and σ_b require further interpretation.

The stress component, σ_B , usually is a maximum at either the root or crown toroids. At the crown location, the pressure bending field creates deformations which increase the convolution radius and therefore decrease the bending stresses. In this sense, σ_B can be considered self-limiting. However, at the root location the reverse is true and eventual collapse of the convolution roots will occur, i.e., the stresses are not self-limiting. Therefore, an argument could be made that the σ_B stresses in the root area should be considered primary (introducing root ring reinforcement would prevent this collapse mechanism and eliminate this argument). However, field experience with bellows indicates safe operating pressures can exist considerably above those limited by a criteria which would invoke the conventional S_o (material allowable primary membrane stress) limits on load-controlled, membrane-plus-bending stresses which may allow some plastic deformations. In particular, the $\sigma_H + \sigma_B$ values must be below those needed to cause a permanent decrease of 7% in spaces between adjacent convolutions after a pressure test to 1.5 times the design pressure. The basis for this rule is not stated, making it difficult to ascertain a Code precedence on categorization of σ_B . Some clarification is, however, contained in Code Case 1177-7 ("Under

what conditions may an expansion joint be used as an integral part of a Code pressure vessel?" — Section VIII Rules). However, the clarification is that "reasonable judgement" is to be exercised:

Note

In considering the acceptability of joints under Par. U-2(g) in combination with Pars. UG-22 and UG-23, it is not the intent that the combination of direct and localized and secondary bending stress need be held to the Code tabulated allowable stress, although the direct (membrane) stress must be held to the tabulated maximum allowable stress value. The Committee recognizes that such localized stresses primarily influence fatigue performance and it is intended that the manufacturer and inspector will exercise reasonable judgment in assessing a joint for an individual application. Footnote 2 to Par. UG-23(b) recognizes that high localized and secondary bending stresses may exist in Code vessels and the stresses inherent in accepted details can be of assistance as a guide.

Overall, it appears that treating the σ_B stress as primary, and adopting the corresponding Class 1 Code philosophy may be overly conservative with respect to a future Code ruling on Class 1 expansion joints. To address this issue, two approaches were taken. The "optimum" geometries were selected using time-independent screening rules which treated σ_B as primary. Then, all remaining evaluations considered σ_B in two ways: both as a primary stress and as a secondary stress. The differences are significant in the creep rupture and creep deformation failure mode considerations.

The rational categorization of σ_b (deflection bending stress) is also uncertain at this time, particularly when the complexity of creep effects is introduced. The uncertainty concerns the degree of significance of σ_b with respect to incremental distortion and to creep damage during elevated-temperature hold periods. The screening rule used to select optimum geometries considered σ_b as a secondary stress. All further evaluations considered σ_b in two ways: as a secondary stress and as a peak stress.

The classification of σ_b is further complicated by the possibility of elastic follow-up, which could localize the displacements of a bellows into only a few

convolutions. Assuming this effect would occur could lead to classification of σ_b as a primary stress. However, with strict fabrication and inspection controls to guarantee that individual convolutions will have nearly identical geometries, this mechanism should not be a problem. This high level of quality is expected of a Class 1 component, and the primary stress categorization was not considered reasonable for σ_b .

During Phase II of the program, efforts will be directed toward firmly establishing degrees of significance of the σ_B and σ_b stresses. These efforts will in large part be interfaced with ASME task forces on bellows, and the resulting perspective of the Code's treatment of the stresses will be used to update the Phase I results, if necessary.

a. Rules for Circumferential Yielding, Rupture or Excessive Deformation

To prevent gross section yielding (or rupture), the hoop stress should be limited to a safe level below the material's yield point (or ultimate strength). Adapting the design condition criteria for primary membrane stress used in Code Case 1592, categorizing the hoop stress as primary membrane, and ignoring membrane components of other principal stress fields, results in the first screening rule:

$$\sigma_H \leq S_o \quad \dots \text{Screening Rule (1a)}$$

where

S_o is the material's allowable primary membrane stress value as defined in Code Case 1592, or S as defined in Section VIII. Note that this value is based in part on the material's 100,000-hr creep properties, making the rule time-independent, but the allowable value time-dependent.

Secondly, the pressure hoop stress should not be allowed to approach that magnitude causing creep rupture during sustained hold periods at elevated temperature. A 50% safety margin is introduced, resulting in:

$$\sigma_H \leq \frac{2}{3} \sigma_{\text{rupt}} \quad \dots \text{Screening Rule (1b)}$$

where

σ_{rupt} is the material's average stress to rupture at the fluid temperature for the time of application of σ_H .

To prevent excessive distortions in a gross section, two screening rules were used. The first rule limits the allowable pressure hoop stress to values at which tertiary creep will not exist. This avoids creep tensile instability by development of voids or microcracks associated with third-stage creep. A 25% safety margin is introduced giving:

$$\sigma_H \leq 0.8 \sigma_{3c} \quad \dots \text{Screening Rule (1c)}$$

where

σ_{3c} is the minimum stress to cause the onset of tertiary creep of the material at the sodium temperature during the time period corresponding to application of σ_H .

The second screening rule for limiting excessive inelastic deformations was selected to prevent more than 1% accumulated strain due to the pressure hoop stress:

$$\sigma_H \leq \sigma_c \quad \dots \text{Screening Rule (1d)}$$

where

σ_c is the minimum stress to produce 1% total strain in the material at the sodium temperature for the period of time corresponding to application of σ_H .

The actual application of Screening Rules (1b), (1c), and (1d) was made using a linear damage assumption:

$$\sum \frac{t_i}{T_i} \leq 1.0$$

where t_i is the actual time at pressure and temperature load condition i , and T_i is the allowable time at this condition obtained from the appropriate time-dependent curves.

b. Rules for Root or Crown Plastic Collapse, Rupture,
or Excessive Deformation

To prevent convolution collapse or rupture, the combined primary membrane-plus-bending stress intensity should be limited to a safe level. Adapting the design condition criteria in Code Case 1592, categorizing the bulging stress as primary bending (see Page 138), and ignoring other principal stress membrane and bending components results in:

$$\sigma_H + \sigma_B \leq 1.5 S_o \quad \dots \text{Screening Rule (2a)}$$

Under the influence of creep, the pressure bending stress field resisting the externally applied meridional moment will redistribute itself in such a manner as to lower the outside fiber stress. The amount of this beneficial relaxation depends on the temperature, time duration of loading, load magnitude, and material properties. Assuming ideal elastic-plastic behavior, the reduction in the elastically calculated extreme fiber bending stress in a simple rectangular beam due to creep is 33%. Assuming that a similar mechanism exists in bellows, but using only a 20% reduction in σ_B and a 50% safety factor on creep rupture, the second rule preventing bending rupture is:

$$\sigma_H + \frac{\sigma_B}{1.25} \leq \frac{2}{3} \sigma_{\text{rupt}} \quad \dots \text{Screening Rule (2b)}$$

To limit excessive creep deformations, two additional screening rules are developed using the safety factors and time-dependent material stress values defined in Subsection VI-B-1-a. The redistribution of pressure bending stress due to creep is again used, resulting in:

$$\sigma_H + \frac{\sigma_B}{1.25} \leq 0.8 \sigma_{3c} \quad \dots \text{Screening Rule (2c)}$$

and

$$\sigma_H + \frac{\sigma_B}{1.25} \leq \sigma_c \quad \dots \text{Screening Rule (2d)}$$

As in Subsection VI-B-1-a, a linear damage approach was actually used to apply Screening Rules (2b)-(2d) to the bellows designs.

In passing, it is noted that these screening rules were developed considering σ_B as having a high degree of significance in the mechanisms leading to excessive distortion or collapse of the convolution roots, or to rupture of the convolution crowns. As noted in the discussion of categorization of σ_B , there may be justification to eliminate or at least scale down its significance to these mechanisms, especially if reinforcing rings are employed. This was done in Subsection VI-B-3, where the screening rules were used on actual configurations, to provide a comparison of time-dependent Screening Rules (2b)-(2d). However, no additional screening rules are developed here to reflect this consideration.

c. Rule for Incremental Distortion of Convolutions

Due to the absence of a detailed elastic-plastic-creep analysis to which strain limits could be imposed, and the potential overly conservative nature of limiting elastically calculated stresses to the elastic regime, an alternative approach was sought to address incremental distortion. Analogous to the Section III low-temperature rules, adapting a limited stress range criteria based on S_0 appeared to be a reasonable compromise in light of the preconceptual nature of the Phase I effort. It must be realized, however, that due to creep effects, this approach is not theoretically sound, and is potentially unconservative. Some compensation for this is made in the following definition of the stress range used, particularly with regard to motion stresses (see Page 139).

The maximum stresses in the bellows element, excluding end concentrations, usually occur at the convolution roots and crowns and are in the meridional direction. During motion cycling about a neutral orientation, the meridional deflection bending field will change direction. In angulating joints, corresponding points on a convolution on opposite sides of the neutral axis will have bending fields of opposite signs. Most of the significant loading cycles consist of internal pressure cycling about a positive non-zero mean, coupled with motion cycling about a mean which can vary in sign. Determination of the maximum stress range for all loading conditions is difficult and was not attempted in the Phase I efforts. Instead, a worst case loading was assumed

which pressure cycled from zero to full operating pressure in concurrence with full range motion cycling about the neutral position. Additionally, the signs of the various stress fields were assumed to be multiaxially additive at the extreme ends of the cycle, with their maximum values occurring at the same convolution location. The result of this methodology is Screening Rule (3):

$$(\sigma_H + \sigma_B + \sigma_b)_{\text{Range}} \leq 3S_o \quad \dots \text{Screening Rule (3)}$$

d. Rule for Squirm

Squirm is lateral buckling of a bellows, due to the resultant axial force of the pressurized fluid in the adjacent piping, acting on the bellows much like an Euler column. The critical buckling pressures generated in the initial scoping effort are for a fixed-end bellows with no built-in deformations or flaws. A flawless, simple-ended bellows would theoretically have a buckling pressure of one-fourth of these values. Since the actual bellows will have end conditions somewhere between simple and fixed-end (but closer to fixed-end), a correction factor of 2 on allowable critical pressure was used. Additionally, to address the real possibility of significant stiffness loss due to flaws and initial deformations, an additional correction factor of 1.2 was introduced. Finally, based on ASME, Section III elevated-temperature design philosophy, a safety factor of 3 was introduced to address the creep buckling phenomenon. This results in:

$$P_c \geq (2)(1.2)(3)P = 7.2P \quad \dots \text{Screening Rule (4)}$$

e. Rule for Fatigue Failure

Since evaluation of fatigue damage requires several lengthy stages of calculation, it was not feasible to screen all geometric shapes which could pass the first four screening rules. Therefore, only 12 selected bellows were evaluated. Furthermore, elevated-temperature, hold-time fatigue curves do not exist for certain of the higher-strength materials considered in Phase I (see Section IV), and detailed quantitative evaluations had to be limited to Type 316 stainless steel (an approved Section III elevated-temperature

material for which such curves are available). However, enough data does exist for the higher-strength materials to make qualitative discussions of their expected behavior.

Two fatigue damage approaches were taken. The first ignores time-dependent effects and is presented in this subsection. The second addresses creep interaction effects and is presented in the next subsection.

The screening rules were developed based on a linear damage approach and a "bellows fatigue curve" which was generated by arbitrarily scaling down (by 25%) material fatigue curves from Code Case 1592 to reflect observed trends in base metal versus bellows fatigue data. The hold-time curves were used for Screening Rule (5) and the rapid cycle curves for Screening Rule (5). The load cycles were determined from the CRBR umbrella transient definitions (see Subsection II-C). Scaling factors were used to convert the computer-generated stresses to stress ranges for each cycle. All three computed stresses (σ_H , σ_B , and σ_b) were geometrically added, thus conservatively ignoring any possible benefits from multiaxial consideration of stress signs. Minimum elevated temperature modulus of elasticity values were used to convert to strain ranges. These elastically calculated strain ranges for each load cycle were used to determine the allowable number of cycles, N , from the bellows "fatigue curve." Damage fractions were linearly added and the total damage was restricted to unity. This resulted in:

$$\sum_{\text{cycles}} \frac{n}{N} \leq 1.0 \quad \dots \text{Screening Rule (5)}$$

where n is the number of normal design cycles for each cyclic loading.

f. Rule for Creep-Fatigue Failure

This screening rule is similar to Screening Rule (5) with the following exceptions:

- The "bellows fatigue curves" were scaled from the hold-time curves for Type 316 stainless steel from Code Case 1592
- The screening rule developed was:

$$\sum_{\text{cycles}} \frac{n}{N} \leq \beta - \alpha_j \quad \dots \text{Screening Rule (6)}$$

where β is induced from Rockwell experience with elevated-temperature elastic solutions, and α_j corresponds to creep damage fractions determined through two approaches ($j = 1$ and $j = 2$).

Two approaches were adopted to assess the creep damage during hold periods between cycles. These approaches follow from the two categorizations of σ_b as discussed in the introduction of Subsection VI-B-1. A sustained stress value during elevated-temperature hold periods was defined as the minimum of:

- σ_H plus one-half of the range of the secondary stresses during the cycle, or
- σ_{yp} at the hold temperature.

The selected minimum value is divided by 0.9 prior to entering the appropriate minimum-stress-to-rupture curve. This parallels the elevated-temperature, elastically calculated, hold-time damage calculation philosophy of Section III of the ASME Code. The allowable time-to-rupture at this stress level is divided into the hold time per cycle, t , and the result is multiplied by the number of such cycles occurring during the plant lifetime. This process is repeated for each cyclic loading for which significant elevated-temperature hold periods exist. This leads to two definitions of α_j , based on the categorization of σ_b . For σ_b considered as a peak stress, we obtain:

$$\alpha_1 = \sum_{\text{cycles}} \frac{t}{T_1}$$

where T_1 is determined from the minimum of:

$$\left[\sigma_H + \frac{(\sigma_H + B) \text{Range}}{2} \right] \div 0.9 \text{ OR } \sigma_{yp} \div 0.9$$

For σ_b considered as a secondary stress, we obtain:

$$\alpha_2 = \sum_{\text{cycles}} \frac{t}{T_2}$$

where T_2 is determined from the minimum of:

$$\left[\sigma_H + \frac{(\sigma_H + \sigma_B + \sigma_b) \text{Range}}{2} \right] \div 0.9 \text{ OR } \sigma_{yp} \div 0.9$$

2. Reduced Set of Design Geometries Based on Considerations of Time-Independent Strength Requirements

Several approaches were used to select the "optimum" configurations for each live length and thickness, based on Screening Rules (1a), (2a), and (3). The first provided selection of geometries regardless of fabricability by considering only minimization of the required allowable primary membrane stress, $S_{o,r}$. This was done by combining Screening Rules (1a), (2a), and (3):

$$\sigma_H \leq S_o \quad \dots \text{Screening Rule (1a)}$$

$$\sigma_H + \sigma_B \leq 1.5 S_o \quad \dots \text{Screening Rule (2a)}$$

$$\sigma_H + \sigma_B + \sigma_b \leq 3 S_o \quad \dots \text{Screening Rule (3)}$$

The required allowable primary membrane stress was usually dictated either by Rule (2a) or (3). Furthermore, as one proceeds through the stress data for each thickness, a trend is usually noticed in $S_{o,r}$ versus convolution height. As convolution height increases, σ_H and σ_B increase while σ_b decreases. Therefore, $S_{o,r}$ from Rule (2a) will increase with increasing convolution height. The required allowable primary membrane stress from Rule (3) will initially decrease with increasing convolution height, but will begin to increase when the pressure stresses become larger than the motion stresses. In most cases, the minimum $S_{o,r}$ to satisfy both Rules (2a) and (3) occurs when

$$\sigma_b \approx \sigma_H + \sigma_B \quad \dots \text{Selection Basis 1}$$

A further trend observed from the stress data was that increasing the number of convolutions caused $S_{o,r}$ to decrease. Therefore, the optimum configuration for each thickness was usually the one with the maximum number of fabricable convolutions. In many cases this number was as large as 50-60 convolutions with a pitch less than 0.5 in. The vendor surveys (see Section V) indicated that these pitch values would be extremely difficult to manufacture using present techniques. Therefore, geometries with pitch values of 0.75 to 2 in. were selected to indicate the effect of pitch on $S_{o,r}$. However, the thickness and convolution height were not limited in this selection, and vendors may not presently have the capabilities to form some geometries (see Selection Basis 3). Within a certain thickness and number of convolutions category, the geometry best satisfying the above selection basis was used. This second approach can be summarized as:

$$\left. \begin{array}{l} \sigma_b \approx \sigma_H + \sigma_B \\ 0.75 \text{ in.} \leq \text{pitch} \leq 2 \text{ in.} \end{array} \right\} \dots \text{Selection Basis 2}$$

A third "selection basis" further considered present fabrication capabilities (see Section V) by using a predetermined convolution geometry with respect to pitch, convolution height, and thickness. A trend noticed in applying Selection Basis 2 was that the convolution heights and thicknesses tended to be large for the selected designs. These types of geometries may be difficult to fabricate. Therefore, an "ideal" geometry was selected, and was combined with Selection Basis 1 to provide a definition of the minimum $S_{o,r}$, based on present vendor capabilities:

$$\left. \begin{array}{l} \sigma_b \approx \sigma_H + \sigma_B \\ \text{pitch} \approx 1.75 \text{ in.} \\ \text{maximum convolution height} = 3 \text{ in.} \\ \text{maximum thickness} = 0.125 \text{ in.} \end{array} \right\} \dots \text{Selection Basis 3}$$

Two other selection bases were only initially used to address total stress (fatigue):

$\sigma_B + \sigma_H + \sigma_b$ minimized ... Selection Basis 4a

$\sigma_B + \sigma_H + \sigma_b$ minimized with convolution height lowered by one incremental value from optimum (see Table VI-1) ... Selection Basis 4b

The second basis, 4b, was intended to weight the minimization towards $\sigma_B + \sigma_H$. This pair of approaches led to geometry selections nearly identical to those from Selection Basis 1. As a consequence, Selection Basis 1 was later adapted for all selections.

b. Application of Selection Bases and Choice of Representative Optimum Geometries

Table A-1 (Appendix A) provides a summary of the application of the five selection bases. This results in a reduction of considered geometries from ~33,000 to 373.

Several significant trends can be recognized by examination of the stresses of this reduced geometry set. Some of these have been previously mentioned:

- As live length increases, $S_{o,r}$ decreases.
- As live length increases, P_c usually decreases.
- As number of plies increases, $S_{o,r}$ usually decreases.
- As thickness increases, convolution height increases for similar $S_{o,r}$.
- As number of plies increases, P_c decreases.

A further reduction in the geometry set was made by selecting representative configurations which bracketed the five trends mentioned above. This was done as follows. For a particular live length and number of plies, the geometry from either Selection Basis 1, 4a, or 4b, which best minimizes $S_{o,r}$ was retained. Also, for the same live length and number of plies, the geometry from Selection Basis 3 which best minimizes $S_{o,r}$ was used. These two geometries would serve to represent the best convolution shape with regard to $S_{o,r}$ with and without restriction to commonly manufactured convolution shapes. Usually, the former

geometry had the maximum thickness and maximum number of convolutions. The latter geometry, as previously mentioned, was restricted to a pitch of approximately 1.75 in. and a maximum convolution height of 3 in. The optimum thickness for this geometry was near the maximum value allowed of 0.12 in. A thin-walled bellows, with small convolution height whose geometry required only a minimal increase in $S_{o,r}$ could exist and not be found by these two selection methods. Towards this end, a third geometry was selected, when possible, to demonstrate the effect of thickness on the optimized convolution height and $S_{o,r}$.

Table A-2 (Appendix A) provides a summary of this selection process. The stability Screening Rule (4) was also applied for this selection process, and failure to meet this rule is indicated in the table. This results in a reduction of the screened geometries from 373 to 56.

It is interesting to note that all 42-in. axial motion bellows had $S_{o,r}$ values greater than those for 42-in. angulating bellows. Also, none of the axial motion bellows passed Screening Rule (4), which indicates that creep buckling could be a problem with these designs. The 56-in. axial bellows versus angulated bellows showed similar trends. Therefore, the remainder of the evaluations were used to assess only the bellows elements for the articulating (angulating) flexible joint designs.

Figures VI-5 through VI-8 plot $S_{o,r}$ and P_c versus bellows live length for various one-, two-, and four-ply configurations. Figure VI-5 shows the 42-in. angulated bellows selected for minimum allowable primary membrane stress requirements with no limits on geometric shape. From this figure, several trends are observed:

- For single-ply bellows, the beneficial effect of increasing live length on $S_{o,r}$ begins to decrease beyond 45 in.
- The four-ply bellows had an $S_{o,r}$ approximately two-thirds that for single-ply bellows, but the advantage diminishes as live length is increased.
- The two-ply bellows data point gives an $S_{o,r}$ closer to the four-ply curve, indicating the advantage of multi-ply configurations begins to diminish as successive numbers of layers are used.

9112-51

Figure VI-5. Behavior Curves for 42 in. Angulating Bellows from Selection Bases 1, 4a, 4b

9112-52

Figure VI-6. Behavior Curves 42 in. Angulating Bellows from Section Basis 3

9112-53

Figure VI- 7. Behavior Curves for 56 in. Angulating Bellows
from Selection Bases 1, 4a, 4b

9112-54

Figure VI-8. Behavior Curves for 56 in. Angulating Belows from Selection Basis 3

- The single-ply bellows buckling pressure becomes less influenced by live length beyond 50 in.
- The multi-ply bellows are less stable than the single-ply bellows, and a 36-in. length is about the maximum allowable for four-ply configurations (for geometries investigated).

Three bellows geometries, (A), (B) and (C), (see Figure VI-5) were selected for application of the remaining screening rules and comparison to selected material yield points.

Figure VI-6 represents the behavior of 42-in. angulated joint bellows selected for minimum allowable primary membrane stress requirements with geometry limits near those of present vendor capabilities. From this figure, several trends can be observed:

- For single-ply bellows, the beneficial effect of increasing live length on $S_{o,r}$ begins to decrease beyond 45 in.
- The four-ply bellows appears advantageous only for very short live lengths. This results from the limitation on pitch which does not allow multi-ply bellows to reduce the stress levels by the normal practice of packing more convolutions into a given length. The same trend would appear true for the two-ply bellows, judging from the single data point location.
- Buckling instability increases fairly uniformly with increasing live length for both ply configurations.
- The four-ply bellows are less stable than the single-ply bellows; they have maximum allowable live lengths of ~ 36 -in. and ~ 70 -in. respectively.

Three bellows geometries, (D), (E) and (F), (see Figure VI-6) were selected for application of the remaining screening rules and comparison to selected material yield points.

Figure VI-7 represents the behavior of 56-in. angulated joint bellows selected for minimum allowable primary membrane stress requirements with no limit on geometric shape. Several trends can be noted:

- The beneficial effect of increasing live length on $S_{o,r}$ decreases with increasing length, and has a similar effect on all ply configurations.
- The four-ply configurations require from 50 to 70% of the $S_{o,r}$ values for single-ply configurations; the two-ply nearly halfway between the single- and four-ply.
- Buckling pressures are greatly influenced by thickness, and the plotted P_c data is thus scattered. All geometries plotted passed Screening Rule 4.
- Overall, the resulting $S_{o,r}$'s are much lower than for the 42-in. joints, due in large part to the reduced pressure loading.

Three bellows geometries, ㉔, ㉕ and ㉖, (see Figure VI-7) were selected for application of the remaining screening rules and for comparison to selected material yield points.

Figure VI-8 represents the behavior of 56-in. angulating joint bellows selected for minimum allowable primary membrane stress requirements with geometry limits near those of present vendor capabilities. The trends are similar to those of Figure VI-7, except that the overall $S_{o,r}$ values are larger, and buckling data is better behaved due to thickness limitations. Two more geometries, indicated by points ㉗ and ㉘, were selected for further evaluation. Geometry ㉙ was not treated in the bellows design computer program, but was selected as a two-ply bellows for further evaluation in Phase II of the development program. Geometry ㉙ will thus not be considered in further evaluations in this report.

This final screening has reduced the number of geometries to be further considered from 56 to 11. Table VI-2 lists these geometries along with their behavior characteristics.

c. Evaluation of Candidate Materials for Selected Representative Optimum Geometries

In Section VIII, Appendix Q, HA-600 of the ASME B&PV Code, Code Case 1592, and related Code Addenda, maximum allowable stresses at 1050°F are given (or can be extrapolated) for the four materials considered in the Phase I

TABLE VI-2
SELECTED PRELIMINARY PRECONCEPTUAL UNREINFORCED
SINGLE BELLOWS GEOMETRIES

ID from Figure VI-5 to VI-8	Joint Diam- eter	Selection Basis from Table A-1	L	N _p	T	N _c	Pitch	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	K _{ang}	W	P _c	S _{o, r}
(A)	42	4b	36	1	0.18	20	1.8	2.5	7.9	26.2	58.8	120,000	912	4290	19.6
(B)	42	1	36	2	0.12	25	1.4	2.2	5.4	21.4	45.1	84,700	>1000	3000	15.0
(C)	42	4a	36	4	0.045	50	0.7	1.0	7.5	19.2	41.4	42,500	912	1690	13.8
(D)	42	3	45	1	0.12	26	1.7	1.5	16.6	31.4	69.6	100,000	522	2950	23.2
(E)	42	3	36	2	0.06	20	1.8	1.0	22.2	35.4	78.4	92,150	305	3500	26.1
(F)	42	3	28	4	0.03	18	1.6	0.8	23.2	40.1	76.3	51,500	230	2500	26.7
(G)	56	1	10	1	0.05	20	0.5	2.25	1.0	16.2	34.8	9860	260	147	11.6
(H)	56	1	10	2	0.035	20	0.5	2.0	0.8	13.0	29.1	9360	330	148	9.7
(I)	56	1	10	4	0.03	20	0.5	2.25	0.4	10.9	22.0	8550	630	353	7.3
(J)	56	3	20	1	0.09	12	1.7	3.0	1.3	9.6	34.9	55,100	410	240	11.6
(K)	56	3	20	2	≤0.06	12	1.7	≤3.0	—	—	—	—	—	—	—
(L)	56	3	20	4	0.03	13	1.5	2.0	1.2	9.5	24.9	14,200	420	140	8.3

*See Figure VI-1 for definition of nomenclature

effort (see Section IV of this document). Note, the maximum allowable stress used for Inconel 718 is based on its use as a bolting material. Although this value was used in the evaluation, it is expected to be lower than the S_o value to be assigned it, if and when it becomes accepted as a pressure vessel material. Table VI-3 summarizes these S_o data.

TABLE VI-3
MATERIAL MAXIMUM ALLOWABLE STRESS VALUES

Material	S_o
Type 316 Stainless Steel	14.5
Hastelloy X	14.5
Hastelloy C	24.0
Inconel 718	38.8

Comparison of the $S_{o,r}$ values of Table VI-2 with the inferred material S_o values of Table VI-3 demonstrates which of the 4 materials are applicable to each of the 11 bellows. The results of this comparison appear in Table VI-4, where the percentage of material strength in excess of the required value is shown.

With respect to material selection, the following conclusions can be made:

- Use of Inconel 718 will easily satisfy the strength (yield and ultimate) requirements of all selected bellows geometries.
- Use of Hastelloy C will easily satisfy the strength requirements of all selected 56-in. joint geometries. It will adequately satisfy the strength requirements of those selected 42-in. joint geometries which will most likely require a fabrication development effort (A, B, and C). It will marginally satisfy the strength requirements of only one of the three 42-in. joint geometries selected near the limits of current fabrication techniques (D).

TABLE VI-4

SELECTED GEOMETRY EVALUATION FOR TIME-INDEPENDENT
SCREENING RULES (1)-(4) USING THE
FOUR PHASE I MATERIALS*

Joint Diameter (in.)	Geometry ID	N _p	Selection Basis	% Additional Strength Available			
				Type 316 Stainless Steel	Hastelloy X	Hastelloy C	Inconel 718
42	Ⓐ	1	4b	NA [†]	NA	23.7	100
	Ⓑ	2	1	-3.4§	-3.4§	61.6	161
	Ⓒ	4	4a	6.1	6.1	75.6	184
	Ⓓ	1	3	NA	NA	4.5	68.9
	Ⓔ	2	3	NA	NA	NA	49.9
	Ⓕ	4	3	NA	NA	NA	44.4
56	Ⓖ	1	1	26.3	26.3	109	240
	Ⓗ	2	1	50.5	50.5	149	303
	Ⓘ	4	1	100	100	230	434
	Ⓙ	1	3	26.3	26.3	109	240
	Ⓛ	4	3	75.6	75.6	190	370

*Nomenclature defined in Figure VI-1.

[†]NA = not acceptable.

§Marginal in light of inaccuracies in stress analysis, therefore will further evaluate this geometry.

- Use of Hastelloy X or Type 316 Stainless Steel will adequately satisfy the strength requirements of all selected 56-in. joint geometries. They might marginally satisfy the strength requirements of two of those selected 42-in. joint geometries which will most likely require a fabrication development effort (B and C). They cannot be used for any of the 42-in. joint geometries selected near the limits of current fabrication techniques.

3. Fatigue and Elevated-Temperature Evaluations

The 11 representative designs selected in Section VI-B-2 were next evaluated for time-dependent (elevated temperature) and fatigue failures. Since only one of the flexible joint materials, Type 316 stainless steel, (see Section IV) is completely defined in the elevated-temperature regime, quantitative evaluation results could be made only for this material. The evaluations for the alternate materials are qualitative in nature.

a. Application of Time-Dependent Screening Rules (1b), (1c), (1d), (2b), (2c), (2d)

In Subsection VI-B-2, seven geometries were selected which could utilize Type 316 stainless steel based on Screening Rules (1a), (2a), (3), and (4) (see Table VI-4). Calculation of usage fractions for these geometries are given in Table B-1 (Appendix B). The sustained loading periods which were used are presented in Table B-2. These were defined by assuming that the 80% and 100% power modes existed for 90% of the 30-year design life. Accumulated sustained hold times at each condition were assumed to be equally distributed between the two operating modes. The sustained pressure stresses were approximated by appropriate linear scaling (Table B-3) of the full pressure stress values generated in the computer solutions.

The allowable values of σ_{rupt} , σ_{3c} , and σ_c for Type 316 stainless steel were assumed equivalent to the S_t values from ASME B&PV Code Case 1592 (at the required temperature). A plot of S_t versus temperature is shown in Figure VI-9. This assumption is conservative because S_t is defined as the minimum of these three properties. This allows treatment of each triplet of screening rules (i.e., 1b, c, d or 2b, c, d) by comparison of the usage fraction (linear damage fraction) generated using S_t , rather than each of σ_{rupt} , σ_{3c} , and σ_c .

Figure VI-9. S_t - Type 316 SS

The results of such treatment are presented in Table VI-5. From these results it appears that none of the 42-in. geometries considered have load-controlled stresses low enough to satisfy the imposed limits on $\sigma_H + \sigma_B$ for Type 316 stainless steel. However, if σ_B is considered a secondary stress, as discussed at the beginning of Subsection VI-B-1, geometries (A), (B) and (C) would be adequately designed with respect to time-dependent, load-controlled stresses.

All of the stainless steel 56-in. geometries except (G) will be adequate to resist the time-dependent, load-controlled failure modes considered. If σ_B is considered a secondary stress, geometry (G) would also be acceptable.

The qualitative discussions for the alternate alloys are presented in the following paragraphs.

The Larson-Miller parameter (P) versus stress for rupture curves (Figure IV-3), gives average rupture stresses for the four materials considered in Phase I at 1050°F and 10^6 hr ($P = 46.8$). Minimum rupture stresses at 1050°F and 10^6 hr are available for Type 316 stainless steel from Code Case 1592 and for Inconel 718 from AI estimated data. These average and minimum values are presented in Table VI-6.

TABLE VI-5
USAGE FRACTIONS FOR TIME-DEPENDENT, LOAD-CONTROLLED
STRESSES IN SELECTED TYPE 316 SS BELLOWS

Joint Diameter (in.)	Geometry ID	$\sigma_H \leq (\sigma_{rupt}, \sigma_{3c}, \sigma_c)^*$	$\sigma_H + \frac{\sigma_B}{1.25} \leq (\sigma_{rupt}, \sigma_{3c}, \sigma_c)^\dagger$
42	Ⓐ	0	$>10^5$
42	Ⓑ	0	295
42	Ⓒ	0	30
42	Ⓓ	24	$>10^5$
42	Ⓔ	$>10^5$	$>10^5$
42	Ⓕ	$>10^5$	$>10^5$
56	Ⓖ	0	1.7
56	Ⓗ	0	0.4
56	Ⓘ	0	0.1
56	Ⓙ	0	0
56	Ⓛ	0	0

* σ_B considered secondary

† σ_B considered primary

TABLE VI-6
COMPARISON OF FLEXIBLE JOINT MATERIAL CREEP
RUPTURE STRENGTHS

Material	Rupture Stress at 1050°F and 10 ⁶ hr (psi)	
	Average	Minimum
Type 316 Stainless Steel	17,000	10,000
Inconel 718	76,000	62,000
Hastelloy C	46,000	not available
Hastelloy X	30,000	not available

The AI estimated data for Inconel 718 indicated that at 1050°F, the minimum value of the three time-dependent allowable stresses (σ_{rupt} , σ_{3c} , and σ_c) would be σ_{rupt} . This was also true of the Type 316 material data used to prepare Code Case 1592, and is expected to be true of the Hastelloy C and Hastelloy X materials data. Because S_t is thus probably directly dependent on σ_{rupt} , it was also informative to examine the various creep rupture curves with respect to one of the 42-in. designs that exhibits lower load controlled stresses. Geometry (B) was selected. Considering σ_B as primary resulted in the following significant sustained loading conditions (see Tables B-1 - B-3): 18.2 ksi for 1.18×10^5 hr at 1050°F and 11.7 ksi for 1.18×10^5 hr at 990°F. Therefore, determination (or extrapolation) of the Larson-Miller parameter, P, (from Figure IV-3) at each of these stress levels (times 1.5 to reflect a $2/3 \sigma_{rupt}$ rule) can give the average time to rupture at the given temperature. This was done for both Inconel 718 and Type 316 stainless with the following results:

TABLE VI-7
COMPARISON OF TIME-DEPENDENT RUPTURE CHARACTERISTICS

Material	Larson-Miller Parameter, P		Average Time to Rupture (hr)	
	At 27.3 ksi	At 17.6 ksi	At 1050°F Condition	At 990°F Condition
Type 316 Stainless Steel	43.8	46.5	1.0×10^4	1.1×10^7
Inconel 718	52.5 (extrapolated)	55 (extrapolated)	5.7×10^9	8.5×10^{12}

If we go one step further and consider the average time to rupture in the same manner as an allowable time to rupture, we would get the damage fractions of 11.8 and 0.01 for Type 316 stainless and Inconel 718, respectively. This indicates the considerable amount of creep performance improvement to be expected by use of Inconel 718. A similar but less dramatic improvement is to be expected from use of either of the Hastelloys.

Therefore, based on Table VI-5 and the above material trends, it would appear that for the selected geometries:

- All of the 56-in. bellows would probably be easily satisfactory with respect to load-controlled, time-dependent stress failure modes using Inconel 718, Hastelloy C, or Hastelloy X. This would be independent of the categorization of σ_B .
- The 42-in. geometries would have improved strength with respect to load-controlled, time-dependent stress failure modes using Inconel 718, Hastelloy C, or Hastelloy X, rather than Type 316 stainless steel. This benefit can only be qualitatively assessed at this time, but there is good indication that most, if not all, of the selected 42-in. geometries could be adequately designed for time-dependent, load-controlled stress failure modes (independent of categorization of σ_B).

b. Application of Fatigue Screening Rule (5)

Seven of the geometries can use Type 316 stainless steel and pass Screening Rules (1) through (4) if σ_B is categorized as a secondary stress. These are now evaluated with respect to fatigue failure, ignoring hold-time effects. The method used was discussed in Subsection II-B-1-e. The loading cycles and adjusted bellows fatigue curves are presented in Table B-4 and Figure VI-10, respectively. Damage fraction computations are given in Table B-5. A summary is presented below in Table VI-8.

The results imply that if the bellows fatigue data is in fact representative of actual bellows fatigue tests, and if creep effects are insignificant, then all seven geometries would meet the design life requirement. However, the effect of creep damage on fatigue life may be significant, as indicated in the next subsection.

165
N112T1310002

Figure VI-10. Type 316 Stainless Steel Bellows "Fatigue Curve"

9112-56

TABLE VI-8
NO-HOLD TIME FATIGUE DAMAGE FRACTIONS FOR SELECTED
TYPE 316 STAINLESS STEEL BELLOWS

Joint Diameter (in.)	Geometry ID	No. of Plies	Selection Basis	Damage Fraction $\Sigma \frac{n}{N}$
42	Ⓑ	2	1	0.273
42	Ⓒ	4	4a	0.161
56	Ⓖ	1	1	0.003
56	Ⓗ	2	1	0.001
56	Ⓘ	4	1	~ 0
56	⓵	1	3	0.012
56	Ⓛ	4	3	~ 0

c. Application of Screening Rule (6)

The seven selected geometries utilizing Type 316 stainless steel were evaluated with respect to creep-fatigue failure. The method used is presented in Subsection VI-B-1-f. The loading cycles and fatigue curves are again given in Table B-4 and Figure VI-10, respectively. The hold-time fatigue damage fraction computations utilizing the strain ranges of Table B-5 are presented in Table B-6. A summary of these damage fractions is presented in Table VI-9. Interpretation of these damage fractions should be combined with a perspective on the amount of creep damage to be sustained. A discussion will follow the creep damage fraction (α_j) results.

The two approaches taken to evaluate the creep damage between cycles are discussed in Subsection VI-B-1-f. The sustained hold periods were defined by assuming that the 80% and 100% power modes (see Subsection II-C) existed for 90% of the 30-year design life. Sustained hold times at each condition were assumed to be equally distributed between the two operating modes (a uniform cycling exists). This led to the hold-period definitions given in Table B-2 of Appendix B.

The sustained pressure stresses were approximated by appropriate linear scaling of the computed full pressure stress values. The sustained motion

TABLE VI-9
HOLD-TIME FATIGUE DAMAGE FRACTIONS FOR SELECTED
TYPE 316 STAINLESS STEEL BELLOWS

Joint Diameter (in.)	Geometry ID	Number of Plies	Selection Basis	Damage Fraction $\left(\sum \frac{n}{N}\right)$
42	Ⓑ	2	1	33.7
42	Ⓒ	4	4a	21.9
56	Ⓒ	1	1	0.43
56	Ⓓ	2	1	0.18
56	Ⓘ	4	1	0.02
56	Ⓢ	1	3	1.20
56	Ⓛ	4	3	0.13

stresses, when used, were approximated by linearly scaling the computed motion stress values. The scale factor was the ratio of the difference between sustained temperature and mid-range temperature divided by one-half of the full temperature range. This resulted in the scale factors presented in Table B-3. The range of pressure and motion stresses during the defined cycles was also used in the creep damage evaluation, and the corresponding scale factors are given in Table B-3.

The creep damage calculation and creep-rupture curve are presented in Table B-7 and Figure VI-11, respectively.

The results of this evaluation indicate that all geometries, except Ⓒ, have creep damage fractions (α_j) which are near zero for σ_b categorized as a secondary stress. For geometry Ⓒ, the damage fraction is 0.12, which is negligible when compared to its damage fraction in fatigue (21.9).

Examining Table VI-9, and considering creep damage during hold periods (between cycles) negligible, several conclusions on the use of Type 316 stainless steel for these seven geometries were made. First, all 56-in. joint bellows appear quite feasible with respect to fatigue life, if built with Type 316 material. More accurate analyses and criteria development during Phase II of the joint development program should bear this out. Opposingly, the 42-in. joint bellows

Figure VI-11. Stress to Rupture Curve, Type 316 Stainless Steel

have considerable fatigue damage fractions, even in light of the conservatism in the hold-time fatigue curve used. Furthermore, creep damage due to peak stresses; neglect of several stress components, and general inaccuracies in using the explicit stress equations, may lead to the conclusion that these selected 42-in. geometries would probably be unable to perform satisfactorily under the present design conditions using Type 316 stainless steel. However, several alternatives are available to allow use of Type 316 stainless steel for all the primary loop joints. Both the pressure and temperature loading conditions could be changed by certain system modifications such as use of a cold leg instead of a hot leg pump. Piping loop length could be increased between joints to relieve some of the displacement requirements. Reinforcing rings could be used to lower the pressure stresses. Double-series bellows articulating joints could be used to reduce the motion requirement per bellows. A full examination of all these alternatives was not possible in the Phase I efforts, however, use of reinforcing rings and double-series angulating geometries was briefly examined. The results of these examinations are presented in the Subsection VI-C.

Examining the elevated temperature fatigue curves for the various Phase I alternate materials (Inconel 718, Hastelloy C, and Hastelloy X) indicates that some of these materials can provide improved fatigue lives for the 42-in. joint bellows. The largest damage fractions were accumulated during loading cycle 5

(see Table B-4) which occurs approximately 12,000 times during life. A comparison of allowable strain ranges for the various materials at this number of cycles was made. The material fatigue curves are presented in Figures IV-7 and IV-8, and the strain ranges (reduced by 25%) are shown in Table VI-10.

TABLE VI-10
COMPARISON OF ALLOWABLE STRAIN RANGES AT
1200°F and 12,000 CYCLES (No Hold Time)

Material	Zero Mean Stress	Maximum Mean Stress Correction
Type 316 SS	1.40×10^{-3}	1.35×10^{-3}
Inconel 718	3.90×10^{-3}	0.90×10^{-3}
Hastelloy X, C	2.20×10^{-3}	2.20×10^{-3}

Note that the mean stress correction for the Inconel 718 is more severe than that for Type 316 stainless steel (see discussion in Section IV). Fatigue performance at elevated temperatures could be better compared if hold-time fatigue curves were available. In light of the creep rupture characteristics shown in Figure IV-4 or Table VI-6, it is reasonable to expect a decrease in stress relaxation for a particular loading situation when one of the higher-strength materials is used instead of Type 316 stainless steel. It can then be anticipated that the hold time fatigue curves for the higher-strength materials will show more advantage over Type 316 stainless steel for zero mean stress cycling. In addition, the effect of hold times on the Inconel 718 fatigue strength is expected to be negligible at 1050°F (see Section IV) compared to a 57% reduction in allowable strain range at 1050°F and 12,000 cycles for Type 316 stainless steel (see Figure VI-10). It is therefore instructive to compare the allowable strain ranges of Inconel 718, using no hold data, to Type 316 stainless steel using hold time data, (at 1050°F and 12,000 cycles with maximum mean stress corrections). The allowable strain ranges are 0.90×10^{-3} for the Inconel 718 and 0.7×10^{-3} for Type 316 stainless steel. This increase in allowable strain range would have a disproportionately greater effect on fatigue damage evaluations due to the inherent characteristics of the

log-log fatigue plots. For example, the damage fraction for the 5th cyclic loading (see Tables B-5 and B-6) for geometry (B) using Type 316 stainless steel was 27.9. If the Inconel 718, 1200°F no hold time curve with mean stress correction (Figure IV-9) was used (reduced by 25%), this damage fraction would be reduced to ~3.0. Therefore, this indicates a good probability that the high penalty paid for mean stress correction in the higher strength alloys may be more than offset by the low creep-fatigue interaction. In conclusion, use of one of the higher strength alloys like Inconel 718, Hastelloy C or Hastelloy X, should allow all the primary loop locations to incorporate expansion joints of the selected geometries, under the present design loadings, with a good assurance of adequate fatigue life.

C. CONSIDERATION OF DOUBLE-SERIES ARTICULATING JOINT BELLOWS ELEMENTS

To further evaluate the use of Type 316 stainless steel in the 42-in. articulating joints, three additional bellows geometries were investigated during Phase I. Each was used in a double-series articulating joint, and thus had reduced motion requirements compared to the single-bellows joint (see Subsection II-C). The individual bellows geometries are:

Bellows (BB): Double-series articulating joint bellows (see Section II). This is geometry (B) with reduced angulation requirements. Bellows (B) stress levels were used with appropriate scaling of the motion stresses for the reduced motion requirements.

Bellows (CC): Similar to (BB) except uses geometry (C).

Bellows (RR): Root-reinforced double-series articulating joint bellows using six 0.05-in.-thick plies, convolution height is 2.0 in., pitch is 2.0 in., and live length is 45 in.

Since it was not possible within the scope of Phase I to perform a parametric study of double-series articulating and/or reinforced bellows, these geometries cannot be considered optimized. They serve to indicate the relative potential for improvement offered by alternate geometric configurations. Screening rule evaluations for these geometries are given in Tables C-1 through C-5 of Appendix C. Many of the damage fractions could be used or deduced from geometry

Ⓑ and Ⓒ results. Table VI-11 summarizes the results of these studies. The results for geometries Ⓑ and Ⓒ have been included for comparison.

It appears that if σ_B is classified as a secondary rather than primary stress, geometry ⒸⒸ will have a reasonable chance of being feasible with the use of Type 316 stainless steel. The damage fraction value of 3.8 using the no-hold time fatigue curve indicates that an elastic-plastic-creep analysis would have a reasonable chance of demonstrating adequate fatigue life.

Bellows ⒿⒿ passes all screening rules examined. Based on this, it would appear that a Type 316 stainless steel double-series articulating joint bellows will have a good chance of success in meeting any future Class 1 expansion joint ruling for elevated-temperature use. Therefore, using double-series joints, additional plies, and/or reinforcing rings can significantly increase the feasibility of successfully using Type 316 stainless steel for all primary loop expansion joints.

However, the fabrication of some of these particular bellows geometries is beyond the present capabilities of the expansion joint manufacturers so far identified. In addition, other failure mode considerations, improved analytical techniques, or Code rules development may indicate that a less favorable situation exists for Type 316 stainless steel than is presently viewed (from a performance standpoint). Therefore, it is prudent to continue developing alternate material data and design options in order to further optimize large-scale breeder reactor bellows design.

D. FEASIBILITY SUMMARY OF PRECONCEPTUAL BELLOWS DESIGNS

1. Pressure-Balanced Axial Motion Joints

In comparison to articulating joints, the motion stresses in the axial joints are much higher for bellows of similar convolution geometries. This is because the axial designs must absorb four to five times the equivalent axial motions of the articulating designs. In order to decrease the motion stresses so that the time-independent screening rules can be met, the length of the axial bellows must be increased into the region where creep buckling may be a problem. Although external guides or supports may possibly be incorporated to minimize or eliminate squirm, the design would be considerably complicated by incor-

TABLE VI-11

SUMMARY OF DOUBLE-SERIES ARTICULATING JOINT BELLOWS EVALUATION

Bellows ID	Screening Rule*								
	(1b, 2a, 3)	(1b, 1c, 1d)	(2b, 2c, 2d)	($\overline{2b}$, $\overline{2c}$, $\overline{2d}$)	(4)	(5)	(6f)	(6c)	($\overline{6c}$)
Ⓑ	15,000	0	295	0	417	0.27	33.7	0	0
Ⓒ	13,800	0	30	0	235	0.16	21.9	0.12	0
ⒷⒷ	14,300	0	295	0	417	0.06	9.3	0	0
ⒸⒸ	12,800	0	30	0	235	0.03	3.8	0	0
ⒶⒶ	7,700	0	1.0	0	1957	0	0.8	0	0
Type 316 SS allowable or Screening Rule allowable	<14,500	<1.0	<1.0	<1.0	>190	<1.0	< $\beta-a$	a_1	a_2

*(1a, 2a, 3) : S_o, r from $\sigma_H \leq S_o$, $\sigma_H + \sigma_B \leq 1.5 S_o$, $\sigma_H + \sigma_B + \sigma_b \leq 3 S_o$

(1b, 1c, 1d) : $\Sigma(t/T)$ from use of 316 SS S_t for $\sigma_H \leq 2/3 \sigma_{rupt}$, $\sigma_H \leq 0.8 \sigma_{3c}$, $\sigma_H \leq \sigma_c$

(2b, 2c, 2d) : $\Sigma(t/T)$ from use of 316 SS S_t for $\sigma_H + \frac{\sigma_B}{1.25} \leq \frac{2}{3} \sigma_{rupt}$, $\sigma_H + \frac{\sigma_B}{1.25} \leq 0.8 \sigma_{3c}$,

$\sigma_H + \frac{\sigma_B}{1.25} \leq \sigma_c$ with σ_B considered primary

($\overline{2b}$, $\overline{2c}$, $\overline{2d}$) : Same as (2b, 2c, 2d) with σ_B considered secondary

(4) : $P_{allowable}$ from $P_c \geq 7.2P$ $\therefore P_{allowable} = P_c/7.2$

(5) : $\Sigma(n/N)$ from no hold time fatigue curves

(6f) : $\Sigma(n/N)$ from hold time fatigue curves

(6c) : $\Sigma(t/T)$ considering σ_b secondary

($\overline{6c}$) : $\Sigma(t/T)$ considering σ_b peak.

porating both this and the pressure-balancing capability. These disadvantages are sufficient to eliminate any further consideration of this type of joint based on present system requirements.

2. Articulating Joints

Completely different performance characteristics exist for the 42-in. hot leg joints and the 56-in. hot leg joints. In addition, although the 42-in. cold leg joints were not specifically analyzed in the Phase I efforts, their response behavior can be inferred from the 42-in. hot leg joints. Therefore, each of these three joints is discussed separately.

a. 56-in. Articulating Joints

These expansion joints would be located in the primary loop between the reactor and hot leg pump, and would, therefore, be subjected to very low pressure loadings. This allowed selection of designs with thinner material and, therefore, reduced motion stresses. The Phase I analysis efforts indicate that bellows with optimum fabrication shapes using a Code-approved, elevated-temperature material appear feasible. This bellows could be made with short live lengths ($\sim 1\text{-}1/2$ ft) and a small number of convolutions (6-10). Use of a higher-strength material could further reduce these parameters.

b. 42-in. Articulating Joints

These expansion joints would be located in the primary loop between a hot leg pump and the IHX, and between the IHX and reactor. The former location is at an elevated temperature while the latter location is at a temperature below the creep range for all the Phase I bellows materials. Both locations operate at a much higher pressure than that for the 56-in. joints. The feasibility of expansion joints will first be addressed for the hot leg location, followed by inferences about the feasibility of expansion joints in the cold leg.

(1) Hot Leg

Based on current system requirements, the use of an ASME Code-approved elevated-temperature material for the hot leg 42-in. joint bellows is marginally feasible. Moreover, it will require significant development efforts in both the fabrication and analytic areas. The design would probably require a relatively thick multi-ply bellows with a high live length (~ 3 ft) incorporated in a double-

series articulating joint configuration. Root ring reinforcement of the bellows may also be required.

The design difficulties stem from potential fatigue damage at elevated-temperature and possible creep rupture under sustained pressure loadings. Use of higher-strength materials will improve the possibility of success using single bellows articulating joint designs. However, some fabrication development may still be required.

(2) Cold Leg

With a maximum normal operating temperature of 750°F, creep effects will be negligible. Referring to only the time-independent rules (and reducing the safety factor on buckling), the Phase I results indicate that many materials (low-temperature Code rules would govern) are available which can be used to fabricate a feasible single bellows articulating joint. It is important to note that this fact has significant impact on the selection of a hot leg or cold leg pump, because the latter selection eliminates the high-pressure, high-temperature location in the primary loop (see Subsection II-C-4). In addition, use of higher strength materials would allow convolution shapes and overall size to be more in line with present vendor capabilities.

VII. PRECONCEPTUAL FLEXIBLE JOINT DESIGNS

The preconceptual engineering and design effort was initiated during Phase I of the Flexible Piping Joint Development Program. The purpose of the preconceptual effort was to provide a diverse joint design base from which several "preferred" conceptual designs could be selected. Four such pre-conceptual level designs were generated during the Phase I studies, and have served to illustrate several joint design features and options. Subsection VII-A discusses some of the general considerations on which these features were based. Subsection VII-B summarizes the four preconceptual designs.

A. SCOPE OF PRECONCEPTUAL JOINT DESIGN EFFORT

The preconceptual investigations evaluated five major areas of joint design based on selection of the articulating flexible joint as the preferred type for commercial LMFBR primary loop applications (see Section II). These five areas were:

- Bellows design
- Articulation unit design
- Flow liner design
- Redundancy - leak detection implementation
- Motion limiter design.

The general design features and considerations in each of these areas are discussed in the following subsections.

1. Bellows Design

The bellows design evaluations have been detailed in Section VI of this document. These evaluations resulted in a matrix of 56 bellows geometries. Four materials were considered for 11 of these geometries: Type 316 stainless steel, Inconel 718, Hastelloy X, and Hastelloy C. The preconceptual joint designs enveloped feasible material - geometry combinations, and employment of specific bellows designs in specific joints was not attempted. The four pre-conceptual joint layouts generated during the Phase I studies thus serve to illustrate design features over the bellows geometry range.

2. Articulation Unit Design

The hinge and gimbal articulation units were preconceptually sized, based on the pressure-separating forces generated in the joint. Simple strength of material and beam formulae were employed to approximate the stresses induced in critical areas of the units. In particular, beam equations were used to size the linkage arms and gimbal rings; bearing design equations and preliminary lubricant properties were used to size the pin lubrication area; and strength of materials was used to size the pin cross-sectional area. These efforts were only intended to provide the designers with relative size information so that the overall joint dimensions could be reasonably enveloped.

As indicated in the discussion of Subsection II-D-6, the high temperature bearing environment will require special lubricants to provide maintenance-free 30-year operation. One of the four articulation unit designs thus employs a "cooled bearing" assembly, which would operate at reduced temperatures and would allow the use of more common, currently available lubricants. The design of this assembly is preconceptual, and only serves to illustrate the special features (fins and thermal shielding) associated with such a joint.

3. Flow Liner Design

All preconceptual joint designs incorporate integral flow liners. These are the cylindrical-hemispherical type. Similar liners have been employed in flexible joints designed in the aerospace industry, and have been used to accommodate angulation requirements up to five times greater than those for the LMFBF joints. The liners are provided with through-holes for pressure relief, and with drainage paths for the liner-to-bellows annulus.

4. Redundancy - Leak Detection Implementation

Redundant joint design was discussed in Subsection II-D-2. One of the four preconceptual joint designs employs a primary and secondary bellows arrangement to provide "true redundancy" in the event of a bellows failure. The incorporation of the additional concentric bellows impacts the assembly sequence and weld inspectability of the design. These items were preliminarily addressed in the preconceptual effort, and the redundant joint design represents a reasonable incorporation of these fabrication considerations.

Note that even a redundant design does not provide a safety function unless it is monitored for failure of each bellows (see Subsection II-D-3). The annular section between bellows in the redundant joint design is provided with a monitoring port for the leak detection apparatus. In addition, if the bellows used in any of the joints were multi-ply designs, a monitoring function could be provided between plies (with the limitations discussed in Subsection II-D-3). This leak detection capability for multi-ply bellows can provide first-level failure monitoring for single bellows and redundant bellows designs.

5. Motion Limiter Design

Motion limiters are employed in all preconceptual flexible joints. The hinged joints employ limiter rods which restrict the total extension at the peak motion point of the joint. Gimbale joint motion limitation is provided by controlled clearances between the gimbal ring and bearing linkage arms. All preconceptual motion limiters were designed to allow $\pm 2.0^\circ$ of angulation from the "straight" position.

B. SUMMARY OF PRECONCEPTUAL JOINT DESIGNS

Preliminary layout drawings for the preconceptual primary loop flexible piping joints are presented in Figures VII-1 through VII-4. Hinged and gimbale joints were generated for each joint size (42-in. and 56-in.), in order to scope the design envelope for articulating joints. Table VII-1 summarizes the features of these four designs. Note that all design features for the various joints are interchangeable, and the selected features for any design are merely representative.

TABLE VII-1
SUMMARY OF PRECONCEPTUAL JOINT DESIGNS

Size, Pipe OD (in.)	Articulation Unit Type	Bearing Design	Flow Liner	Redundancy	Motion Limiter Design	Presented in Figure
56	Hinge	Cooled	Integral Cylindrical - Hemispherical	Multi-ply Monitoring Possible	Rod	VII-1
56	Gimbal	Uncooled	Integral Cylindrical - Hemispherical	Multi-ply Monitoring Possible	Integral to Gimbal	VII-2
42	Hinge	Uncooled	Integral Cylindrical - Hemispherical	Multi-ply Monitoring Possible	Rod	VII-3
42	Gimbal	Uncooled	Integral Cylindrical - Hemispherical	Concentric Redundant Used and Multi-ply Monitoring Possible	Integral to Gimbal	VII-4

N112T1310002
179

Figure VII-1. 56-in. Hinged Joint

9112-58

Figure VII-2. 56-in. Gimbaled Joint

9112-59

Figure VII-3. 42-in. Hinged Joint

9112-60

Figure VII-4. 42-in. Gimbaled Joint

9112-61

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APPENDIX A
SELECTED OPTIMUM BELLOWS GEOMETRIES PER
TIME-INDEPENDENT SCREENING RULES
1a, 2a, 3, AND 4

Table A-1 is the matrix of 373 optimum bellows element geometries selected by consideration of time-independent stresses and critical buckling pressures. Table A-2 is the array of 56 preconceptual, unreinforced, geometries of formed bellows from which the conceptual geometries may be selected.

The nomenclature used in Table A-1 and A-2 is:

SB = Selection basis as defined in Subsection VI-B

L = Live-length of bellows (in.)

N_p = Number of plies

T = Ply thickness (in.)

N_c = Number of convolutions

H = Convolution height (in.)

σ_H = Hoop stress (ksi)

σ_B = Pressure bending stress (ksi)

σ_b = Deflection bending stress (ksi)

p_c = Critical buckling pressure (psi)

$S_{o, r}$ = Minimum required allowable primary membrane stress
for time-independent screening rules in Section VI (ksi).

N112TI310002

TABLE A-1
REDUCED SET OF OPTIMUM GEOMETRIES PER
SCREENING RULES 1a, 2a, 3 AND 4
(Sheet 1 of 9)

42-in. Articulating Bellows										
SB	L	N _p	T	N _c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	P _c	$3/2 S_{o,r}$
3	14	1	0.08	8	1.50	25.0	58.4	134.9	9,210	67.4
3	14	1	0.10	8	2.00	16.4	54.4	116.7	8,400	58.4
3	14	1	0.12	8	2.50	11.6	52.8	105.6	8,100	52.8
3	14	1	0.02	7	1.00	35.7	95.1	131.7	2,800	95.1
3	14	1	0.025	7	1.00	28.5	66.5	111.0	4,800	66.5
3	14	1	0.03	9	1.00	20.1	46.5	95.6	5,600	47.8
3	18	1	0.08	10	1.50	25.6	59.0	120.5	5,700	60.3
3	18	1	0.10	10	2.00	16.7	54.7	104.9	5,200	54.7
3	18	1	0.12	10	2.00	14.0	40.4	99.5	8,900	49.8
3	22	1	0.08	12	1.50	25.9	59.3	111.2	3,900	59.3
3	22	1	0.10	12	1.50	20.7	42.1	105.1	7,400	52.5
3	22	1	0.12	12	2.00	14.2	40.6	90.2	6,100	45.1
4a	22	1	0.08	27	1.5	13.4	46.8	81.0	2,200	46.8
4a	22	1	0.10	22	2.0	10.2	48.2	78.6	2,340	48.2
4a	22	1	0.12	18	2.0	10.1	36.5	75.8	4,640	38.0
4a	22	1	0.14	15	2.5	8.6	38.9	74.9	4,650	38.9
4a	22	1	0.16	13	3.0	7.4	40.8	74.5	4,760	40.8
4a	22	1	0.18	12	3.5	6.3	42.2	73.4	4,770	42.2
4b	22	1	0.08	27	1.0	18.5	33.3	94.1	6,520	47.1
4b	22	1	0.10	22	1.5	12.8	34.2	79.8	5,030	39.9
4b	22	1	0.12	18	1.5	12.6	27.4	87.8	9,800	43.9
4b	22	1	0.14	15	2.0	10.1	29.5	79.7	8,340	39.9
4b	22	1	0.16	13	2.5	8.5	31.7	75.8	7,650	37.9
4b	22	1	0.18	12	3.0	7.1	33.5	72.7	7,160	36.3
1	22	4	0.030	45	0.80	9.6	26.5	53.1	1,840	26.5
2	22	4	0.030	17	0.80	20.5	37.4	78.9	3,420	39.5
2	22	4	0.030	13	0.90(min)	22.8	44.2	83.1	3,320	44.2
1	22	4	0.035	39	0.90	8.5	24.2	51.3	2,350	25.6
2	22	4	0.035	15	1.00	16.6	36.0	73.1	3,330	36.5
2	22	4	0.035	11	1.10(min)	19.3	42.8	78.9	3,620	42.8
1	22	4	0.040	34	1.10	7.9	25.9	52.2	2,450	26.1
2	22	4	0.040	18	1.10	11.9	29.9	63.8	3,330	31.9
2	22	4	0.040	10	1.20(min)	17.1	38.5	75.8	4,620	38.5
1	22	4	0.045	30	1.20	6.6	23.5	48.8	2,760	24.4
2	22	4	0.045	18	1.30	9.4	29.2	59.10	3,040	29.6
2	22	4	0.045	10	1.30	14.4	34.2	71.7	5,130	35.8
1	22	4	0.050	27	1.40	5.7	24.3	47.8	3,270	24.3
2	22	4	0.050	19	1.50	7.7	26.3	54.9	3,350	27.4
2	22	4	0.050	11	1.50	11.1	32.5	65.7	4,240	32.8
3	22	4	0.03	13	0.90	22.8	44.2	83.1	3,300	44.2
3	28	1	0.08	16	1.50	25.0	58.4	100.4	2,300	58.4
3	28	1	0.10	16	1.50	20.0	41.4	91.0	4,500	45.5
3	28	1	0.12	16	2.00	13.6	40.0	79.2	3,600	40.0

N112TI310002

TABLE A-1
REDUCED SET OF OPTIMUM GEOMETRIES PER
SCREENING RULES 1a, 2a, 3 AND 4
(Sheet 2 of 9)

SB	L	N _p	T	N _c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	P _c	$3/2 S_{o,r}$
4a	28	1	0 080	35	1 5	13 2	46 6	75 7	1,440	46 5
4a	28	1	0 100	28	1 5	12 8	34 2	71 3	3,140	34 2
4a	28	1	0 120	23	2 0	10 1	36 5	68 5	2,850	36 5
4a	28	1	0 140	20	2 5	8 2	38 5	67 1	2,870	38 6
4a	28	1	0 160	17	2 5	8 3	31 5	66 4	4,670	33 1
4a	28	1	0 180	15	3 0	7 2	33 6	65 5	4,480	33 6
4b	28	1	0 080	35	1 0	18 2	33 0	81 6	4,030	40 8
4b	28	1	0 100	28	1 0	17 4	26 9	93 5	8,830	46 8
4b	28	1	0 120	23	1 5	12 6	27 4	75 6	6,140	37 8
4b	28	1	0 140	20	2 0	9 8	29 2	68 4	5,060	34 2
4b	28	1	0 160	17	2 0	9 8	24 6	73 1	8,260	36 6
4b	28	1	0 180	15	2 5	8 2	26 5	68 2	7,170	34 0
2	28	4	0 03	46	0 050	16 5	23 1	68 6	4,330	34 3
2	28	4	0 03	42	0 050	17 7	24 3	71 9	4,530	35 9
2	28	4	0 03	38	0 050	19 0	25 6	75 5	4,830	37 8
2	28	4	0 035	46	1 00	8 3	27 7	49 5	1 160	27 7
2	28	4	0 035	42	1 00	9 0	28 4	51 1	1 260	28 4
2	28	4	0 035	38	1 00	9 8	29 2	53 0	1,350	29 2
2	28	4	0 040	43	1 00	7 7	22 5	45 9	1,700	22 9
2	28	4	0 040	39	1 00	8 4	23 2	47 9	1,860	23 9
2	28	4	0 040	35	1 00	9 1	23 9	50 0	2,060	25 0
2	28	4	0 045	38	1 00	7 6	19 3	46 1	2,770	23 0
2	28	4	0 045	34	1 00	8 3	20 0	48 5	2,960	24 3
2	28	4	0 045	30	1 00	9 2	20 9	51 3	3,160	25 6
2	28	4	0 020	27	1 00	22 4	81 8	139 3	300	81 8
2	28	4	0 025	22	1 00	20 9	58 9	88 0	725	58 9
2	28	4	0 030	18	1 00	20 1	46 5	75 4	1,430	40 5
2	28	4	0 035	14	1 00	20 4	39 8	73 0	2,930	39 8
2	28	4	0 040	11	1 50	16 4	49 8	74 5	1,800	40 8
2	28	4	0 045	10	1 50	15 5	41 9	68 9	3,030	41 9
3	28	4	0 02	15	1 00(min)	34 1	93 5	122 3	600	93 5
3	28	4	0 025	14	1 00(min)	28 5	66 5	93 5	1,200	66 5
3	28	4	0 03	18	0 80	23 2	40 1	76 3	2,500	46 1
4a	28	4	0 020	87	0 50	14 9	29 7	58 1	830	29 7
4a	28	4	0 025	70	0 50	14 3	23 8	56 4	1,940	28 2
4a	28	4	0 030	58	1 00	7 9	34 3	55 9	670	34 4
4a	28	4	0 035	50	1 00	7 7	27 1	48 2	1,070	27 0
4a	28	4	0 040	43	1 50	5 5	38 9	57 1	700	38 9
4a	28	4	0 045	38	1 00	7 6	19 3	46 1	2,770	23 1
4b	28	4	0 030	58	0 50	13 9	20 5	61 1	3 740	30 6
4b	28	4	0 035	50	0 50	13 3	18 1	67 6	6,480	33 8
4b	28	4	0 040	43	1 00	7 7	22 5	45 9	1,770	23 0
4b	28	4	0 045	38	0 50	12 7	15 6	88 0	16,900	44 0

N112TI310002

TABLE A-1
REDUCED SET OF OPTIMUM GEOMETRIES PER
SCREENING RULES 1a, 2a, 3 AND 4
(Sheet 3 of 9)

SB	L	N _p	T	N _c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	P _c	$3/2 S_{o,r}$
2	36	1	0.14	25	1.0	16.4	21.2	108.1	6,400	54.0
2	36	1	0.14	24	1.0	16.8	21.6	110.3	18,000	55.0
2	36	1	0.14	23	1.0	17.0	22.2	112.8	19,400	56.4
2	36	1	0.16	22	1.5	11.9	20.2	78.3	10,000	38.6
2	36	1	0.16	21	1.5	12.4	20.7	80.2	10,000	40.1
2	36	1	0.16	20	1.5	12.8	21.1	82.0	11,000	41.0
2	36	1	0.16	19	1.5	13.3	21.6	84.1	11,000	42.0
2	36	1	0.16	18	1.5	13.9	22.2	86.4	12,000	43.2
2	36	1	0.18	20	2.0	9.3	21.0	56.6	7,600	28.3
2	36	1	0.18	19	2.0	9.7	21.4	67.6	7,800	33.8
3	36	1	0.10	21	1.50	19.7	41.1	81.0	2,700	42.0
3	36	1	0.12	21	1.50	16.5	31.3	77.3	4,600	38.6
4a	36	1	0.080	45	1.5	13.2	46.6	73.5	852	46.5
4a	36	1	0.100	36	1.5	12.8	34.2	65.0	1,950	34.2
4a	36	1	0.120	30	2.0	10.0	36.4	62.9	1,770	36.4
4a	36	1	0.140	25	2.0	10.0	29.4	61.6	3,170	30.8
4a	36	1	0.160	22	2.5	8.2	31.4	59.6	2,890	31.4
4a	36	1	0.180	20	3.0	6.9	33.3	58.6	2,600	33.3
4b	36	1	0.080	45	1.0	18.2	33.0	72.7	2,440	36.3
4b	36	1	0.100	36	1.0	17.4	26.9	79.8	5,340	39.9
4b	36	1	0.120	30	1.5	12.4	27.2	65.6	3,660	32.8
4b	36	1	0.140	25	1.5	12.3	23.2	71.4	6,460	35.7
4b	36	1	0.160	22	2.0	9.7	24.5	62.9	5,070	31.4
4b	36	1	0.180	20	2.5	7.9	26.2	58.8	4,290	29.4
1	36	2	0.06	50	1.00	11.2	24.4	53.2	1,960	26.6
2	36	2	0.06	30	1.00	16.7	29.9	66.6	2,640	33.3
2	36	2	0.06	20	1.00	22.2	35.4	78.4	3,530	39.2
2	36	2	0.06	12	1.60(min)	23.4	57.2	85.7	1,830	57.2
1	36	2	0.07	40	1.20	10.0	24.0	52.2	2,170	26.1
2	36	2	0.07	30	1.20	12.6	26.6	59.2	2,650	29.6
2	36	2	0.07	20	1.20	17.0	31.0	69.6	3,440	39.8
2	36	2	0.07	10	2.00(min)	19.9	58.7	84.5	1,935	58.7
1	36	2	0.08	35	1.40	8.7	23.2	50.3	2,380	25.2
2	36	2	0.08	25	1.40	11.4	25.9	58.0	2,960	29.0
2	36	2	0.08	15	1.60	15.1	34.1	67.4	3,050	34.1
2	36	2	0.08	11	1.80(min)	17.4	41.4	72.8	3,240	41.4
1	36	2	0.09	33	1.60	7.3	22.3	47.7	2,490	23.8
2	36	2	0.09	25	1.60	9.2	24.2	53.6	2,980	26.8
2	36	2	0.09	15	1.80	12.5	31.5	63.1	3,160	31.6
2	36	2	0.09	11	1.80(min)	15.5	34.5	69.2	4,340	34.5
1	36	2	0.10	30	1.80	6.5	21.9	46.5	2,600	23.2
2	36	2	0.10	24	1.80	7.8	23.2	51.0	3,090	25.5
2	36	2	0.10	15	2.00	10.0	29.0	58.3	3,171	29.2
2	36	2	0.10	10	2.00	14.0	33.0	67.3	4,950	33.6

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TABLE A-1
REDUCED SET OF OPTIMUM GEOMETRIES PER
SCREENING RULES 1a, 2a, 3 AND 4
(Sheet 4 of 9)

SB	L	N _p	T	N _c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	P _c	$3/2 S_{o,r}$
1	36	2	0.11	27	2.00	6.0	21.7	45.9	2,800	22.9
2	36	2	0.11	21	2.00	7.4	23.1	50.9	3,400	25.4
2	36	2	0.11	15	2.20	9.0	28.0	56.8	3,380	28.4
2	36	2	0.11	11	2.20	11.2	30.2	62.5	4,460	31.3
1	36	2	0.12	25	2.20	5.4	21.4	45.1	3,000	22.5
2	36	2	0.12	19	2.40	6.4	25.4	50.0	2,900	25.4
2	36	2	0.12	15	2.40	7.7	26.7	54.3	3,490	27.2
2	36	2	0.12	11	2.40	9.8	28.8	60.0	4,470	30.0
3	36	2	0.06	20	1.00	22.2	35.4	78.4	3,500	39.2
2	36	4	0.03	47	0.50	19.5	26.1	67.1	3,030	33.5
2	36	4	0.03	43	0.50	20.7	27.3	69.8	3,232	34.9
2	36	4	0.03	39	0.50	22.1	28.7	72.9	3,430	36.4
2	36	4	0.035	48	0.50	16.5	21.3	67.2	4,740	36.6
2	36	4	0.035	44	0.50	17.5	22.3	70.0	4,940	35.0
2	36	4	0.035	40	1.00	11.5	30.9	54.6	960	30.9
2	36	4	0.040	48	1.00	8.7	23.5	45.4	1,180	23.5
2	36	4	0.040	44	1.00	9.3	24.1	46.9	1,270	24.1
2	36	4	0.040	40	1.00	10.1	24.9	48.6	1,370	24.9
2	36	4	0.045	50	1.00	7.5	19.2	41.4	1,690	20.7
2	36	4	0.045	46	1.00	8.0	19.7	42.9	1,785	21.5
2	36	4	0.045	42	1.00	8.6	20.3	44.5	1,870	22.3
3	36	4	0.025	22	1.00(min)	25.0	53.0	82.5	500	53.0
3	36	4	0.030	19	1.00(min)	23.0	49.4	75.9	1,100	49.4
4a	36	4	0.020	112	0.50	14.9	29.7	59.8	540	29.8
4a	36	4	0.025	90	0.50	14.3	23.8	52.3	1,250	26.1
4a	36	4	0.030	75	0.50	13.8	20.4	53.6	2,250	26.8
4a	36	4	0.035	64	1.00	7.8	27.2	48.0	690	27.2
4a	36	4	0.040	56	1.00	7.6	22.4	43.0	1,090	21.4
4a	36	4	0.045	50	1.00	7.5	19.2	41.4	1,690	20.7
4b	36	4	0.035	64	0.50	13.4	18.2	58.1	3,950	29.1
4b	36	4	0.040	56	0.50	12.9	16.6	64.4	6,350	32.2
4b	36	4	0.045	50	0.50	12.5	15.4	71.7	9,650	35.8
1	45	1	0.08	56	1.00	18.3	33.1	67.3	1,550	33.7
1	45	1	0.10	45	1.50	12.8	34.2	61.4	1,270	34.2
3	45	1	0.10	26	1.50	19.9	41.3	75.5	1,740	41.3
1	45	1	0.12	37	1.50	12.6	27.4	59.7	2,370	27.4
3	45	1	0.12	26	1.50	16.6	31.4	69.6	2,950	34.8
1	45	1	0.14	32	2.00	9.8	29.2	56.0	1,990	28.0
1	45	1	0.16	28	2.00	9.6	24.4	55.8	3,190	27.9
1	45	1	0.18	25	2.00	9.3	21.0	57.7	4,890	28.8
1	45	4	0.03	281	0.60	12.1	21.6	46.6	970	23.3
3	45	4	0.03	25	1.00	22.2	48.6	74.7	640	48.6
1	45	4	0.035	80	0.60	11.7	18.7	45.8	1,670	22.9
1	45	4	0.040	70	0.80	9.1	18.9	41.5	1,290	20.8
1	45	4	0.045	62	1.00	7.5	19.2	44.1	1,000	22.0

N112TI310002

TABLE A-1
REDUCED SET OF OPTIMUM GEOMETRIES PER
SCREENING RULES 1a, 2a, 3 AND 4
(Sheet 5 of 9)

SB	L	N _p	T	N _c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	P _c	$3/2 S_{o,r}$
1	55	1	0 080	68	1 00	18 4	33 2	64 5	1,060	33 2
1	55	1	0 100	55	1 00	17 4	26 9	64 1	2,360	33 0
1	55	1	0 120	45	1 50	12 6	27 2	55 5	1,680	27 8
3	55	1	0 120	26	1 50	19 2	34 0	69 6	2,250	34 8
1	55	1	0 140	39	1 50	12 1	33 0	66 1	2,790	33 0
1	55	1	0 160	34	2 00	9 6	24 4	51 4	2,100	25 7
1	55	1	0 180	30	2 50	8 1	26 4	50 0	1,800	26 4
1	65	1	0 080	81	1 00	18 3	33 1	63 7	700	33 1
1	65	1	0 100	65	1 00	17 4	26 9	60 1	1,670	26 9
1	65	1	0 120	54	1 50	12 5	27 3	53 2	1,100	26 6
3	65	1	0 120	35	1 50	17 5	32 3	62 7	1,570	32 3
1	65	1	0 140	46	1 50	12 2	23 1	52 4	2,000	26 2
1	65	1	0 160	40	2 00	9 7	24 5	48 8	1,500	24 5
1	65	1	0 180	36	2 00	9 4	21 1	48 2	2,300	24 1
1	75	1	0 08	93	1 00	18 4	33 2	64 9	670	33 2
1	75	1	0 10	75	1 00	17 4	26 9	57 7	1,280	28 8
1	75	1	0 12	62	1 50	12 5	27 3	52 4	800	27 3
3	75	1	0 12	43	1 50	16 7	31 5	59 4	1,180	31 5
1	75	1	0 14	53	1 50	12 2	23 1	49 9	1,500	24 9
1	75	1	0 16	46	2 00	9 7	24 5	47 2	1,100	24 5
1	75	1	0 18	41	2 00	9 4	21 1	45 8	1,800	22 9

42-in Axial Motion Bellows

1	45	1	0 08	56	1 5	13 3	46 7	96 0	500	48 0
1	45	1	0 10	45	2 0	10 2	48 2	92 3	500	48 2
1	45	1	0 12	37	2 5	8 5	49 7	91 2	600	49 7
1	45	1	0 14	32	3 0	7 2	50 8	90 0	600	50 8
1	45	1	0 16	28	3 5	6 3	51 8	89 5	700	51 8
1	45	1	0 18	25	3 5	6 2	42 1	88 1	1,100	44 0
3	45	1	0 10	26	2 0	16 2	54 2	116 6	850	58 3
3	45	1	0 12	26	2 5	11 5	52 7	104 9	780	52 5
1	45	4	0,020	140	0 50	14 9	29 7	71 7	320	35 3
1	45	4	0 025	112	0 50	14 3	23 8	84 3	700	42 2
1	45	4	0 030	93	0 50	13 9	20 5	101 6	1,400	50 8
1	45	4	0,035	80	1 00	7 8	27 2	60 7	400	30 4
1	45	4	0 040	70	1 00	7 6	22 4	64 2	600	32 1
1	45	4	0 045	62	1 00	7 5	19 2	70 0	1,000	35 0
3	45	4	0 030	25	1 00	22 2	48 6	101 1	640	50 6

56-in Articulating Bellows

1	5	1	0 03	16	1 75	1 3	26 8	50 2	68	26 8
2	5	1	0 03	11	1 75	1 9	27 4	59 4	47	29 7
2	5	1	0 03	6	2 00	2 9	36 2	76 7	38	38 4
3	5	1	0 03	3	2 25	4 9	47 1	98 9	20	49 3
1	5	1	0 05	10	2 75	0 8	23 5	48 8	316	24 4
2	5	1	0 05	8	3 00	1 0	28 0	54 1	314	28 0
2	5	1	0 05	6	3 00	1 3	28 3	61 3	211	30 6

N112TI310002

TABLE A-1
REDUCED SET OF OPTIMUM GEOMETRIES PER
SCREENING RULES 1a, 2a, 3 AND 4
(Sheet 6 of 9)

SB	L	N _p	T	N _c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	P _c	$3/2 S_{o,r}$
3	5	1	0 05	3	2 75	2 5	2 2	88 4	110	44 2
1	5	1	0 07	7(max)	4 00	0 6	25 1	49 4	720	25 0
2	5	1	0 07	6	4 00	0 7	25 2	52 9	620	26 4
2	5	1	0 07	5	4 00	0 8	25 3	57 4	516	28 6
3	5	1	0 07	3	3 00	1 7	15 5	92 7	110	46 4
1	5	1	0 09	5	4 00	0 7	15 5	56 8	720	28 3
2	5	1	0 09	4	4 00	0 8	15 6	65 0	617	32 4
2	5	1	0 09	3	4 00	1 0	15 8	77 2	413	38 6
3	5	1	0 09	3	3 00	1 3	9 6	109 4	210	54 7
1	5	2	0 025	16	1 75	0 8	19 2	38 6	113	19 4
2	5	2	0 025	12	1 75	1 0	19 4	44 1	86	22 0
2	5	2	0 025	10	1 75	1 2	19 6	48 3	72	24 2
2	5	2	0 025	8	2 00	1 4	25 4	52 4	85	26 2
3	5	2	0 025	3	2 25	2 9	33 3	75 7	60	37 7
1	5	2	0 035	11	2 5	0 6	19 7	39 1	304	19 5
2	5	2	0 035	8	2 5	0 8	19 9	45 2	216	22 5
2	5	2	0 035	7	2 75	0 8	24 0	47 8	316	23 8
3	5	2	0 035	3	3 00	1 7	29 3	66 3	110	33 2
1	5	4	0 01	31	0 75	1 2	11 7	32 9	15	16 4
2	5	4	0 01	20	1 00	1 3	20 0	38 3	22	20 0
2	5	4	0 01	12	1 00	2 1	20 8	47 9	13	24 0
1	5	4	0 02	15	1 75	0 5	14 9	31 3	169	15 6
2	5	4	0 02	10	2 00	0 7	19 4	37 4	169	19 4
2	5	4	0 02	6	2 00	1 1	19 8	46 3	112	23 1
3	5	4	0 02	3	2 25	1 8	25 5	48 7	110	29 3
1	5	4	0 03	10	2 75	0 3	16 1	31 3	623	16 0
2	5	4	0 03	8	2 75	0 4	16 2	34 5	531	17 2
2	5	4	0 03	6	3 00	0 5	19 2	38 8	525	19 4
3	5	4	0 03	3	3 00	1 0	19 7	50 8	210	25 4
1	10	1	0 03	33	1 25	1 7	14 7	36 2	26	18 1
2	10	1	0 03	22	1 50	2 2	20 9	42 7	30	21 4
2	10	1	0 03	14	1 75	2 9	28 4	52 1	30	28 4
1	10	1	0 05	20	2 25	1 0	16 2	34 8	147	17 4
2	10	1	0 05	16	2 50	1 1	19 9	38 4	161	20 0
2	10	1	0 05	12	2 50	1 5	20 3	43 5	118	21 8
3	10	1	0 05	6	2 75	2 5	25 2	57 3	110	28 7
1	10	1	0 07	14	3 25	0 7	16 9	34 8	437	17 0
2	10	1	0 07	12	3 25	0 8	17 0	37 3	432	18 6
2	10	1	0 07	10	3 50	0 9	19 7	40 3	430	20 1
2	10	1	0 07	8	3 75	1 1	22 6	44 3	425	22 5
3	10	1	0 07	6	3 00	2 9	16 7	73 4	110	36 7
1	10	1	0 09	11	4 00(max)	0 6	15 4	34 7	846	17 4
2	10	1	0 09	10	4 00	0 7	15 5	36 4	742	18 2

N112TI310002

TABLE A-1
REDUCED SET OF OPTIMUM GEOMETRIES PER
SCREENING RULES 1a, 2a, 3 AND 4
(Sheet 7 of 9)

SB	L	N _p	T	N _c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	P _c	$3/2 S_{o,r}$
2	10	1	0.09	9	4.00	0.7	15.5	38.2	638	19.0
2	10	1	0.09	8	4.00	0.8	15.6	40.5	489	20.2
2	10	1	0.09	6	4.00	1.0	15.8	46.7	426	23.4
3	10	1	0.09	6	3.00	1.3	9.6	59.7	220	29.8
1	10	2	0.025	30	1.50	1.0	14.5	28.6	68	14.6
2	10	2	0.025	25	1.50	1.2	14.7	31.0	57	15.4
2	10	2	0.025	20	1.50	1.4	14.9	34.3	46	17.1
2	10	2	0.025	14(min)	1.75	1.7	26.1	39.6	50	20.1
1	10	2	0.035	20	2.00	0.8	13.0	29.1	148	14.6
2	10	2	0.035	14	2.25	1.0	16.5	33.9	147	17.0
2	10	2	0.035	10	2.25	1.3	16.8	39.0	120	19.5
3	10	2	0.035	6	2.50	1.9	21.0	46.8	114	23.4
1	10	4	0.02	30	1.5	0.6	11.1	22.3	109	11.1
2	10	4	0.02	20	1.5	0.9	11.1	26.8	74	13.4
2	10	4	0.02	12(min)	1.75	1.2	15.6	32.8	69	16.4
1	10	4	0.03	20	2.25	0.4	10.9	22.0	353	11.0
2	10	4	0.03	12	2.50	0.6	13.6	27.4	291	13.6
2	10	4	0.03	10	2.50	0.7	13.7	29.4	236	14.7
2	10	4	0.03	8	2.50	0.9	13.9	32.1	230	16.0
3	10	4	0.03	6	2.75	1.0	16.8	35.2	225	17.6
2	10	4	0.03	4	3.00	1.4	20.1	40.4	120	20.2
4a	15	1	0.03	50	2.00	1.1	34.4	42.9	100	34.4
4a	15	1	0.05	30	2.00	1.1	13.1	28.7	106	14.4
4a	15	1	0.07	21	3.00	0.8	14.6	28.7	350	14.6
3	15	1	0.07	9	3.00	1.7	15.5	41.7	120	20.8
4a	15	1	0.09	16	4.00	0.6	15.4	28.8	770	15.4
3	15	1	0.09	9	3.00	1.3	9.6	43.2	230	21.6
4a	15	1	0.11	13	4.50	0.6	13.2	28.9	1,080	14.4
3	15	1	0.11	9	3.00	1.1	6.7	47.7	240	23.8
4a	15	1	0.13	11	5.50	0.5	13.9	28.8	1,890	14.4
4b	15	1	0.05	30	1.5	1.4	8.2	33.8	27	17.0
4b	15	1	0.07	21	2.5	0.9	10.5	29.8	240	14.8
4b	15	1	0.09	16	3.5	0.7	12.0	28.9	560	14.4
4b	15	1	0.11	13	4.0	0.6	10.5	29.7	770	14.8
4b	15	1	0.13	11	5.0	0.5	11.6	29.1	1,480	14.6
1	15	4	0.020	46	1.5	0.6	11.1	19.0	110	11.1
2	15	4	0.020	22	1.5	1.2	11.7	25.1	54	12.5
3	15	4	0.020	10	1.5	2.3	12.8	33.7	17	16.8
1	15	4	0.025	37	1.5	0.6	7.4	18.5	110	9.3
2	15	4	0.025	17	2.0	0.9	12.9	25.1	120	12.9
3	15	4	0.025	9	2.0	1.6	13.6	31.2	120	15.6
1	15	4	0.030	31	2.0	0.4	8.7	17.9	260	8.9
2	15	4	0.030	15	2.0	0.9	9.2	24.8	145	12.4
3	15	4	0.030	11	2.5	0.9	13.9	27.1	175	13.9

N112TI310002

TABLE A-1
REDUCED SET OF OPTIMUM GEOMETRIES PER
SCREENING RULES 1a, 2a, 3 AND 4
(Sheet 8 of 9)

SB	L	N _p	T	N _c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	P _c	$3/2 S_{o,r}$
1	15	1	0.035	26	2.0	0.51	6.6	18.7	390	9.4
2	15	1	0.035	10	2.5	0.9	10.5	26.8	245	13.4
1	15	1	0.040	23	2.5	0.4	7.7	17.9	500	9.0
2	15	1	0.040	11	3.0	0.6	11.1	23.9	460	12.0
1	15	1	0.045	20	3.0	0.3	8.5	17.9	800	9.0
2	15	4	0.045	12	3.0	0.5	8.8	22.4	580	11.2
4a	20	1	0.03	66	2.0	1.1	34.4	43.8	102	34.4
4a	20	1	0.05	40	2.0	1.1	13.1	25.3	106	13.0
4a	20	1	0.07	28	3.0	0.8	15.4	26.3	330	15.4
3	20	1	0.07	12	3.0	1.7	15.5	35.5	130	17.7
4a	20	1	0.09	22	3.5	0.7	12.0	24.6	581	13.0
3	20	1	0.09	12	3.0	1.3	9.6	34.9	240	17.4
4a	20	1	0.11	18	4.0	0.6	10.5	24.7	890	12.3
3	20	1	0.11	12	3.0	1.1	6.7	37.6	250	18.8
4a	20	1	0.13	15	5.0	0.5	11.6	24.7	1,400	12.3
4b	20	1	0.03	66	1.0	2.1	10.4	27.3	14	13.6
4b	20	1	0.05	40	1.5	1.4	8.2	27.7	36	13.8
4b	20	1	0.07	28	2.5	0.9	10.5	35.7	258	17.8
4b	20	1	0.09	22	3.0	0.8	9.1	25.5	371	12.8
4b	20	1	0.11	18	3.5	0.7	9.3	27.1	580	13.5
4b	20	1	0.13	15	4.5	0.5	9.5	25.2	1,000	12.6
1	20	4	0.020	62	1.5	0.6	11.1	17.6	110	11.1
2	20	4	0.020	30	2.0	0.9	19.7	27.0	130	19.6
3	20	4	0.020	14	1.5	2.2	12.7	28.4	23	14.2
1	20	4	0.025	50	1.5	0.6	7.4	15.9	110	8.0
2	20	4	0.025	22	2.0	1.0	13.0	22.8	120	13.0
3	20	4	0.025	14	2.0	1.4	13.4	26.0	70	13.4
1	20	4	0.030	41	1.5	0.6	5.3	16.9	190	8.4
2	20	4	0.030	17	2.0	1.0	9.3	22.6	150	11.2
3	20	4	0.030	13	2.0	1.2	9.5	24.9	140	12.5
1	20	4	0.035	35	2.0	0.4	6.5	15.7	300	7.8
2	20	4	0.035	15	2.5	0.8	10.4	22.0	270	11.0
1	25	1	0.07	35	2.5	0.9	10.5	22.6	270	11.3
1	25	1	0.09	27	3.0	0.8	9.1	22.6	390	11.3
3	25	1	0.09	15	3.0	1.3	9.6	30.1	250	15.0
1	25	1	0.110	22	4.0	0.6	10.5	22.2	700	11.0
3	25	1	0.110	15	3.0	1.1	6.7	31.5	260	15.7
1	25	1	0.130	19	4.5	0.5	9.5	22.0	1,100	11.0
2	25	1	0.07	16	3.0	1.6	15.4	31.0	240	15.4
2	25	1	0.09	10	4.0	1.5	16.3	33.1	350	16.5
2	25	1	0.11	10	4.5	1.1	13.7	30.5	560	15.3
2	25	1	0.13	10	5.5	0.8	14.2	28.4	1,090	14.2

N112TI310002

TABLE A-1
REDUCED SET OF OPTIMUM GEOMETRIES PER
SCREENING RULES 1a, 2a, 3, AND 4
(Sheet 9 of 9)

SB	L	N _p	T	N _c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	P _c	$3/2 S_{o,r}$
1	25	4	0.020	78	1.5	0.6	11.1	17.1	110	11.1
2	25	4	0.020	34	1.5	1.3	11.8	21.3	50	11.8
3	25	4	0.020	14	1.5	2.6	13.1	27.2	18	13.1
1	25	4	0.025	62	1.5	0.6	7.4	14.7	110	7.4
2	25	4	0.025	22	1.5	1.5	8.3	22.3	44	11.1
3	25	4	0.025	14	1.5	1.7	8.5	24.0	37	12.0
1	25	4	0.030	52	1.5	0.6	5.3	14.7	100	7.4
2	25	4	0.030	20	2.0	1.0	9.3	20.6	100	10.4
3	25	4	0.030	16	2.0	1.3	9.6	22.3	150	11.2
1	25	4	0.035	44	2.0	0.4	6.5	14.1	260	7.0
2	25	4	0.035	16	2.5	0.9	10.5	21.0	270	10.5
56-in. Axial Motion Bellows										
1	25	1	0.03	83	1.5	1.5	20.2	40.9	45	20.5
1	25	1	0.05	50	2.5	0.9	19.7	40.2	200	20.1
1	25	1	0.07	35	3.5	0.7	19.7	40.5	510	20.2
3	25	1	0.07	16	3.0	1.6	15.4	65.8	240	32.9
1	25	1	0.09	27	4.0	0.6	15.4	61.4	740	20.7
3	25	1	0.09	16	3.0	1.3	9.6	77.7	250	38.8
1	25	1	0.11	22	4.0	0.6	10.5	48.2	700	24.1
3	25	1	0.11	16	3.0	1.0	6.6	86.7	270	43.3
1	25	1	0.13	19	4.0	0.6	7.7	58.0	800	29.0
1	25	4	0.02	78	1.5	0.6	11.1	25.6	110	12.8
3	25	4	0.02	14	2.0	2.1	20.9	50.1	43	25.0
1	25	4	0.025	62	2.0	0.5	12.5	25.4	260	12.7
3	25	4	0.025	14	2.5	1.4	20.2	47.3	110	23.7
1	25	4	0.030	52	2.0	0.4	8.7	26.6	260	13.3
3	25	4	0.030	16	3.0	0.9	19.7	40.0	260	20.0
1	25	4	0.035	44	2.5	0.4	10.0	25.9	490	12.9

N112TI310002

TABLE A-2
GEOMETRIES SELECTED FROM TABLE A-1 WHICH: (1) MINIMIZE $S_{o,r}$
(2) REPRESENT COMMONLY MANUFACTURED SHAPES; AND
(3) REPRESENT THICKNESS EFFECTS ON CONVOLUTION
HEIGHT AND $S_{o,r}$
(Sheet 1 of 3)

42-in. Articulating Bellows										
SB	L	N_p	T	N_c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	p_c	$3/2 S_{o,r}$
3	14	1	0.12	8	2.50	11.6	52.8	105.6	8,100	52.8
3*	14	1	0.08	8	1.50	25.0	58.4	134.9	9,210	67.4
3	14	4	0.03	9	1.00	20.1	46.5	95.6	5,600	47.8
3*	14	4	0.02	7	1.00	35.7	95.1	131.7	2,800	95.1
3	18	1	0.12	10	2.00	14.0	40.4	99.5	8,900	49.8
3*	18	1	0.08	10	1.50	25.6	59.0	120.5	5,700	60.3
3	22	1	0.12	12	2.00	14.2	40.6	90.2	6,100	45.1
4b	22	1	0.18	12	3.00	7.1	33.5	72.7	7,160	36.3
4a*	22	1	0.08	27	1.50	13.4	46.8	81.0	2,200	46.8
1	22	4	0.05	27	1.40	5.7	24.3	47.8	3,270	24.3
3	22	4	0.03	12	0.90	22.8	44.2	83.1	3,300	44.2
1*	22	4	0.03	45	0.80	9.6	26.5	53.1	1,840	26.5
4a	28	1	0.16	17	2.50	8.3	31.5	66.4	4,670	33.1
3	28	1	0.12	16	2.00	13.6	40.0	79.2	3,600	40.0
4b*	28	1	0.08	35	1.00	18.2	33.0	81.6	4,030	40.8
4b	28	4	0.04	43	1.00	7.7	22.5	45.9	1,770	23.0
3	28	4	0.03	18	0.80	23.2	40.1	76.3	2,500	40.1
4a*	28	4	0.025	70	0.50	14.3	23.8	56.4	1,940	28.2
4b	36	1	0.18	20	2.50	7.9	26.2	58.8	4,290	29.4
3	36	1	0.12	21	1.50	16.5	31.3	77.3	4,600	38.6
4b*	36	1	0.08	45	1.00	18.2	33.0	72.7	2,440	36.3
1	36	2	0.12	25	2.20	5.4	21.4	45.1	3,000	22.5
3	36	2	0.06	20	1.00	22.2	35.4	78.4	3,500	39.2
2*	36	2	0.06	50	1.00	11.2	24.4	53.2	1,960	26.6
4a	36	4	0.045	50	1.00	7.5	19.2	41.4	1,690	20.7
3	36	4	0.03	19	1.00	23.0	49.4	75.9	1,100 [†]	49.4
4a*	36	4	0.025	90	0.50	14.3	23.8	52.3	1,250 [†]	26.1
1	45	1	0.12	37	1.50	12.6	27.4	59.7	2,370	27.4
3	45	1	0.12	26	1.50	16.6	31.4	69.6	2,950	34.8

N112TI310002

TABLE A-2

GEOMETRIES SELECTED FROM TABLE A-1 WHICH: (1) MINIMIZES
 (2) REPRESENT COMMONLY MANUFACTURED SHAPES; AND $S_{o,r}$
 (3) REPRESENT THICKNESS EFFECTS ON CONVOLUTION
 HEIGHT AND $S_{o,r}$
 (Sheet 2 of 3)

SB	L	N _p	T	N _c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	P _c	$3/2 S_{o,r}$
1	45	4	0.04	70	0.80	9.1	18.9	41.5	1,290 [†]	20.8
3	45	4	0.03	25	1.00	22.2	48.6	74.7	640 [†]	48.6
1	55	1	0.16	34	2.00	9.6	24.4	51.4	2,100	25.7
3	55	1	0.12	26	1.50	19.2	34.0	69.6	2,250	34.8
1	65	1	0.18	36	2.00	9.4	21.1	48.2	2,300	24.1
3	65	1	0.12	35	1.50	17.5	32.3	62.7	1,570	32.3
1	75	1	0.18	41	2.00	9.4	21.1	45.8	1,800	22.9
3	75	1	0.12	43	1.50	16.7	31.5	59.4	1,180 [†]	31.5

42-in. Axial Motion Bellows

1	45	1	0.18	25	3.5	6.2	42.1	88.1	1,100 [†]	44.0
3	45	1	0.12	26	2.5	11.5	52.7	104.9	780 [†]	52.5
1*	45	1	0.08	56	1.5	13.3	46.7	96.0	500 [†]	48.0
1	45	4	0.035	80	1.00	7.8	27.2	60.7	400 [†]	30.4
3	45	4	0.030	25	1.00	22.2	48.6	101.1	640 [†]	50.6
1*	45	4	0.025	112	0.50	14.3	23.8	84.3	700 [†]	42.2

56-in. Articulating Bellows

1	5	1	0.05	10	2.75	0.8	23.5	48.8	316	24.4
3	5	1	0.05	3	2.75	2.5	25.2	88.4	110	44.2
1*	5	1	0.09	3	4.00	1.0	15.8	77.2	413	38.6
1	5	2	0.025	16	1.75	0.8	19.2	38.6	113	19.4
3	5	2	0.035	3	3.00	1.7	29.3	66.3	110	33.2
1	5	4	0.02	15	1.75	0.5	14.9	31.3	169	15.6
3	5	4	0.03	3	3.00	1.0	19.7	50.8	210	25.4
1*	5	4	0.01	12	1.00	2.1	20.8	47.9	13 [†]	24.0
1	10	1	0.05	20	2.25	1.0	16.2	34.8	147	17.4
3	10	1	0.05	6	2.75	2.5	25.2	57.3	110	28.7
1	10	1	0.09	11	4.00	0.6	15.4	34.7	846	17.4
1	10	2	0.035	20	2.00	0.8	13.0	29.1	148	14.6
3	10	2	0.035	6	2.50	1.9	21.0	46.8	114	23.4
1*	10	2	0.025	20	1.50	1.4	14.9	34.3	46 [†]	17.1

N112TI310002

TABLE A-2

GEOMETRIES SELECTED FROM TABLE A-1 WHICH: (1) MINIMIZE $S_{o,r}$
 (2) REPRESENT COMMONLY MANUFACTURED SHAPES; AND
 (3) REPRESENT THICKNESS EFFECTS ON CONVOLUTION
 HEIGHT AND $S_{o,r}$
 (Sheet 3 of 3)

SB	L	N _p	T	N _c	H	σ_H	$(\sigma_H + \sigma_B)$	$(\sigma_H + \sigma_B + \sigma_b)$	P _c	$3/2 S_{o,r}$
1	10	4	0.030	20	2.25	0.4	10.9	22.0	353	11.0
3	10	4	0.030	6	2.75	1.0	16.8	35.2	225	17.6
1*	10	4	0.020	20	1.50	0.9	11.4	26.8	74	13.4
4b	15	1	0.09	16	3.50	0.7	12.0	28.9	560	14.4
3	15	1	0.07	9	3.00	1.7	15.5	41.7	120	20.8
4a*	15	1	0.05	30	2.00	1.1	13.1	28.7	106	14.4
1	15	4	0.030	31	2.00	0.4	8.7	17.9	260	8.9
3	15	4	0.030	11	2.50	0.9	13.9	27.1	175	13.9
2*	15	4	0.045	12	3.0	0.5	8.8	22.4	580	11.2
4a	20	1	0.13	15	5.0	0.5	11.6	24.7	1,400	12.3
3	20	1	0.09	12	3.0	1.3	9.6	34.9	240	17.4
4a*	20	1	0.05	40	2.0	1.1	13.1	25.3	106	13.0
1	20	4	0.035	35	2.0	0.4	6.5	15.7	300	7.8
3	20	4	0.030	13	2.0	1.2	9.5	24.9	140	12.5
3*	20	4	0.020	14	1.5	2.2	12.7	28.4	23 [†]	14.2
1	25	1	0.130	19	4.5	0.5	9.5	22.0	1,100	11.0
3	25	1	0.090	15	3.0	1.3	9.6	30.1	250	15.0
2*	25	1	0.070	16	3.0	1.6	15.4	31.0	240	15.4
1	25	4	0.035	44	2.0	0.4	6.5	14.1	260	7.0
3	25	4	0.030	16	2.0	1.3	9.6	22.3	150	11.2
3*	25	4	0.025	14	1.5	1.7	8.5	24.0	37 [†]	12.0
56-in. Axial Motion Bellows										
1	25	1	0.05	50	2.5	0.9	19.7	40.2	200	20.1
3	25	1	0.07	16	3.0	1.6	15.4	65.8	240	32.9
3*	25	1	0.11	16	3.0	1.0	6.6	86.7	270	43.3
1	25	4	0.025	62	2.0	0.5	12.5	25.4	260	12.7
3	25	4	0.030	16	3.0	0.9	19.7	40.0	260	20.0
3*	25	4	0.020	14	2.0	2.1	20.9	50.1	43 [†]	25.0

*Representative geometry; not minimum $S_{o,r}$

[†]Fails to satisfy stability requirements of screening rule (4), will not be further considered

N112TI310002

A-141

APPENDIX B
COMPUTATIONAL AND DESIGN TABLES FOR APPLICATION OF TIME-
DEPENDENT AND FATIGUE SCREENING RULES
1b, c, d, 2b, c, d, 5, AND 6

This Appendix contains the computational analyses performed during the Phase I effort. These analyses were used to evaluate 12 unreinforced, single, formed bellows from the 56 screened designs of Appendix A.

The nomenclature used in Tables B-1 through B-7 is:

ID = Description letter for bellows geometries listed in Table VI-2

σ_H = Hoop stress (ksi)

σ_B = Pressure bending stress (ksi)

HP = Hold period identifier listed in Table B-2

T_H = Allowable time for sustained loading with σ_H for a particular hold period (hr)

t = Design time of sustained operation during hold period (hr)

T_{HB} = Allowable time for sustained loading with $\sigma_H + \sigma_B/1.25$ for a particular hold period (hr)

n = Number of cycles for a particular loading cycle

T_{max} = Maximum metal temperature existing during a particular loading cycle

LC = Load cycle identifier listed in Table B-4

σ_b = Deflection bending stress (ksi)

$\Delta\sigma_b$ = Deflection bending stress range obtained by multiplying σ_b by the appropriate motion stress range scale factor listed in Table B-4 (ksi)

$\Delta(\sigma_H + \sigma_B)$ = Total pressure stress range obtained by multiplying $(\sigma_H + \sigma_B)$ by the appropriate pressure stress range scale factor listed in Table B-4 (ksi)

$\Delta\sigma$ = Total alternating stress range (ksi)

N112TI310002

$\Delta\epsilon$ = Total alternating strain range (10^{-3} in./in.)

N = Number of allowable cycles for a particular cyclic loading history

$(\sigma_H + \sigma_B)_R$ = Range of sustained pressure stress obtained by multiplying $\sigma_H + \sigma_B$ by the appropriate pressure stress range scale factor listed in Table B-3

$(\sigma_H + \sigma_B + \sigma_b)_R$ = Range of sustained pressure plus deflection stress obtained by multiplying σ_b by the appropriate motion stress range scale factor listed in Table B-3 and adding to $(\sigma_H + \sigma_B)_R$

T_1 = Allowable time for sustained loading using governing stress of Approach 1 (see footnote to Table B-7)

T_2 = Allowable time for sustained loading using governing stress of Approach 2 (see footnote to Table B-7).

TABLE B-1
CALCULATION OF PRIMARY STRESS
USAGE FACTORS

ID	σ_H	$\sigma_H + \frac{\sigma_B}{1.25}$	HP	T_H	t/T_H	T_{HB}	t/T_{HB}
A	7.9	22.5	1	$>10^6$	0	<1	$>10^5$
	5.1	14.4	2	$>10^6$	0	3×10^5	2.54
					0		$>10^5$
B	5.4	18.2	1	$>10^6$	0	400	295
	3.5	11.7	2	$>10^6$	0	$>10^6$	0
					0		295
C	7.5	16.8	1	$>10^6$	0	4000	29.5
	4.8	10.8	2	$>10^6$	0	$>10^6$	0
					0		29.5
D	16.6	28.4	1	5×10^3	23.6	<1	$>10^5$
	10.6	18.2	2	$>10^6$	0	10^4	11.8
					23.6		$>10^5$
E	22.2	32.8	1	<1	$>10^5$	<1	$>10^5$
	14.2	21.0	2	3×10^5	0.4	<1	$>10^5$
					$>10^5$		$>10^5$
F	23.2	36.7	1	<1	$>10^5$	<1	$>10^5$
	14.9	23.6	2	2×10^5	0.6	<1	$>10^5$
					$>10^5$		$>10^5$
G	1.0	13.2	1	$>10^6$	0	7×10^4	1.68
	0.6	8.4	2	$>10^6$	0	$>10^6$	0
					0		1.68
H	0.8	10.6	1	$>10^6$	0	3×10^5	0.39
	0.5	6.8	2	$>10^6$	0	$>10^6$	0
					0		0.39
I	0.4	8.8	1	$>10^6$	0	10^6	0.11
	0.2	5.6	2	$>10^6$	0	$>10^6$	0
					0		0.11
J	1.3	7.9	1	$>10^6$	0	$>10^6$	0
	0.8	5.1	2	$>10^6$	0	$>10^6$	0
					0		0
L	1.2	7.8	1	$>10^6$	0	$>10^6$	0
	0.8	5.0	2	$>10^6$	0	$>10^6$	0
					0		0

N112TI310002

TABLE B-2
HOLD PERIOD DEFINITIONS*

Joint Size	HP	Sustained Pressure (psig)	Temperature (°F)	t
42 in.	1	190	1050	1.18×10^5
	2	122	990	1.18×10^5
56 in.	1	0	1050	1.18×10^5
	2	0.5	990	1.18×10^5

*Refer to Section VI-B for selection of these hold periods.

TABLE B-3
HOLD PERIOD STRESS SCALE FACTORS*

Joint Size	HP	Pressure Stress	Pressure Stress Range	Motion Stress	Motion Stress Range
42 in.	1	1.000	0.358	1.00	0.12
	2	0.642		0.88	
56 in.	1	0.000	0.033	1.00	0.12
	2	0.033		0.88	

*Factors by which stress values on the computer-generated designs are multiplied to give sustained stresses and stress ranges.

N112TI310002

TABLE B-4
SIGNIFICANT DESIGN LOADING CYCLES*

42-in. Joint						
LC	Temperature Cycle (°F)	Pressure Cycle (psi)	Stress Value Scale Factors†		n	T _{max}
			Motion Stress	Pressure Stress		
1	50-1050-50	0-190-0	2.00	1.00	13	1,050
2	400- 870-400	2- 30-2	0.94	0.15	80	870
3	400-1050-400	2-190-2	1.30	0.99	60	1,050
4	600-1050-600	2-190-2	0.90	0.99	260	1,050
5	870-1050-870	30-190-30	0.36	0.84	12,000	1,050
6	990-1050-990	122-190-122	0.12	0.36	55,000	1,050
56-in. Joint						
1	50-1050-50	2-0-2	2.00	0.13	13	1,050
2	400- 870-400	2-1-2	0.94	0.07	80	870
3	400-1050-400	2-0-2	1.30	0.13	60	1,050
4	600-1050-600	2-0-2	0.90	0.13	260	1,050
5	870-1050-870	1-0-1	0.36	0.07	12,000	1,050
6	990-1050-990	0.5-0-0.5	0.12	0.03	55,000	1,050

*Refer to Section II-C for selection of these loading cycles.

†Factors by which stress values in computer-generated designs are multiplied to give cyclic stress ranges.

N112TI310002

TABLE B-5
NO HOLD-TIME FATIGUE DAMAGE
FRACTION CALCULATIONS

ID	LC	σ_b	$(\sigma_H + \sigma_B)$	$\Delta\sigma_b$	$\Delta(\sigma_H + \sigma_B)$	$\Delta\sigma$	$\Delta\epsilon$	N	n	n/N
B	1	23.7	21.4	47.4	21.4	68.8	3.11	1100	13	0.0118
	2			22.3	3.2	25.5	1.09	1.1×10^5	80	0.0007
	3			30.8	21.2	52.0	2.35	3000	60	0.0200
	4			21.3	21.2	42.5	1.92	6400	260	0.0406
	5			8.5	18.0	26.5	1.20	6×10^4	12,000	0.2000
	6			2.8	7.7	10.5	0.48	$>10^6$	55,000	≈ 0
										0.273
C	1	22.2	19.2	44.4	19.2	63.6	2.88	1400	13	0.0092
	2			20.9	2.9	23.8	1.01	1.7×10^5	80	0.0004
	3			28.9	19.0	47.9	2.17	4000	60	0.0150
	4			20.0	19.0	39.0	1.76	9500	260	0.0273
	5			8.0	16.1	24.1	1.09	1.1×10^5	12,000	0.1090
	6			2.7	6.9	9.6	0.43	$>10^6$	55,000	≈ 0
										0.161
G	1	18.6	16.2	37.2	2.1	39.3	1.77	9000	13	0.0014
	2			17.5	1.1	18.6	0.79	$>10^6$	80	≈ 0
	3			24.2	2.1	26.3	1.19	6.6×10^4	60	0.0009
	4			16.7	2.1	18.8	0.85	5×10^5	260	0.0005
	5			6.7	1.1	7.8	0.35	$>10^6$	12,000	≈ 0
	6			2.2	0.5	2.7	0.12	$>10^6$	55,000	≈ 0
										0.003
H	1	16.1	13.0	32.2	1.7	33.9	1.53	1.57×10^4	13	0.0008
	2			15.1	0.9	16.0	0.72	$>10^6$	80	≈ 0
	3			20.9	1.7	22.6	1.02	1.6×10^5	60	0.0003
	4			14.5	1.7	16.2	0.73	$>10^6$	260	≈ 0
	5			5.8	0.9	6.7	0.30	$>10^6$	12,000	≈ 0
	6			1.9	0.4	2.3	0.10	$>10^6$	55,000	≈ 0
										0.001
I	1	11.1	10.9	22.2	1.4	23.6	1.06	1.3×10^5	13	0.0001
	2			10.4	0.8	11.2	0.48	$>10^6$	80	≈ 0
	3			14.4	1.4	15.8	0.71	$>10^6$	60	≈ 0
	4			10.0	1.4	11.4	0.51	$>10^6$	260	≈ 0
	5			4.0	0.8	4.8	0.21	$>10^6$	12,000	≈ 0
	6			1.3	0.3	1.6	0.07	$>10^6$	55,000	≈ 0
										≈ 0
J	1	25.3	9.6	50.6	1.2	51.8	2.34	2100	13	0.0061
	2			23.8	0.7	24.5	1.04	1.4×10^5	80	0.0005
	3			32.9	1.2	34.1	1.54	1.7×10^4	60	0.0035
	4			22.8	1.2	24.0	1.08	1.1×10^5	260	0.0023
	5			9.1	0.7	9.8	0.44	$>10^6$	12,000	≈ 0
	6			3.0	0.3	3.3	0.14	$>10^6$	55,000	≈ 0
										0.012
L	1	15.4	9.5	30.8	1.2	32.0	1.44	2.3×10^4	13	0.0005
	2			14.5	0.7	15.2	0.64	$>10^6$	80	≈ 0
	3			20.0	1.2	21.2	0.95	2.5×10^5	60	0.0002
	4			13.9	1.2	15.1	0.68	$>10^6$	260	≈ 0
	5			5.5	0.7	6.2	0.28	$>10^5$	12,000	≈ 0
	6			1.8	0.3	2.1	0.09	$>10^6$	55,000	≈ 0
										≈ 0

N112TI310002

TABLE B-6
HOLD TIME FATIGUE DAMAGE FRACTION CALCULATIONS
(Sheet 1 of 2)

ID	LC	$\Delta\epsilon$	N	n	n / N
B	1	3.11	27	13	0.4814
	2	1.09	700	80	0.1142
	3	2.35	45	60	1.3333
	4	1.92	70	260	3.7142
	5	1.20	430	12,000	27.9069
	6	0.48	2.9×10^5	55,000	0.1896
					33.7
C	1	2.88	30	13	0.4333
	2	1.01	1000	80	0.0800
	3	2.17	53	60	1.1320
	4	1.76	86	260	3.0232
	5	1.09	700	12,000	17.1428
	6	0.43	7×10^5	55,000	0.0785
					21.9
G	1	1.77	86	13	0.1511
	2	0.79	6600	80	0.0121
	3	1.19	460	60	0.1304
	4	0.85	3700	260	0.0702
	5	0.35	$>10^6$	12,000	≈ 0
	6	0.12	$>10^6$	55,000	≈ 0
					0.434
H	1	1.53	140	13	0.0928
	2	0.72	1.4×10^4	80	0.0057
	3	1.02	920	60	0.0652
	4	0.73	1.25×10^4	260	0.0208
	5	0.30	$>10^6$	12,000	≈ 0
	6	0.10	$>10^6$	55,000	≈ 0
					0.185

N112TI310002

TABLE B-6
HOLD TIME FATIGUE DAMAGE FRACTION CALCULATIONS
(Sheet 2 of 2)

ID	LC	$\Delta\epsilon$	N	n	n/N
I	1	1.06	800	13	0.0162
	2	0.48	$>10^6$	80	0
	3	0.71	1.5×10^4	60	0.0040
	4	0.51	1.8×10^5	260	0.0014
	5	0.21	$>10^6$	12,000	≈ 0
	6	0.07	$>10^6$	55,000	≈ 0
					<hr/> 0.0216
J	1	2.34	46	13	0.2826
	2	1.04	860	80	0.0930
	3	1.54	140	60	0.4285
	4	1.08	720	260	0.3611
	5	0.44	6×10^5	12,000	0.0333
	6	0.14	$>10^6$	55,000	≈ 0
					<hr/> 1.1985
L	1	1.44	180	13	0.0722
	2	0.64	3.3×10^4	80	0.0024
	3	0.95	1500	60	0.0400
	4	0.68	2×10^4	260	0.0130
	5	0.28	$>10^6$	12,000	≈ 0
	6	0.09	$>10^6$	55,000	≈ 0
					<hr/> 0.1276

N112TI310002

TABLE B-7
CREEP DAMAGE FRACTION CALCULATIONS

A. Stress Calculations							
ID	Computer Stresses			HP	σ_H	$(\sigma_H + \sigma_B)_R$	$(\sigma_H + \sigma_B + \sigma_b)_R$
	σ_H	σ_B	σ_b				
B	5.4	16.0	23.7	1	5.4	7.7	10.5
				2	3.5		
C	7.5	11.7	22.2	1	7.5	6.9	9.6
				2	4.8		
G	1.0	15.2	18.6	1	0	0.53	2.76
				2	0.03		
H	0.8	12.2	16.1	1	0	0.43	2.36
				2	0.03		
I	0.4	10.5	11.1	1	0	0.36	1.69
				2	0.01		
J	1.3	8.3	25.3	1	0	0.32	3.36
				2	0.04		
L	1.2	8.3	15.4	1	0	0.31	2.16
				2	0.04		
B. Damage Fraction Calculations							
ID	HP	Governing Stress*		T_1	T_2	t/T_1	t/T_2
		Approach 1	Approach 2				
B	1	10.3	11.8	$>10^6$	$>10^6$	0	0
	2	8.2	9.7	$>10^6$	$>10^6$	0	0
C	1	12.2	13.7	$>10^6$	$>10^6$	0	0.12
	2	9.2	10.7	$>10^6$	$>10^6$	0	0
G-L		1.9 max.		negligible damage		0	0.12

*Computed as the minimum of:

$$\text{Approach 1: } \frac{\sigma_H + (\sigma_H + \sigma_B)_R}{2} / 0.9$$

$$\text{Approach 2: } \sigma_H + \frac{(\sigma_H + \sigma_B + \sigma_b)_R}{2} / 0.9$$

where

σ_H , $(\sigma_H + \sigma_B)_R$ and $(\sigma_H + \sigma_B + \sigma_b)_R$ are defined in Part A (Note - yield stress did not govern any of these designs)

N112TI310002

B-10

APPENDIX C
DESIGN AND COMPUTATION SHEETS FOR ALTERNATE FLEXIBLE
JOINT DESIGNS

This Appendix presents the calculational analyses for the 42-in. double-series articulating bellows (BB and CC) and the 42-in. six-ply double-series articulating bellows (RR, with reinforcing rings).

The nomenclature in this Appendix, unless otherwise referenced, is as follows:

- LC = Loading cycle identifier (see Subsection II-C and Table B-4)
- ID = Bellows geometry identifier (see Table VI-2, Section VI)
- σ_b = Motion bending stress (ksi)
- σ_H = Pressure hoop stress (ksi)
- σ_B = Pressure bending stress (ksi)
- $\Delta\sigma_b$ = Motion bending stress range (ksi)
- $\Delta(\sigma_H + \sigma_B)$ = Pressure stress range (ksi)
- $\Delta\sigma$ = Total stress range (ksi)
- $\Delta\epsilon$ = Total strain range (in./in. $\times 10^{-3}$)
- N = Allowable cycles for a particular loading cycle
- n = Number of design cycles for a particular loading cycle
- T_H = Allowable hold time for pressure hoop stress for particular hold period (hr)
- T_{HB} = Allowable hold time for pressure hoop stress plus bending stress for particular hold period (hr)
- t = Design hold time for particular hold period (hr)
- HP = Hold period identifier (from Table B-2, Appendix B)

N112TI310002

TABLE C-1
DOUBLE SERIES ARTICULATING BELLOWS SCALE FACTORS*
FOR FATIGUE EVALUATION

LC	42-in. Joint	
	Motion Stress Range Scale Factor	Pressure Stress Range Scale Factor
1	0.71	1.00
2	0.34	0.15
3	0.46	0.99
4	0.32	0.99
5	0.13	0.84
6	0.04	0.36

*These scale factors were used to convert computer-generated stress values to the stress ranges for each load cycle. The motion stress range scale factors are based on the motion requirements of Subsection II-C for double-series articulating joints. The pressure stress range scale factors are obtained from Appendix B. The load cycles are defined in Subsection II-C.

TABLE C-2
NO HOLD-TIME FATIGUE DAMAGE FRACTION CALCULATIONS FOR
42-in. DOUBLE-SERIES ARTICULATING BELLOWS
DESIGNS BB AND CC

ID	LC	σ_b	$(\sigma_H + \sigma_B)$	$\Delta\sigma_b$	$\Delta(\sigma_H + \sigma_B)$	$\Delta\sigma$	$\Delta\epsilon$	N	n	n/N
BB	1	23.7	21.4	16.8	21.4	38.2	1.73	10^4	13	0.0013
	2	23.7	21.4	8.1	3.2	11.3	0.48	$>10^6$	80	≈ 0.0
	3	23.7	21.4	10.9	21.2	32.1	1.45	2.3×10^4	60	0.0026
	4	23.7	21.4	7.6	21.2	28.8	1.30	4×10^4	260	0.0065
	5	23.7	21.4	3.1	18.0	21.1	0.95	2.5×10^5	12,000	0.0480
	6	23.7	21.4	0.9	7.7	8.6	0.39	$>10^6$	55,000	≈ 0.0
CC	1	22.2	19.2	15.8	19.2	35.0	1.58	1.5×10^4	13	0.0008
	2	22.2	19.2	7.5	2.9	10.4	0.44	$>10^6$	80	≈ 0.0
	3	22.2	19.2	10.2	19.0	29.2	1.32	3.5×10^4	60	0.0017
	4	22.2	19.2	7.1	19.0	16.1	0.73	10^6	260	0.0002
	5	22.2	19.2	2.9	16.1	19.0	0.86	4.9×10^5	12,000	0.0244
	6	22.2	19.2	0.9	6.9	7.8	0.35	$>10^6$	55,000	≈ 0.0
										0.027

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TABLE C-3
HOLD-TIME FATIGUE DAMAGE FRACTION CALCULATIONS FOR
42-in. DOUBLE SERIES ARTICULATING BELLOWS
DESIGNS BB AND CC

ID	LC	$\Delta\epsilon$	N	n	n/N
BB	1	1.73	88	13	0.1477
	2	0.48	2.9×10^5	80	0.0002
	3	1.45	180	60	0.3333
	4	1.30	300	260	0.8666
	5	0.95	1500	12,000	8.0000
	6	0.39	$>10^6$	55,000	<u>≈ 0.0</u> 9.35
CC	1	1.58	120	13	0.1083
	2	0.44	6×10^5	80	0.0001
	3	1.32	280	60	0.2142
	4	0.73	1.25×10^4	260	0.0208
	5	0.86	3500	12,000	3.4285
	6	0.35	$>10^6$	55,000	<u>≈ 0.0</u> 3.77

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TABLE C-4
STRESS AND STABILITY CALCULATIONS FOR 42-in. SIX-PLY
DOUBLE-SERIES ARTICULATING BELLOWS RR WITH
REINFORCING RINGS (SEE SUBSECTION III-B
FOR NOMENCLATURE)
(Sheet 1 of 2)

$$H = Pd_{pq} = (190)(47)(2.0) = 17,860$$

$$P = 190 \text{ psi}$$

$$d_p = d + w = 45 + 2 = 47 \text{ in.}$$

$$q = 2.0 \text{ in.}$$

$$A_b = (0.571q + 2w)t_p n = [(0.571)(2) + (2)(2)](0.0489)(6) = 1.5086$$

$$w = 2.0$$

$$t_p = t \left(\frac{d}{d_p} \right)^{1/2} = 0.05 \left(\frac{45}{47} \right)^{1/2} = 0.0489 \text{ in.}$$

$$n = 6$$

$$t = 0.05 \text{ in.}$$

$$d = 45 \text{ in.}$$

$$R = \frac{A_b}{A_r} = \frac{1.5086}{2.2698} = 0.6646$$

$$A_r = \frac{\pi}{4} \left(\frac{q}{2} - nt \right)^2 = \frac{\pi}{4} \left[\frac{2}{1} - (6)(0.05) \right]^2 = 2.2698 \text{ in.}^2$$

$$E_b = 22.1 \times 10^6 \text{ psi}$$

$$e = \frac{\theta d_p}{2N} = \frac{(1.34)(2\pi)(47)}{(360)(2)(23)} = 0.0238 \text{ in.}$$

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TABLE C-4
STRESS AND STABILITY CALCULATIONS FOR 42 in. SIX-PLY
DOUBLE-SERIES ARTICULATING BELLOWS RR WITH
REINFORCING RINGS (SEE SUBSECTION III-B
FOR NOMENCLATURE)
(Sheet 2 of 2)

$$\theta = 1.34^\circ$$

$$N = 23$$

$$C_p = 0.64$$

$$C_d = 1.75$$

$$C_f = 1.37$$

$$\sigma_H = \frac{H}{2A_b} \left(\frac{R}{R+1} \right) = \frac{17,860}{(2)(1.5086)} \left(\frac{0.6646}{1.6646} \right) = 2360 \text{ psi}$$

$$\sigma_B = \frac{P}{2n} \left(\frac{w - 0.3q}{t_p} \right)^2 C_p = \frac{190}{(2)(6)} \left(\frac{2 - (0.3)(2)}{0.0489} \right)^2 (0.64) = 8310 \text{ psi}$$

$$\sigma_b = \frac{5E_b t_p^e}{3(w - 0.3q)^2 C_d} = \frac{(5)(22.1)(10^6)(0.0489)(0.0238)}{3[2.0 - (0.3)(2.0)]^2 1.75} = 12,500 \text{ psi}$$

$$\therefore \sigma_H = 2360, \quad \sigma_H + \sigma_B = 10,670, \quad \sigma_H + \sigma_B + \sigma_b = 23,170 \text{ psi}$$

$$p_c = \left(\frac{3.4\pi}{N^2 q} \right) \left[\frac{d_p E_b t_p^3}{(w - 0.3q) C_f} \right] = \left[\frac{3.4\pi}{(23)^2 (2)} \right] \left[\frac{(47)(22.1)(10^6)(0.0489)^3 (6)}{[2 - (0.3)(2)]^3} \right]$$

$$= 1957 \text{ psi}$$

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TABLE C-5
 STATIC STRESS AND TIME-DEPENDENT SCREENING RULES APPLIED
 TO 42-in. SIX-PLY DOUBLE-SERIES ARTICULATING BELLOWS
 RR WITH REINFORCING RINGS

1. Required Allowable Primary Membrane Stress - 7,723 psi

2. Time-Dependent Primary Stress Usage Factors

σ_H	$\sigma_H + \frac{\sigma_B}{1.25}$	HP	T_H	t/T_H	T_{HB}	t/T_{HB}
2360	9008	1	$>10^6$	0	1.113×10^5	1.00
1515	5783	2		<u>0</u>	$>10^6$	<u>0</u>
				0		1.00

3. Hold Time Fatigue Damage Fraction

LC	σ_b	$(\sigma_H + \sigma_B)$	$\Delta\sigma_b$	$\Delta(\sigma_H + \sigma_B)$	$\Delta\sigma$	$\Delta\epsilon$	N	n	n/N
1	12,500	10,670	25,000	10,670	35,670	1.61	110	13	0.1181
2	12,500	10,670	11,750	1,600	13,350	0.64	3.2×10^4	80	0.0025
3	12,500	10,670	16,250	10,560	26,810	1.21	410	60	0.1463
4	12,500	10,670	11,250	10,560	21,810	0.99	1000	260	0.2600
5	12,500	10,670	4,500	8,960	13,460	0.62	45,000	12,000	0.2666
6	12,500	10,670	1,500	3,840	5,340	0.24	$>10^6$	55,000	<u>0</u>
									0.7935

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