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DESIGN CRITERIA
(CHEMICAL PROPULSION)

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LIQUID ROCKET LINES, BELLOWS, FLEXIBLE HOSES, AND FILTERS

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

FOREWORD

NASA experience has indicated a need for uniform criteria for the design of space vehicles. Accordingly, criteria are being developed in the following areas of technology:

Environment
Structures
Guidance and Control
Chemical Propulsion

Individual components of this work will be issued as separate monographs as soon as they are completed. This document, part of the series on Chemical Propulsion, is one such monograph. A list of all monographs issued prior to this one can be found on the final pages of this document.

These monographs are to be regarded as guides to design and not as NASA requirements, except as may be specified in formal project specifications. It is expected, however, that these documents, revised as experience may indicate to be desirable, eventually will provide uniform design practices for NASA space vehicles.

This monograph, "Liquid Rocket Lines, Bellows, Flexible Hoses, and Filters", was prepared under the direction of Howard W. Douglass, Chief, Design Criteria Office, Lewis Research Center; project management was by M. Murray Bailey. The monograph was written by C. M. Daniels, Rocketdyne Division, Rockwell International Corporation and was edited by Russell B. Keller, Jr. of Lewis. Significant contributions to the text were made by T. Nelson, J. R. Rollins, and L. Sack of Rocketdyne Division, Rockwell International Corporation. To assure technical accuracy of this document, scientists and engineers throughout the technical community participated in interviews, consultations, and critical review of the text. In particular, J. W. Akkerman of the Lyndon B. Johnson Space Center; K. W. Baud of the Lewis Research Center; R. H. Henry of Space Division, Rockwell International Corporation; H. S. Hillbrath of The Boeing Company; P. L. Muller of Marshall Space Flight Center; F. D. Sullivan of Aerojet Liquid Rocket Company; and T. M. Weathers of Systems Group, TRW, Inc. individually and collectively reviewed the monograph in detail.

Comments concerning the technical content of this monograph will be welcomed by the National Aeronautics and Space Administration, Lewis Research Center (Design Criteria Office), Cleveland, Ohio 44135.

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GUIDE TO THE USE OF THIS MONOGRAPH

The purpose of this monograph is to organize and present, for effective use in design, the significant experience and knowledge accumulated in development and operational programs to date. It reviews and assesses current design practices, and from them establishes firm guidance for achieving greater consistency in design, increased reliability in the end product, and greater efficiency in the design effort. The monograph is organized into two major sections that are preceded by a brief introduction and complemented by a set of references.

The State of the Art, section 2, reviews and discusses the total design problem, and identifies which design elements are involved in successful design. It describes succinctly the current technology pertaining to these elements. When detailed information is required, the best available references are cited. This section serves as a survey of the subject that provides background material and prepares a proper technological base for the *Design Criteria* and Recommended Practices.

The *Design Criteria,* shown in italics in section 3, state clearly and briefly what rule, guide, limitation, or standard must be imposed on each essential design element to assure successful design. The *Design Criteria* can serve effectively as a checklist of rules for the project manager to use in guiding a design or in assessing its adequacy.

The Recommended Practices, also in section 3, state how to satisfy each of the criteria. Whenever possible, the best procedure is described; when this cannot be done concisely, appropriate references are provided. The Recommended Practices, in conjunction with the *Design Criteria,* provide positive guidance to the practicing designer on how to achieve successful design.

Both sections have been organized into decimally numbered subsections so that the subjects within similarly numbered subsections correspond from section to section. The format for the Contents displays this continuity of subject in such a way that a particular aspect of design can be followed through both sections as a discrete subject.

The design criteria monograph is not intended to be a design handbook, a set of specifications, or a design manual. It is a summary and a systematic ordering of the large and loosely organized body of existing successful design techniques and practices. Its value and its merit should be judged on how effectively it makes that material available to and useful to the designer.

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LIQUID ROCKET LINES, BELLOWS, FLEXIBLE HOSES, AND FILTERS

1. INTRODUCTION

A liquid-propellant rocket engine and the vehicle for which it provides a propulsive force represent an assemblage of fluid-flow components that must be interconnected if the components are to perform their respective functions. These interconnections are provided by the lines, bellows, and flexible hoses used on a typical rocket propulsion system. The fluids in a given system are maintained at a required level of cleanliness by filters that remove contaminant material.

A line (or duct) is an enclosed leak-tight passageway that conveys fluid from one fluid-system component to another. (The terms "line" and "duct" are used interchangeably in the aerospace industry and will be so used in this monograph.) A bellows is a thin-wall circumferentially corrugated cylinder that when integrated into a line can accommodate line movement through deflection of the corrugations (convolutions). A bellows joint normally consists of a bellows and a restraint linkage, but it may be only a free bellows. Flexible hose consists of a corrugated (convoluted) innercore restrained by an outer sheath or wire braid. A filter is a device that removes contaminants from a fluid system by trapping particulate matter within or on the surface of porous material; criticality of most rocket engine components requires filtration to some degree for almost all control fluids prior to their entry into valves, actuators, and other fluid-system components.

The successful design of line or duct assemblies and their components is based on many considerations and factors. Some considerations are quite basic and are well grounded in theory, while others are more subtle and have been learned through many failure analyses and experiments.

Flexibility is designed into a line assembly so that it can accommodate deflections imposed by various conditions such as thermal expansions and contractions, installation misalignments, operational structural deflections, and thrust-vector gimballing. The most common method for providing flexibility is to incorporate in the line a bellows or a section of flexible hose or combinations thereof. A second method is to provide a hard line (no hose or bellows) with sufficient flexibility to absorb the imposed deflections, flexibility being achieved by special attention to line configuration, location of elbows, length-to-diameter ratio, and line material properties. In some instances (e.g., the Titan I engines), rotating joints with dynamic seals have been used, but to a much lesser extent than bellows joints or "flexible" hard lines.

The major problems in line assemblies have involved the details of the flexible-joint elements utilized. Less frequent problems have involved the entire assembly and include mechanical vibration, corrosion, contamination, thermal-cycling fatigue, static-seal leakage, handling damage, and mismade parts. The major problems with the flexible elements, in addition to those just listed, have been fatigue failures (from flexure and mechanical or flow-induced vibration), buckling-instability failure, and failure of the duct-to-bellows attachment joint. The major problems in filter design have been determining the amount of built-in, system-generated, and environmental contaminants to be filtered; contaminant capacity; flow versus pressure drop for filter elements; and the physical size of filter required to satisfy the system requirements.

This monograph begins with a discussion of the line assembly that treats the key design elements in the assembly in the logical, chronological order in which a designer would proceed. Bellows (including linkages, liners, and associated components), flexible hoses, and filters then are treated in order. Some of the design considerations in these areas overlap. For example, similar considerations apply to the design of line assemblies with flexible joints and to the design of a flexible-hose assembly; also, considerations for design of a bellows in a flexible joint are similar to those for a bellows in a flexible-hose innercore.

Because the materials-selection factors, handling-protection devices, test requirements, and test instrumentation are very similar for lines, bellows, and flexible hoses, this information is presented only once in the section "LINE ASSEMBLY." Filters, while not intimately related to ducting, are a critical part of a rocket engine fluid system; they are discussed separately and last in the monograph.

2. STATE OF THE ART

Lines, bellows, and flexible hoses. – Table I* presents the chief design features of lines bellows, and hoses in typical uses on representative operational and advanced-development engines and vehicles. In the design and application of these components, many problems were solved; some problems still exist and have at best only partial solutions.

Flexible lines are more prevalent on large liquid rocket engines (booster or upper-stage main propulsion) than on small engines (attitude control or reaction control). Small engines seldom have exterior lines except for propellant feedlines. Little flexibility is required in these lines because the engines usually are not gimballed and utilize storable** (noncryogenic) propellants. The state of the art of rocket engine flexible lines thus is based primarily on experience with large engines. Figure 1 shows the line, bellows, and hose assemblies on a typical large pump-fed engine (J-2).

Early launch vehicles, as characterized by the Vanguard and Redstone, had relatively low engine thrust and operating pressures. The flexible-line design problems were solvable within the state of the art. Free bellows, restrained only by their installation attachment points, were used for achieving flexibility in the propellant feed systems. The materials used for both bellows and feedlines were stainless steels of the 18-percent-chromium, 8-percent-nickel family (18-8 CRES).

Propulsion systems for later launch vehicles (e.g., Jupiter, Thor, Atlas, and Titan) included gimballed engines with greater operating pressures and larger line sizes than those on the Redstone or Vanguard. Flexible lines were more sophisticated and required considerably more development; 18-8 CRES remained as the predominant type of material, however.

The engines of the Saturn vehicles and the associated propellant feed systems presented severe design requirements in flexible ducting. Advances in the state of the art were necessary to meet those requirements. Line sizes, operating pressures, flow velocities, and gimballing life – all far exceeded anything previously done. Higher strength materials were necessary to minimize structural weights and to provide adequate fatigue life. The hardware, being man-rated, had to be extremely high in reliability; reliability was proved through many component, engine, and unmanned-flight tests.

The Saturn vehicle systems made extensive use of the pressure-volume-compensator (PVC) type of duct assembly. This design concept was not used on any engines or vehicles prior to the Saturn. Tension-type systems, which include PVC ducts, are the predominant systems on the Saturn. The free-bellows compression system also was frequently used; examples are

*Factors for converting U.S. customary units in the table to the International System of Units (SI units) are given in Appendix A.

**Terms and symbols, materials, and abbreviations are defined or identified in Appendix B.

Table I. – Chief Design Features of Line Assemblies in Typical Uses on Operational and Advanced-Development Engines and Vehicles

Program, Engine or Airframe Model, Date	Part description or function	Fluid	Temperature range, °F	Operating pressure range, psig	Line ID, in.	Materials			Type of bellows restraint
						Line	Bellows	Restraint	
Saturn V F-1 1959	Misc. small diam. flex hoses: Tank pressurization, hydraulic system (RP-1), bleed, purge, bypass and drain functions	LOX	-297	0 to 3000	0.75 to 1.00	321 CRES Inconel 718	321 CRES	321 CRES	Braid
		RP-1	35 to 100	0 to 3125	0.25 to 2.25	321 CRES A286	321 CRES	321 CRES	Braid
		He	-350 to 370	0 to 3000	0.25 to 0.75	321 CRES Inconel 718	321 CRES	321 CRES	Braid
		GN_2	-240 to 400	0 to 2000	0.5 to 0.75	321 CRES A286	321 CRES	321 CRES	Braid
		Exhaust gas	1000 to 1600	0 to 3125	0.5	321 CRES	321 CRES	321 CRES	Braid
	Bellows: Propellant feed, hydraulic system (RP-1), and drain lines	LOX	-297	0 to 2360	1.25 to 17	321 CRES	321 CRES Inconel 718 Inconel X-750	321 CRES Inconel 718	External gimbal (liner)
		RP-1	35 to 100	0 to 3000	1.37 to 12	321 CRES	321 CRES Inconel 718	321 CRES Inconel 718	External gimbal (liner)
	Bellows: Tank pressurization	GOX	400 to 500	0 to 1850	1.25	321 CRES	Inconel 718	Inconel 718	External gimbal
		He	-350 to 370	0 to 400	1.25 to 1.50	321 CRES	Inconel 718	Inconel 718	External gimbal
	Bellows: Turbine exhaust system	Exhaust gas	1000 to 1200	0 to 65	27	Hastelloy C 347 CRES	Hastelloy C	---	Compression system
	Bellows: Pump inlets	LOX	- 297	0 to 150	18	Inconel 718 321 CRES A286	Inconel 718 Hastelloy C	321 CRES Inconel 718	Pressure-volume compensating external gimbal (liner)
	Bellows: Pump inlets	RP-1	35 to 100	0 to 54	12 to 17.4	Inconel 718	Inconel 718	321 Inconel 718	Pressure-volume compensating external gimbal (liner)

Saturn V J-2 1961	Misc. small diam. flex lines: Tank pressurization, bleed, purge, propellant feed system drain, and start system	LOX	297	0 to 1400	0.375 to 1.5	321 CRES	321 CRES	321 CRES	Braid
		GOX	100	0 to 84	0.5 to 1.25	321 CRES	321 CRES	321 CRES	Braid
		He	420 to 200	0 to 3200	0.25 to 0.75	321 CRES	321 CRES	321 CRES	Braid
		LH_2	-420	0 to 1340	0.25 to 1.5	321 CRES	321 CRES	321 CRES	Braid
		GH_2	-250	0 to 1500	0.5 to 2.0	321 CRES Hastelloy C	321 CRES Hastelloy C	321 CRES Hastelloy C	Braid
		LN_2	-290	0 to 225	0.375	321 CRES	321 CRES	321 CRES	Braid
		GN_2	140	165 to 1000	0.375	321 CRES	321 CRES	321 CRES	Braid
	Bellows: Pump discharge ducts	LOX	-297	0 to 1390	4.00	310 CRES Inconel 718	321 CRES	Inconel 718	Thrust-balancing linkage
		LH_2	-420	0 to 1350	4.00	310 CRES Inconel 718	321 CRES	Inconel 718	Thrust-balancing linkage
	Bellows: Pump inlet ducts	LOX	-297	0 to 50	8.00	321 CRES Inconel 718	Inconel 718 321 CRES	A286	External-stabilizer
		LH_2	-423	0 to 50	8.00	321 CRES	Inconel 718 321 CRES	A286	External-stabilizer
	Bellows: Turbine exhaust system	LO_2/LH_2 combustion products	0 to 1034	0 to 126	4 to 12	321 CRES Hastelloy C	321 CRES Hastelloy C	321 CRES A286	Internal linkage (liner)
Saturn V S-II stage (feed systems for J-2 engine) 1962	Braided hoses: purge, pressurization, fill and drain functions	GH_2	-350 to +180	0 to 3000	0.29 to 1.00	321 CRES	321 CRES	321 CRES	Braid
		He	-395 to +180	0 to 3000	0.25 to 1.00	321 CRES	321 CRES	321 CRES	Braid
	Bellows: Propellant feed	LOX	-297	88	8.00	321 CRES	Inconel 718	Inconel 718	External gimbal
		LH_2	-423	88	8.00	321 CRES	Inconel 718	Inconel 718	External gimbal
	Bellows: LOX tank pressurization	GOX	-290 to +440	0 to 1100	1.50	Inconel 718	Inconel 718	Inconel 718	External gimbal
		GOX	-290 to +440	0 to 1100	3.00	Inconel 718	Inconel 718	Inconel 718	External gimbal
		GOX	-290 to +440	0 to 115	3.00	321 CRES	19–9 DL	321 CRES	External gimbal
	Bellows: LH_2 tank pressurization	GH_2	-320 to +70	0 to 860	1.50	Inconel 718	Inconel 718	Inconel 718	External gimbal
		GH_2	-320 to +70	0 to 860	3.00	Inconel 718	Inconel 718	Inconel 718	External gimbal
		GH_2	-320 to +70	0 to 100	3.00	Inconel 718	Incone 718	Inconel 718	External gimbal; free lined bellows
	Bellows: LOX recirculation	LOX	-297	45	1.50 to 3.00	321 CRES	321 CRES	321 CRES	External gimbal

(continued)

Table I. – Chief Design Features of Line Assemblies in Typical Uses on Operational and Advanced-Development Engines and Vehicles (contd.)

Program, Engine or Airframe Model, Date	Part description or function	Fluid	Temperature range, °F	Operating pressure range, psig	Line ID, in.	Materials: Line	Materials: Bellows	Materials: Restraint	Type of bellows restraint
Saturn V S-II stage (feed systems for J-2 engine) 1962	Bellows: LH_2 recirculation	LH_2	-423	44	1.50 to 3.00	321 CRES	321 CRES	321 CRES	External gimbal
	Bellows: LOX tank vent	GOX	-290	6	7.00	321 CRES	Inconel 718	321 CRES	External gimbal
	Bellows: LH_2 tank vent	GH_2	-417	6	7.00	321 CRES	Inconel 718	321 CRES	External gimbal
M-1* 1964	Pump inlet line	LH_2	-420 to +65	0 to 150	19.5	321 CRES	321 CRES	---	No restraint
	Pump discharge line	LH_2	-420 to +65	0 to 2000	9.62	---	Inconel 718	18% nickel maraging steel, gold plate, ball and socket; Inconel 718 tripod	Internal tripod with ball and socket bearing
	Bellows[a]	GH_2	65 to 1000	0 to 1000	4.90	---	Inconel 718	Haynes Star J, ball; Haynes 6B, socket; Inconel 718 tripod	Internal tripod with ball and socket bearing
	Bellows[a]	LH_2	-420 to +65	0 to 2000	3.25	347 CRES	321 CRES	18% nickel, maraging steel, nickel plate, ball & socket; Inconel 718 tripod	Internal tripod with ball and socket bearing
Apollo Service Module Engine, 1965	Fuel feedline	A-50	-20 to +140	0 to 240	2.95	321 CRES	321 CRES	321 CRES struts; 17-4 PH tie rods	Internal strut and "U" tie rods
	Engine inlet system	A-50	- 20 to +140	0 to 240	2.5	321 CRES	321 CRES	321 CRES	3 external tie rods
		N_2O_4	-20 to +140	0 to 240	3.00	321 CRES	321 CRES	321 CRES	3 external tie rods
	Oxidizer feedline	N_2O_4	-20 to +140	0 to 240	2.95	321 CRES	321 CRES	321 CRES struts; 17-4 PH tie rods	Internal strut and "U" tie rods

Nerva XE* 1966	Pump discharge line[b]	LH_2	-420 to +65	0 to 1500	4.62	347 CRES	Inconel 718	Inconel 718, Electroless nickel coat on pin; Electrofilm 77S coat on bore	Internal ring gimbal
	Turbine inlet line[b]	GH_2	65 to +800	0 to 600	3.84	Hastelloy C	Hastelloy C	Hastelloy C, electroless nickel plate on pin; Electrofilm 77S coat on bore	Internal ring gimbal
	Turbine exhaust line[b]	GH_2	65 to +440	0 to 100	6.36	321 CRES	321 CRES	321 CRES	Wire braid
	Vent line[b]	GH_2	-420 to +65	0 to 350	4.62	304 CRES	321 CRES	321 CRES	Wire braid
Nerva XE* 1968	Braided hoses[a]	GH_2	65 to +800	0 to 600	3.84	---	Inconel 718	321 CRES	Wire braid
		GH_2	65 to +800	0 to 600	3.84	---	Hastelloy C	321 CRES	Wire braid
Titan III 1969	Pump inlet line	A-50	65 to 200	0 to 125	5.88	321/347 CRES	321/347 CRES	---	No restraint
		N_2O_4	65 to 200	0 to 125	6.88	321/347 CRES	321/347 CRES	---	No restraint
	Pump discharge line	A-50	65 to 200	0 to 1575	3.70	321/347 CRES	321/347 CRES	350 CRES	Internal tripod with ball and socket
		N_2O_4	65 to 200	0 to 1465	4.75	321/347 CRES	321/347 CRES	350 CRES	Internal tripod with ball and socket
	Pump inlet line	A-50	65 to 200	0 to 125	3.75	321/347 CRES	321/347 CRES	---	No restraint
		N_2O_4	65 to 200	0 to 125	5.68	321/347 CRES	321/347 CRES	---	No restraint
	Pump discharge line	A-50	65 to 200	0 to 1700	2.70	321/347 CRES	321/347 CRES	350 CRES	Internal tripod with ball and socket
	Tank pressure line	Hot gas[c]	65 to 1750	0 to 625	0.75	321/347 CRES	321/347 CRES	321/347 CRES	External braid
		Hot gas[c]	65 to 1750	0 to 625	1.00	321/347 CRES	321/347 CRES	321/347 CRES	External braid
	Roll control line	Hot gas[c]	65 to 1200	0 to 25	7.00	321/347 CRES	321/347 CRES	N-155	Internal link

(continued)

Table I. – Chief Design Features of Line Assemblies in Typical Uses on Operational and Advanced-Development Engines and Vehicles (concluded)

Program, Engine or Airframe Model, Date	Part description or function	Fluid	Temperature range, °F	Operating pressure, psig	Line ID, in.	Materials: Line	Materials: Bellows	Materials: Restraint	Type of bellows restraint
Space Shuttle Main Engine (SSME) 1972	Discharge duct, low-pressure fuel pump	LH_2	-423 to 65	280	5.20	Armco 21-6-9 and Inconel 718	Inconel 718	Inconel 718	Internal tripod with ball-and-socket bearing
	Supply duct to turbine for low-pressure fuel pump	Hot gas[d]	65 to 273	5260	2.00	Hastelloy C and Inconel 718	Inconel 718 with 316L inner ply	Titanium and Inconel 718	External gimbal ring
	Supply duct to turbine for low-pressure oxidizer pump	Hot gas[d]	-279 to 65	5308	2.30	Inconel 718	Inconel 718	Titanium and Inconel 718	External gimbal ring
	Oxidizer tank pressurization duct	GOX	65 to 940	5500	0.75	Inconel 718	Inconel 718	Inconel 718	External gimbal ring
	High-pressure oxidizer pump discharge	LOX	-272	4632	4.00	Inconel 718	None (hard line)	NA	NA
	High-pressure fuel-pump discharge	LH_2	-367	6174	3.40	Ti-5Al-2.5Sn ELI	None (hard line)	NA	NA
Shuttle Orbiter 1972	Fuel line, external tank disconnect to manifold	LH_2	-423 to +200	55	16.85	Inconel 718	Inconel 718	Inconel 718	Internal tripod with ball-and-socket bearing
	Oxidizer line, manifold to engines #1, 2, and 3 (hard section)	LOX	-297 to +200	220	16.85 inlet 21.00 max 13.60 outlets (3 outlets)	Inconel 718	None (hard line)	NA	NA
	Oxidizer line, manifold to engines #1, 2, and 3 (soft section)	LOX	-297 to +200	270	12.07	Inconel 718	Inconel 718	Inconel 718 Titanium	Internal tripod with ball-and-socket bearing External gimbal ring
	Oxidizer fill and drain duct	LOX	-297 to +200	0 to 260	8.00	Inconel 718	Inconel 718	Inconel 718	Internal cross with bearing end ("Gimbar")
	SSME pre-conditioning system return duct	LH_2	-423 to +200	105	2.00 to 4.00	Inconel 718	Inconel 718	Inconel 718 and titanium	External gimbal ring
	External tank pressurization duct	GOX	+600	650	2.00	Armco 21-6-9	Inconel 718	Inconel 718	External gimbal ring

[*] Advanced development, not operational

[a] Bellows qualification testing only – no line experience

[b] Flight qualified (shown for sake of completeness only)

[c] Combustion products of N_2O_4/A-50

[d] Combustion products of LOX/LH_2

NA = not applicable

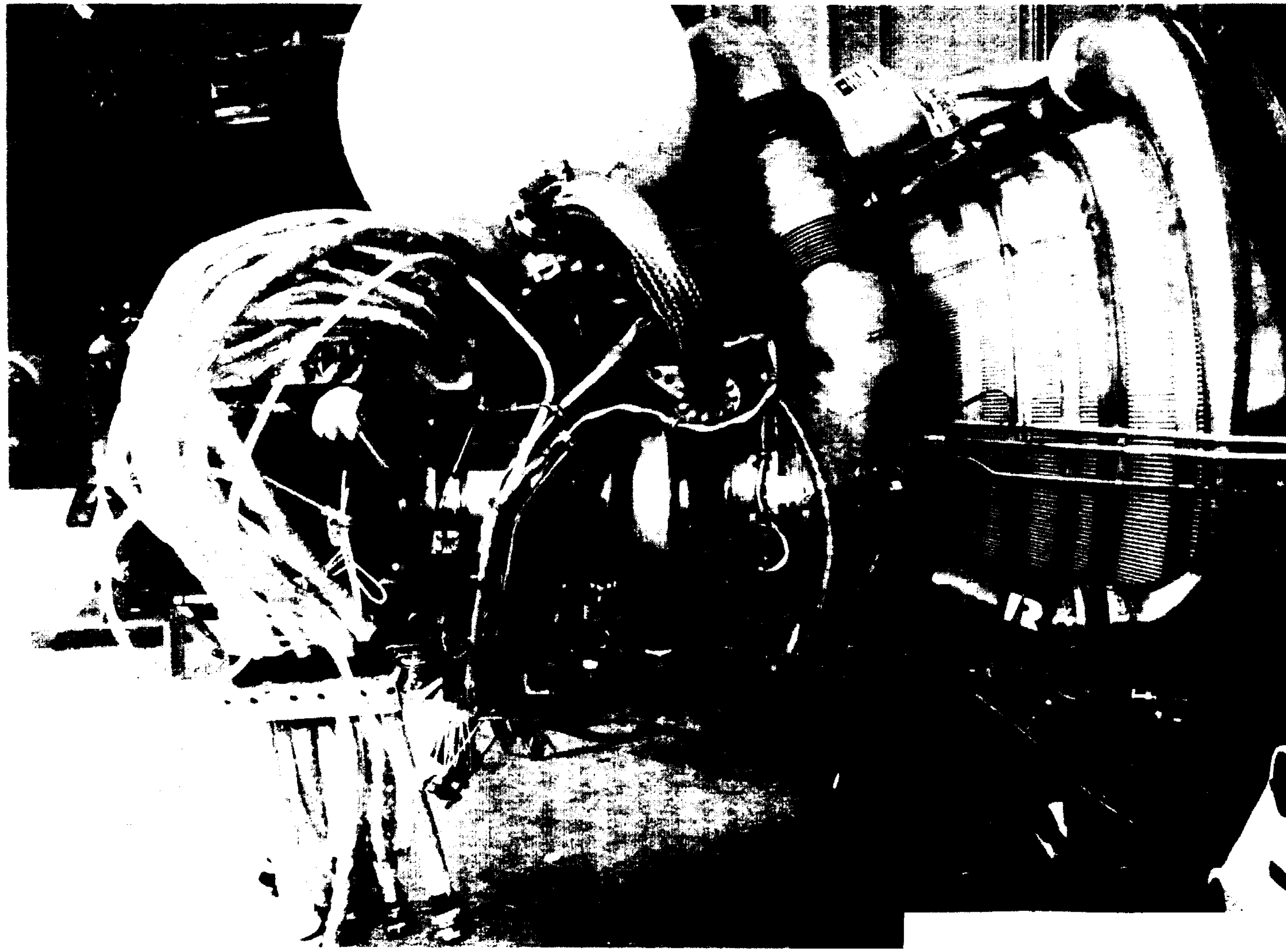


Figure 1. — Typical line, bellows, and hose assemblies on a large pump-fed rocket engine (J-2).

the LOX suction ducts (non-gimballing) and fuel drain duct on the S-1C and the pump inlet ducts on the J-2 engine.

With the advent of the Space Shuttle Program (ca. 1969), the specifications for hardware design included one new feature never before required for space vehicles: reusability. The Space Shuttle must be capable of performing 100 missions. This requirement introduced the need for longer low- and high-cycle fatigue lives, improved long-term corrosion resistance, a greater number of gimballing cycles for articulating ducts, and ease of maintenance and refurbishment. For example, some of the flexible vacuum-jacketed lines in the Shuttle Orbiter main propulsion system are made entirely of Inconel 718 for long-term resistance to damage and corrosion; in comparison, in the Saturn vehicle lines only the basic bellows were made of this material. In addition, the high chamber pressure of the Space Shuttle main engine (3000 psi, three times greater than any previous operational rocket engine), with pump discharge pressures in excess of 7000 psi and pressures in the articulating flex ducts greater than 5000 psi, made necessary a giant step in the state of the art for the design and fabrication of components for service with high-pressure fluid.

Both the long calendar-life requirement (10 years) of the Space Shuttle and the earlier established long-life requirements for unmanned missions to the outer planets make consideration of the long-term aspects of materials selection in component design a must. For example, caution must be exercised in using plastic materials beyond a two-year life. Real-time tests are required to prove life compatibility of materials; accelerated tests are not representative.

Filters. – Recognition of the need for filter protection of critical control components occurred with testing of the Redstone engines. Periodic disassembly or disconnecting of control lines and replacement of components permitted contamination to enter the engine system. Failures of components, generally because of excessive leakage, were traced to the contamination that had been introduced during the periods of disassembly. In subsequent engine programs (Jupiter, Atlas, Thor, Saturn, Space Shuttle), filters were recognized as being vital components in all systems containing contamination-sensitive control components.

Commercial filters were used initially, but as engine requirements became more stringent insofar as envelope and line connections were concerned, specifications to control these features were generated. Filter specifications subsequently were expanded to control such items as allowable pressure drop, material compatiblity, construction features, filter material, and cleanliness requirements.

The majority of filters are made of metallic wire cloth or stacked etched metallic disks; various metals are used, but stainless steel is most common. Most filters have been wire-mesh types because, in comparison with stacked etched disks, they offer maximum surface area for minimum weight and can handle high flowrates. Metallic-disk types have been restricted largely to relatively small engines with low flow requirements. Chief design features of wire-mesh filters used in various rocket engine systems are summarized in table II. The propellants and pressurizing gases supplied to the engine are controlled with respect to moisture and hydrocarbon content, and additional devices such as absorptive separators are not necessary.

2.1 LINE ASSEMBLY

As noted, the line (or duct) assembly may be either a combination of hard lines and flexible elements (bellows joints or flexible hoses) or a completely hard line that achieves flexibility through the use of bends and material elasticity. Basic design considerations in establishing a line configuration are routing, sizing, pressure-drop and vibration control, and location and type of flexible joints; other considerations include materials, cost, weight, reliability, and maintainability. References 1 through 5 present details on line-system design.

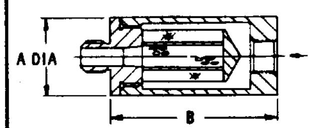
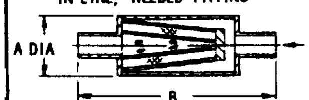
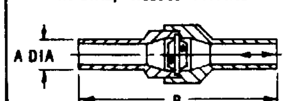
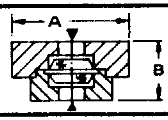
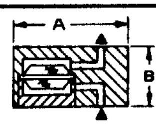
Flexible lines. – A typical flexible-line assembly consists of tubular straight runs and elbows, with flexible bellows joints located along the line, separable connectors with static seals at either end for installation into mating components, and branch-off line connections. Flexible-line assemblies have been utilized successfully in sizes from ¼-in. diameter to 27-in. diameter, at temperatures from -455°F (liquid helium) for pressurization systems and -423°F (liquid hydrogen) for propellants to + 1300°F (turbine-exhaust gases), with pressures ranging from vacuum to 3000* psi, and with every cryogenic- and storable-propellant combination successfully used in rocket engines. There are two basic flexible-line configurations: the tension type (fig. 2), and the compression type (fig. 3).

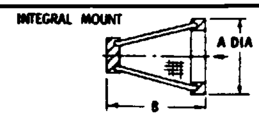
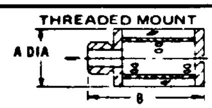
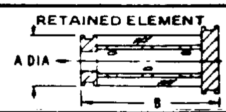
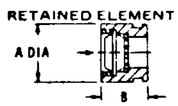
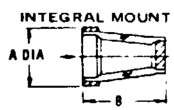
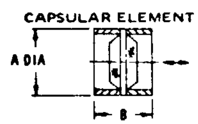
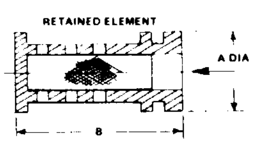
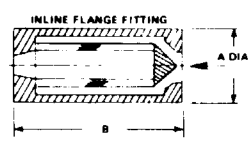
The tension-type system utilizes bellows restrained by linkages that withstand the pressure separating load (i.e., the axial load due to the force generated by internal pressure that tends to separate the ends of the bellows); the linkages may be either external or internal and permit angular motion. The line routing requires two or more bends with bellows joints located between bends. Overall length changes in the line assembly are accommodated through angulation of the flexible joints.

Figure 2 presents two examples of tension-type configurations: the three tension-tie bellows joints in a dog-leg arrangement, and the axial bellows with two tension-tie bellows joints. In the three-tension-tie configuration (fig. 2(a)), one end of the duct is capable of three-dimensional movement and angulation in any plane with respect to the other end. The duct is shortened by angulation of the individual flexible joints. In the axial bellows (fig. 2(b)), angulation and shear motion of one end with respect to the other are accommodated by two tension-tie bellows joints. Necessary changes of length are achieved with a thrust-compensating bellows joint. This configuration usually is employed when large axial deflections are necessary (e.g., in the inlet ducts of a gimballed pump).

*Up to 5500 psi in the advanced-development SSME.

Table II. – Chief Design Features of Wire-Mesh Filters in Typical Uses on Operational and Advanced-Development Engines and Vehicles

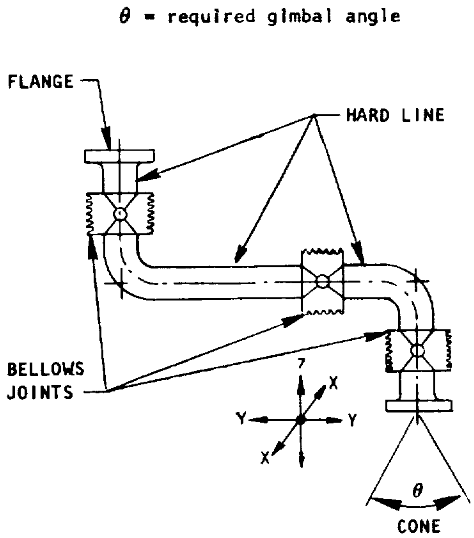
Filter type	Vehicle or Engine	Application	Filter element description						Performance requirements			
			Type weave	Configuration	Min. flow area, in.^{2}	Maximum assembly weight, lbm	Absolute filtration rating,[a] μ	Size A x B, in.	Operating fluid	Temp., °F	Operating pressure, psig	Maximum pressure drop at noted flow
IN-LINE, TUBE FITTING 	Atlas	Pressure regulator inlet	Twilled double Dutch	Supported pleated cylinder	30	1.5	25	1.6 x 4.9 (1/2 in. line)	He	-65 to +160	0 to 3000	50 psi at 1200 psi and 0.5 lbm/sec
		Pressure regulator inlet	Twilled double Dutch	Supported pleated cylinder	—	0.55	25	1.0 x 4.4 (1/2 in. line)	He	-65 to +160	0 to 3000	18 psi at 800 psig and 0.1 lbm/sec
IN-LINE, WELDED FITTING 	J-2S[b]	Control system lines	Twilled double Dutch	Supported pleated cylinder	—	0.2	15	0.87 x 5.0 (3/8 in. line)	He	-200 to +140	0 to 640	20 psi at 400 psig and 0.004 lbm/sec
IN-LINE, WELDED FITTING 	J-2S[b]	Purge system lines	Twilled double Dutch	Pleated disk	1.9	0.5	25	0.38 x 3.0	He	-200 to +140	0 to 650	5 psi at 650 psig and 0.052 lbm/sec
		Accumulator inlet and discharge line	Twilled double Dutch	Pleated disk	—	0.5	18	0.5 x 3.5	He	-200 to +140	0 to 500	10 psi at 500 psig and 0.024 lbm/sec
 INTERFACE	Saturn V J-2	Accumulator inlet and discharge line	Plain Dutch single	Pleated disk	1.6	—	25	1.6 x 0.61	He	-250 to +140	0 to 425	10 psi at 400 psig and 0.03 lbm/sec
INTERFACE 	Saturn V J-2	Helium tank vent valve	Twilled double Dutch	Pleated disk	1.5	0.4	15	2.4 x 0.5	He	-350 to +140	0 to 600	5 psi at 600 psig and 0.002 lbm/sec

	Saturn V J-2	Heat exchanger inlet	8 x 8 mesh (.028 in. wire)	Conical screen	3.15	0.2	6600 (0.26 in.)	1.8 x 1.5	LOX	-290	0 to 1100	3 psi at 1100 psig and 2.0 lbm/sec
	J-2S[a]	Pressure regulator inlet	Twilled double Dutch	Supported pleated cylinder	21.5	0.5	18	1.2 x 2.3 (3/8 in. line)	He	-200 to +140	0 to 4000	15 psi at 600 psig and 0.1 lbm/sec
	Saturn V F-1	Hydraulic control package	Plain Dutch single	Supported pleated cylinder	116	—	40	2.3 x 7.0	RP-1	-65 to +165	0 to 2500	12 psi at 85 gpm
	Lance	Propellant valve inlet	Plain Dutch single	Supported pleated disk	—	0.15	25	1.3 x 0.5	UDMH	-65 to +160	0 to 1250	25 psi at 6.0 gpm H_2O
	LEM ascent engine	Propellant manifold	Square	Pleated cylinder	-	0.03	120	1.0 x 1.4	UDMH, N_2O_4	+40 to +120	0 to 155	8 psi at 155 psig N_2O_4 and 7.0 lbm/sec
	Saturn V J-2, F-1	Control system components	Twilled double Dutch	Pleated disk	0.38	0.01	25	0.5 x 0.25	He RP-1	-320 to +140	0 to 400	10 psi at 400 psig He and 0.03 lbm/sec or 10 psi at 0.32 gpm "Freon-TF"
	SSME[b]	Pneumatic control system	Twilled double Dutch	Supported cylinder	6.25	0.38	15	1.7 x 2.8	He (vehicle) N_2 (GSE)	-30 to +130	750	25 psi at 625 psig N_2 at 70°F and 0.424 lbm/sec
	SSME[b]	Hydraulic actuator system	Twilled double Dutch	Supported pleated cylinder	180	3.0	25	2.0 x 7.5	Hydraulic fluid[c]	+10 to +250	4000	55 psi at 4000 psig at 10°F and 1.67 lbm/sec

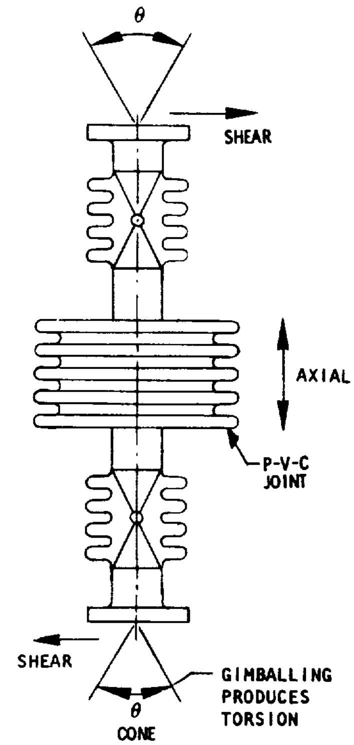
[a]Glass bead

[b]Advanced development, not operational

[c]Meets specifications in MIL-H-83282, Rev. A, Feb. 22, 1974.

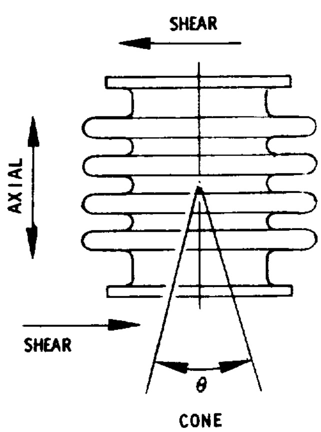


(a) Three tension-tie bellows joints in a dog-leg line arrangement

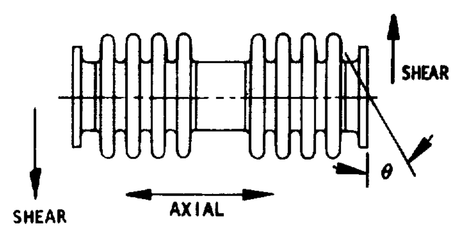


(b) Axial bellows with two tension-tie bellows joints

Figure 2. — Tension-type flexible-line configurations.



(a) Single bellows



(b) Double bellows

Figure 3. — Compression-type flexible-line configurations.

The compression-type duct system (fig. 3) utilizes free bellows (i.e., no tension-tie linkage across the bellows) to absorb the deflections imposed on the duct during operation. The pressure separating loads on the bellows are reacted by supporting engine or vehicle structure. In addition, support bracketry may be required to offset column buckling if the ducting has a large length-to-diameter ratio (> 1). If the operating pressures are high and lines are large, the pressure separating forces are correspondingly large, and restraint structure becomes necessarily heavy. Consequently, the compression-type system, although widely used, generally is limited to low-pressure (< 400 psi) applications such as the inlet ducts of the engine propellant pumps.

Examples of compression-type configurations shown in figure 3 are the single bellows and the double-bellows spoolpiece. A single free bellows (fig. 3(a)) can be utilized to absorb all motions (angular, shear, or axial) when short-coupled components are to be connected and when the line pressures and motion requirements are relatively low. In this configuration, the pressure separating load on the bellows is resisted by the end-mounting structure. The bellows of necessity will be long enough to accommodate all motions simultaneously. This length could lead to buckling instability, but bellows can be stabilized by external linkages if increased weight and size are not critical considerations. A double-bellows spoolpiece configuration (fig. 3(b)), which is similar to the single bellows, can be used in applications where greater shear (offset) motion is required and sufficient space to accommodate the increase in length is available. The turbine exhaust ducts of the engines on Thor, Atlas, and Saturn vehicles incorporate compression-type bellows. They are acceptable for use in these systems because the ducts are straight and relatively short and the operating pressure is low.

Hard lines. – Hard lines (lines with no bellows or flexible hose to act as flexible members) may be used in non-gimballing applications where line flexibility is required only to accommodate installation misalignment and thermal expansion or contraction. Because hard lines are quite rigid in comparison with lines with bellows joints, they must be custom-made to fit the particular installation, or specially sized spacers must be used at the separable connectors to make up installation tolerances. The feature of installation interchangeability that is inherent in ducts with bellows joints thus is sacrificed with hard lines. Material selection, wall thickness, and routing are the variables available to the designer for achieving flexibility necessary to accommodate thermal deflections and deflections resulting from engine operation or in-flight forces. Figure 4 depicts some common hardline configurations used to achieve flexibility.

The successful use of rigid ducts for nongimballing applications in rocket engines has been well established. The pump discharge ducts of the F-1 engine were large-diameter (3-in. for fuel and 6-in. for oxidizer), high-pressure (1600 to 1850 psi) rigid lines made of 6061-T6 aluminum. The lines had a long history of operation free of failures (45 engines launched, 3179 tests, and 272 500 seconds accumulated engine time).

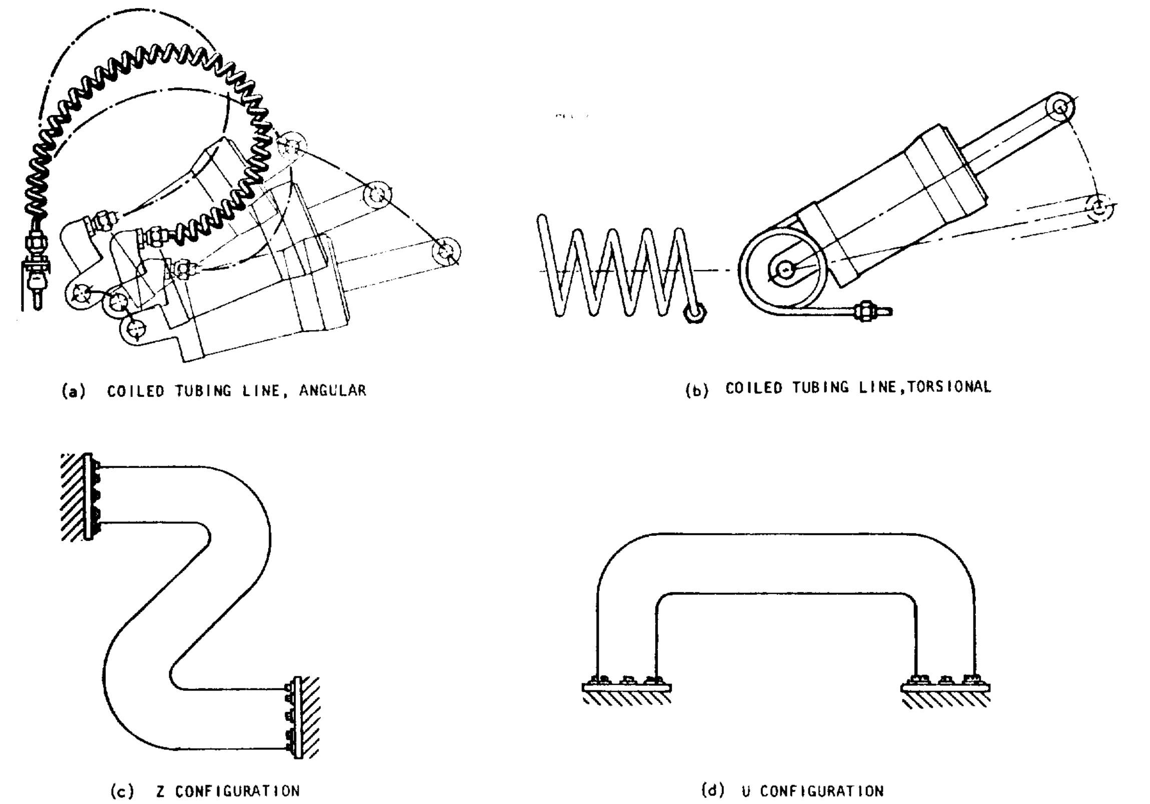


Figure 4. – Common configurations used to provide flexibility in hard lines.

The pump discharge ducts of the J-2S engine also were rigid. These ducts were 3 in. in diameter and were made of Armco 21-6-9 alloy. The operating pressure was 1400 to 1600 psi. No failures or misalignment problems occurred in 603 tests, in which 28 444 seconds of firing time on 18 engines were accumulated.

Hard lines are used less frequently than are flexible lines in current aerospace vehicle and engine systems. However, this relation could be reversed in future applications if operating pressures increase to the point where the use of bellows joints becomes impractical.

2.1.1 Routing

2.1.1.1 CENTERLINE GEOMETRY

The locations of the propulsion-system components largely determine line routing. In the preliminary design layout of a typical liquid-propellant rocket engine, the primary components such as pumps, thrust chamber, and gas generator are located in their optimum system positions, and the interconnecting ducts are then routed, with a minimum of flexible joints, within the intervening space; this procedure, however, does not always make for the best ducting arrangement. The ducting often is made up of many turns and elbows, expansion and contraction sections, and attachment points. The centerline geometry of engine ducting thus represents a compromise in routing between space available and the best ducting design practice.

Vehicle and spacecraft ducting also must compromise with practicality. Boost vehicles, for example, to a great extent are cylindrical propellant tanks, and the propellant feed ducting must be limited to long, straight lengths leading from the tank outlets to the engine inlets. Spacecraft lines typically are small in diameter, have no flexible joints, and are circuitously routed to fit the limited space available in a structure where size and weight are at a premium.

2.1.1.1.1 Flexible Lines

A flexible-element duct system may be a tension, compression, or combination tension-compression type. The type of system selected depends on a number of factors (e.g., the space available, the availability of mounting structure for anchoring, the requirement for thrust-vector gimballing).

For a line containing flexible joints, centerline geometry is established to minimize the number of flexible joints. Since a flexible joint (bellows) is complex, costly, and not as reliable as a hard line, the necessity for each flexible joint is carefully scrutinized in the design phase. System considerations that must be kept in mind during the centerline routing and locating of flexible joints include provisions for (1) line deflection and clearance under vibration, thermal contraction and expansion, and engine gimballing, (2) accessibility for installation and removal, (3) wrench clearance around bosses and fittings, (4) accessibility for in-place welding or brazing, (5) consideration for clearance if foam insulation is required on the line, and (6) accessibility to other major components.

A classic configuration adhering to the minimum-number-of-joints principle, believed to have originated with pump discharge ducts of the Atlas booster engines, is the wraparound duct arrangement for gimballing engines; this configuration, shown in figure 5, locates a flex joint centered on each of the two gimbal axes. This geometry was or is used for the Navaho, Atlas, Thor, and Jupiter engine pump discharge ducts, the pump inlet ducts of the H-1 engine, and the gimballing feedlines of the Apollo Service Module engine and the descent engine for the Lunar Excursion Module (LEM). This configuration was used for all of the vehicle-to-engine interface lines of the F-1 engine on the S-IC except the main propellant lines.

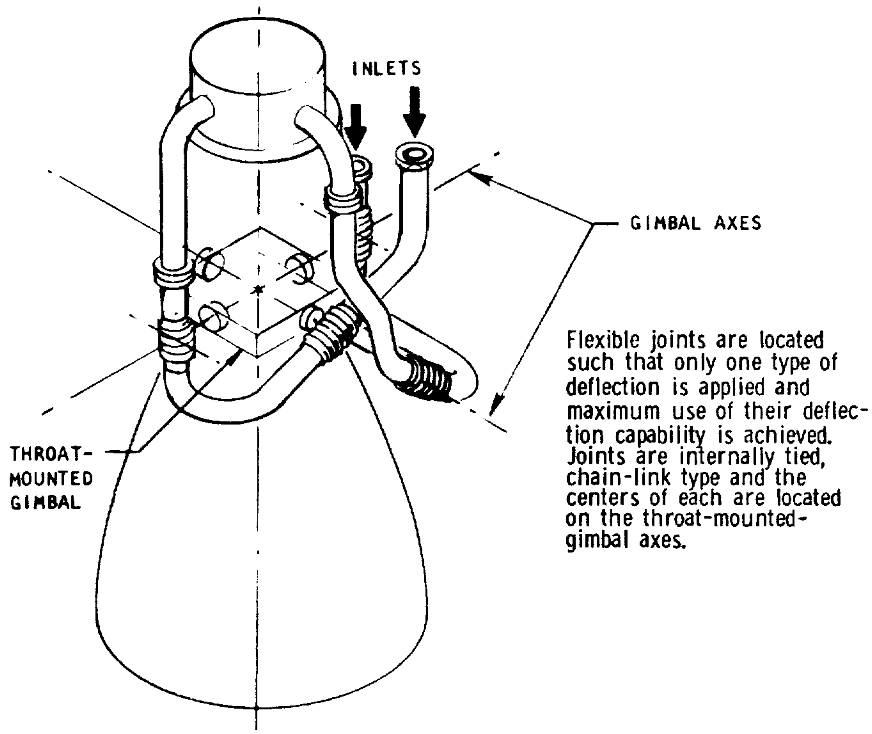


Figure 5. – Propellant feedline arrangement on the LEM descent engine.

The wraparound concept is used extensively on the SSME, with seven articulating ducts and six braided metal hose assemblies crossing the engine gimbal plane. Each of the ducts has three tension-type bellows joints with one of the joints centered as closely as possible on each of the gimbal axes. The ducts are arranged in parallel routing around the gimbal in a plane perpendicular to the engine thrust line (fig. 6).

The second important consideration in duct routing is the location of flexible joints. Once the number of flexible joints is established, a kinematic analysis is performed to determine the optimum location for each in the system. Each joint is positioned in the duct assembly to maximize the deflection capability of the assembly. Further, each joint is located and designed to accommodate as few modes of deflection as possible (e.g., axial only, with no shear or angulation) and to minimize any deflection of the joint so that minimum strain is imposed.

The design of the SSME involved many articulating ducts that wrapped around the gimbal plane, and early in the layout phase it became necessary to develop a space-frame program

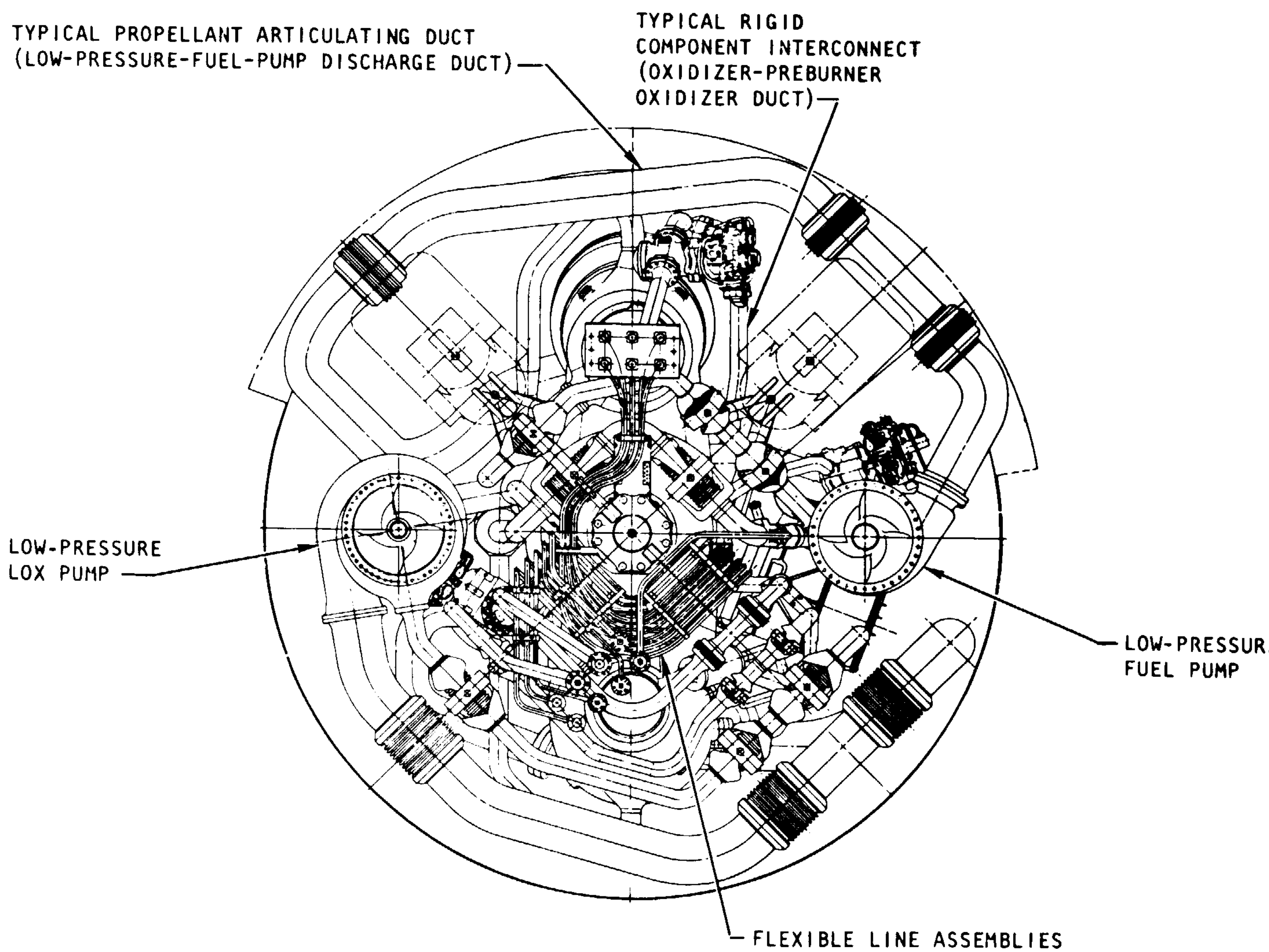


Figure 6. – Plan view of articulating duct arrangement in gimbal plane of Space Shuttle main engine.

for locating the flex joints in a duct assembly at the optimum locations, i.e., at the points of minimal angular deflection. This space-frame program models the duct or line as a series of beam finite elements and accounts for duct flexibility and simulated joint properties. The appropriate displacements of one end of the duct with respect to the other are applied, and the outputs are joint loads and deflections. Location of joints can be subjected to perturbation analysis to minimize joint motions. This program became invaluable for locating the joints to ensure the smallest engine-duct envelope possible.

2.1.1.1.2 Hard Lines

Since flexible bellows joints are more expensive and less reliable than rigid ducting, such joints are utilized only if a hard line will not satisfy the design requirements. Hard lines are preferred, if at all possible, in any ducting application, because they offer simplicity in design and fabrication, low cost, and high reliability; their drawbacks, which must be evaluated and weighed in design-phase trade studies, are loss of interchangeability and high attachment-point reactions to any applied loads. The centerline geometry of hard lines thus is established to minimize the loads on attachment points.

Recent experience indicates that some engine designers have changed their attitude toward line design. Ducts originally were designed to include flexible joints, but later engine models were designed with "hard" ducts. Two such cases are the RL10 engine on the Centaur upper stage and the F-1 engine on the Saturn V booster (S-IC).

In the early RL10 engine, braided flexible sections were used throughout the ducting; the present version is completely devoid of flexible joints. Loops and bends in the hard ducts provide flexiblity.

The early F-1 engines incorporated bellows joints in the pump discharge ducts, but these were later replaced by aluminum hard lines with generous bends for flexibility (fig. 7); the low modulus of elasticity of the aluminum permitted lower end reactions for a given deflection than did a comparable duct of steel or nickel-base alloy. The change was possible because the ducts were nongimballing and were required to absorb only misalignments and thermal effects. Manufacturing lead time and costs for the original bellows-joint ducts dictated the change even though some interchangeability was lost.

2.1.1.2 DEFLECTION LIMITATION

Different deflection modes (i.e., axial, angular, and shear) produce the same type of bending stresses in the bellows convolutions. If a bellows is required to deflect in all of these modes simultaneously, the sum of the deflections for all modes must be kept within acceptable stress limits. (Reference 1 presents methods for converting different types of bellows

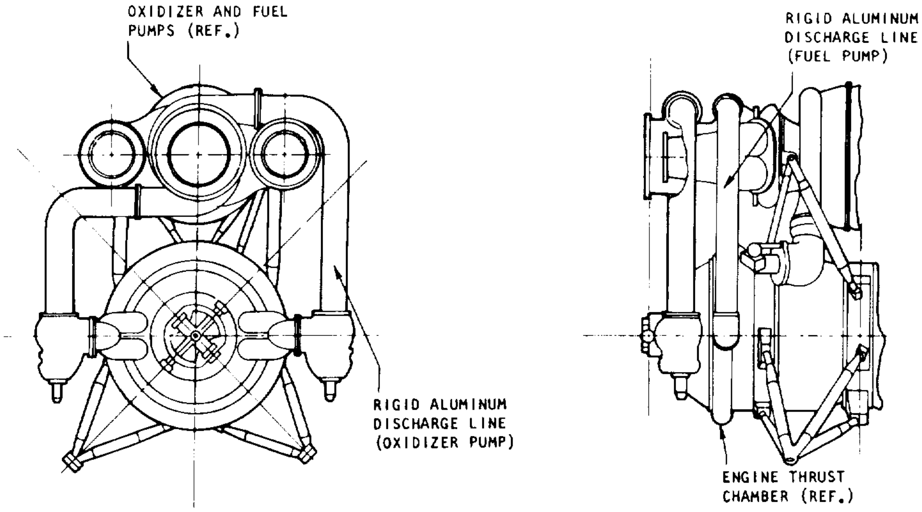


Figure 7. – Arrangement of hard lines on pump discharge of F-1 engine.

deflections into equivalent axial deflection.) The duct system therefore is routed so that all accumulated deflection stresses are kept within the safe working stress value of the material; thus reasonable fatigue life is achieved.

2.1.1.3 TORSIONAL DEFLECTION

Torsional loading (twist about the duct longitudinal centerline) is avoided in the design and application of a duct system. Because of the inherent torsional rigidity of a typical duct, torsional deflections can be imposed only by high external loads, which must be resisted by the attaching structure. Structural failure or early fatigue failure can be the result. When, because of design constraints, torsional loading is unavoidable. the torsional effects are minimized by appropriate design.

The duct system is routed so that torsional deflection imposed on any one joint is minimized. If space limitations force a duct to be designed relatively short and straight, so that torsional deflections cannot be absorbed through angulation of bellows joints in dog legs of the duct, other means of permitting torsional deflection must be provided. Shown in figure 8 is a device designed to absorb torsional deflection; this device, a tightly formed bellows assembly, was incorporated into the pump inlet ducts of the J-2 engine (used on the Saturn S-II and S-IV stages) when the torsional moment of the ducts was found to be too high for the pump casings to resist. The bellows is thin-walled (0.010-in.) and over 100 in. long. It has 40 deep convolutions stacked in a 1.25-in. height. Flanges encompass the

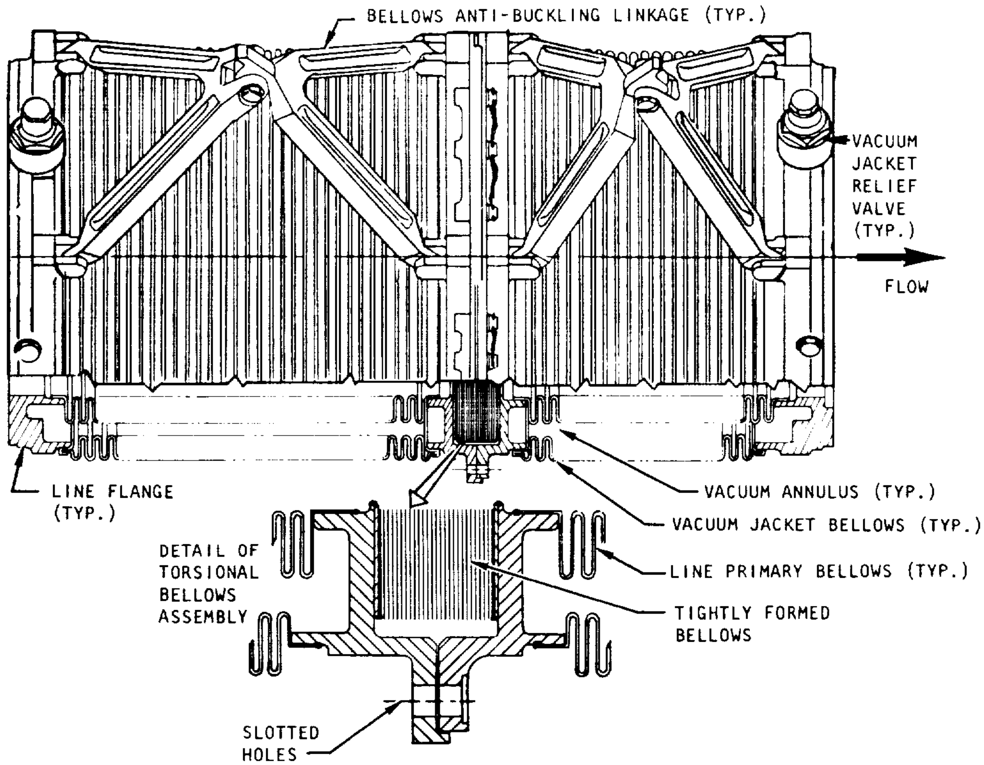


Figure 8. – Tightly formed bellows subassembly used to absorb torsional deflection of primary bellows in inlet line of pump on J-2 engine.

bellows and permit the application of only torsional deflections. The joint is, in effect, a low-spring-rate torque tube that can absorb torsional rotation up to 3°.

2.1.2 Sizing

2.1.2.1 FLOW AREA

After the centerline routing of the duct has been established, the inside diameter (ID) must be determined. The ID is a compromise among tolerable system pressure drop, available space, weight, spring rate and pressure-thrust reaction of bellows, system dynamic considerations, and cost.

Careful analysis of the pressure losses in the individual components gives a high degree of confidence to the engineering judgment involved in weighing that parameter in the design compromise. Typical rocket-engine ducting (and most vehicle ducting) is composed of a series of close-coupled, high-energy-loss components such as bends, branches, valves, expansions, contractions, and flex joints. Any prediction of system energy loss based on the test data for individual elements is likely to be very conservative. The reason for this condition is that flow tests made to determine the pressure loss of a particular component usually are run to evaluate the maximum component loss. The maximum loss is realized when the installation incorporates a long straight run downstream of the test component; only part of the loss chargeable to the component actually occurs within its confines, the remainder arising from flow disturbances downstream of it. Obviously, if another component closely follows the first, part of the effect of the first is cancelled. Thus a 180° return bend causes no more than four-thirds the loss of a 90° elbow, or slightly over twice the loss of a 45° elbow. For calibration of a particular component, provision is made to subtract test-setup tare* from total differential pressure. References 6 and 7 provide detailed procedures. References 8 through 12 treat pressure losses in duct components.

2.1.2.2 WALL THICKNESS

After the duct flow area has been sized, the duct wall thicknesses are determined. Wall thicknesses are established through a stress analysis, in which the magnitudes of the stresses produced by fluid pressure, fluid flow, thermal gradients, external forces, and acceleration forces are evaluated so that the optimum thickness for reliability, fabricability, and weight minimization can be selected (ref. 12). Methods of calculating stress levels are those commonly available in strength-of-materials textbooks (e.g., refs. 13 through 15). In the structural analysis, consideration must be given to the following design elements and structural influences: branches, brackets, doublers, bosses, stiffeners, column buckling, external collapsing pressure, and mechanical vibration (ref. 16).

Determination of hard-line wall thickness is particularly important because the lack of flexible sections requires that the line itself and its attachment points be capable of handling any applied load. The structural analysis of hard lines is based on the maximum envelope resulting from tolerance stackups. The minimum wall thickness, developed from basic thin-shell theory, is based on maximum operating pressure at the critical operating temperature and includes the effect of thinning in the bends. The flanges are designed to develop the axial yield strength of the line at operating temperature. Maximum misalignment, thermal, and discontinuity loads are determined by use of a space-frame analysis. The maximum effective line stress typically is limited to provide a minimum safety factor of 1.4 on combined loading, and 1.5 on ultimate and 1.1 on yield for pressure loading.

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*Basic pressure drop of the test setup only, with the component under test removed.

Life considerations for hard lines include both high- and low-cycle fatigue. For high-cycle fatigue, the effective alternating stress due to pressure and vibration usually is limited to provide a minimum safety factor of 1.4 on the material endurance limit. For low-cycle fatigue, the effective peak strain resulting from both primary stresses (pressure and vibration) and secondary stresses (misalignment, thermal, and discontinuity) typically are limited to provide a minimum factor of 4 on cycles. The strain levels of hard lines can be up to twice the yield strain. Yielding during the initial installation will permit operation in the elastic region for subsequent loadings.

In general, on the basis of fabrication, handling, and maintainability considerations, a minimum wall thickness of 0.032 in. is used for hard lines 1 in. or more in diameter; lines less than 1 in. in diameter have wall thicknesses based on pressure requirements and are supported to keep mechanically induced vibration stresses below the material endurance limit.

All lines are proof-pressure tested. Lines operating at cryogenic temperature may be proof-pressure tested using liquid nitrogen. Testing of the final design often is performed at a component level to verify structural integrity under simultaneous application of as many different applied loads as is economically feasible.

2.1.3 Control of Pressure Drop

2.1.3.1 FLOW-DIRECTION CHANGE

When engine-vehicle interfaces do not permit a straight-in approach, flow must be directed into the engine (or pump) through elbows. For flow in a given Reynolds-number regime, the loss coefficient for an elbow attains a minimum value and then increases as the ratio of bend radius to inside diameter (R/D) increases. If line routing requires small-radius elbows, pressure drop can be minimized by selecting an optimum R/D value for the elbow; otherwise, pressure drop can be minimized by addition of flow guide vanes (fig. 9).

In an elbow having a small R/D value, guide vanes provide parallel-flowing elbows of more nearly optimum R/D. Flow guide vanes as shown in figure 10 are utilized in sharp elbows at the pump inlets for the Centaur (RL10), Thor, Atlas, and Saturn S-IC (F-1) engines; they have also been used in the pump discharge ducts of the Thor and Jupiter engines. Minimization of pressure drop in all these applications directly enhances engine performance. (References 17 and 18 are sources of design data on pressure loss for flow guide vanes.) Vanes can be excited by system fluid oscillations; therefore, vane frequencies are calculated, so that potential vibration may be avoided.

Branch or takeoff duct centerlines are aligned with mainline flow directions to capture mainline velocity pressure. This alignment provides higher available pressure at the branch

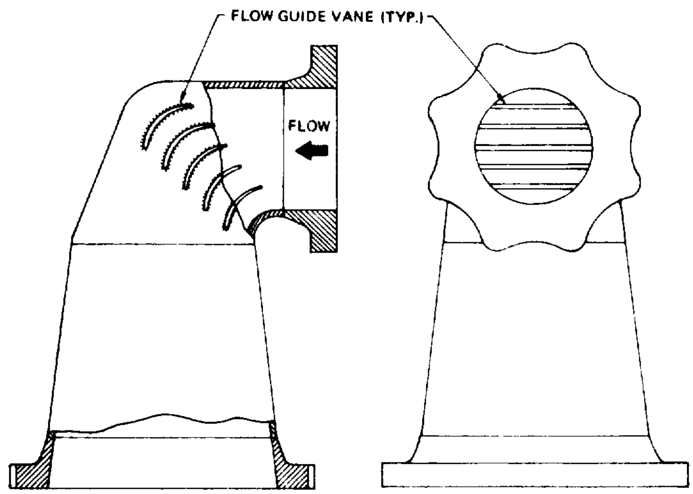


Figure 9. – Typical vaned elbow configuration.

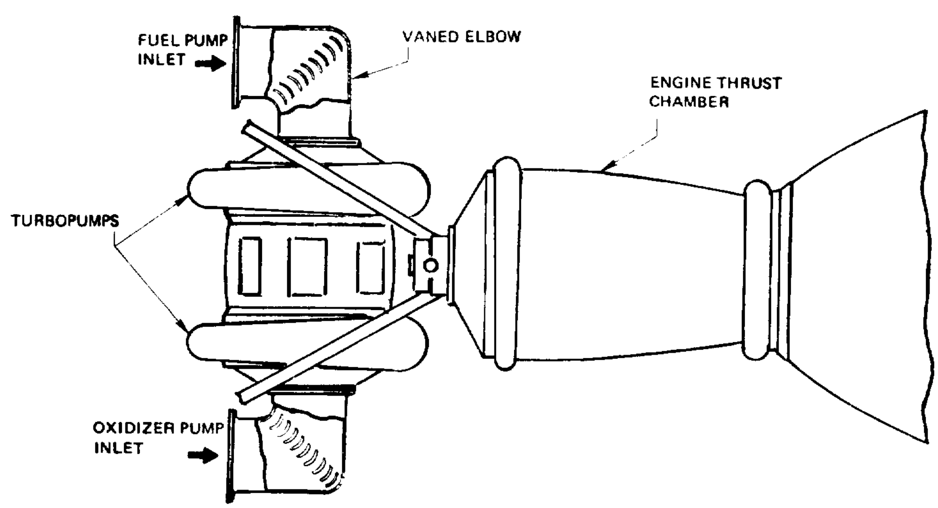


Figure 10. – Flow guide vanes in sharp elbows of pump inlet lines.

inlet than would be available at a flush wall tap at a right angle to the mainline flow. Minimum tapoff pressure drop is important in applications such as tapoffs from main propellant ducts to gas-generator bootstrap lines. Reference 19 provides information on optimum takeoff angle.

Bellows flow liners (sec. 2.2.4) frequently are used to minimize the high friction losses of the convolutions. References 20 and 21 provide data on loss factors for liners.

2.1.3.2 FLOW-AREA CHANGE

Almost every interface where a line begins or terminates involves an area change to or from the line. The shape of these transitions requires careful consideration if excessive pressure loss from abrupt flow-area changes is to be avoided. Flow-contraction pressure losses can be reduced by about 90 percent if the edges of the contraction are radiused rather than sharp. Radii larger than 0.15 times the contracted diameter add little improvement. For expanding sections, little reduction in pressure loss for a given area ratio can be expected unless the divergence cone angle is kept small (total included angle near 10°). References 22 and 23 treat these phenomena in detail.

In small-diameter lines with threaded fittings, care must be exercised in calculating the line losses, because standard fittings (unions, tees, elbows, and crosses) usually have a smaller inside diameter than the tubing itself. Reference 24 presents data on loss factors for fittings.

2.1.3.3 FLOW DISTRIBUTION

The flow distribution (to maintain equal pressure drop) at the exit of engine feed ducting is an important factor in duct design. For example, the flow profile at pump, valve, and injector entrances can have a great effect on the performance of these components. In pumps having dual opposed discharge ports, the attached ducts must have equal pressure loss to ensure equal flow out of each port. Flow distribution has been improved with vaned elbows (fig. 10) and with "egg-crate" straighteners (fig. 11) downstream of elbows. This latter design evolved as a solution to the uneven flow distribution of LOX from the inlet elbow to the injector on the Atlas and Thor engines.

The flow splitter in the propellant feedlines of the Lunar Excursion Module (LEM) descent engine (fig. 12) represents another flow-directing device for achieving even distribution. The engine utilizes redundant shutoff valves; in normal operation, both valves open and close simultaneously. However, in a partial failure mode, either of the valves can be closed. Flow into the valves therefore must be evenly distributed for minimum effect on engine performance. Even distribution is achieved with the flow splitter in the duct elbow immediately upstream of the valves.

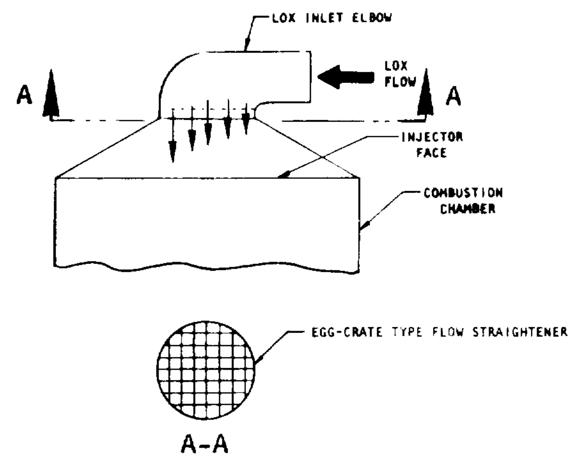


Figure 11. – Flow-distribution device incorporating an "egg-crate" type of flow straightener.

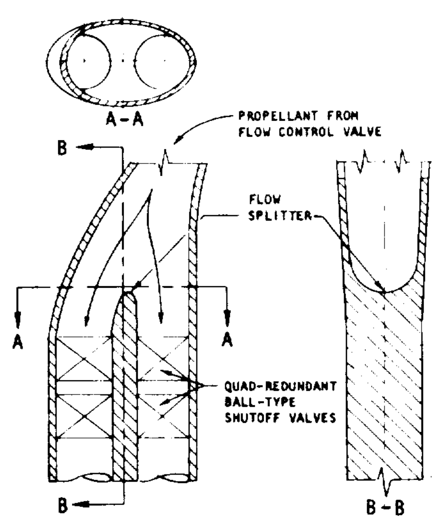


Figure 12. – Flow splitter in propellant feed system of LEM descent engine.

One problem associated with flow straighteners or splitters, or any other structure that rigidly spans the diameter of a duct, is structural loading by differential thermal expansion or contraction. In a duct, the vane usually heats or cools during thermal transients at a faster rate than the restraining duct wall. When this happens, the vanes try to move farther than the duct will allow. If the differentials are large enough, distortion or destruction of the vanes or duct can occur. The problem is solved by providing an expansion element in the duct-to-vane connection that will slide or flex, absorbing thermally-induced motion while transmitting minimum loads to the wall. One effective solution has been a sliding clevis joint, which allows sliding normal to the wall but adequately carries the flow loads on the vane to the wall.

2.1.3.4 FLOW RESISTANCE

Surface friction between duct walls and the flowing fluid can produce large pressure losses if the walls are rough; as the ratio of roughness-height to duct-diameter increases, so does the pressure loss. Castings generally have undesirably rough walls and are smoothed to improve flow efficiency. Drawn tubing and sheet metal ducts have been adequately smooth for all applications, but weld protrusions into the flow stream have to be controlled when they contribute sufficient disturbance to the flow stream. Weld protrusions are critical for small lines, where they contribute more to large contraction ratios than they do in larger lines.

In flexible-joint assemblies with internal-tie linkages (e.g., chain link, internal tripod with ball-and-socket bearing, internal gimbal ring), the linkages protrude into the flow stream and cause head losses. References 25 and 26 present data on pressure loss factors for some of these bellows restraints.

2.1.4 Control of Pump-Inlet-Line Vibration

Virtually every pump-fed liquid-rocket vehicle developed in the United States has experienced some form of vibrational instability during flight. This vibration first caused real concern in the Titan II in 1962. A significant longitudinal instability occurred late in first-stage flight. Analysis showed that the vibration was caused by a regenerative-feedback interaction between the vehicle's propulsion and structural systems. This lengthwise oscillation was named Pogo, after the motion of a pogo stick (ref. 27). The incorporation of simple hydraulic-suppression devices into the first-stage propellant feedlines solved the Titan Pogo problem, and the oscillation amplitude at the payload was maintained within ± 0.25 g, a level tolerable to the astronauts in the Gemini spacecraft. Pogo is a system design problem; however, duct designers need to be aware of the phenomenon because it can influence their designs, particularly those of the pump inlet ducts of a pump-fed system.

The longitudinal oscillation involved in Pogo can be described in terms of effects that occur in a closed-loop system. The oscillation can be initiated by a perturbation or pulsation in the thrust force that causes a response in the vehicle structure. This structural response applies accelerations to the propellant feed system. The fuel and oxidizer suction lines of the feed system respond separately to these accelerations, the result being pressure pulses at the inlet to the fuel and oxidizer pumps. The pumps and discharge lines act on these pressure pulses to transmit a varying rate of propellant flow to the combustion chamber. The combustion process then generates a pulsating chamber pressure and a pulsating thrust. If this feedback thrust tends to reinforce the initial perturbation, instability can occur.

The devices used to attenuate the flow perturbations in the Titan II feedlines (figs. 13 and 14) are similar in operation to the surge tanks employed in the pressure regulation of large pipelines that have water-hammer problems. Surge tanks have been used as pressure-stabilizing devices in the flow lines of hydroelectric plants and pumping stations for more than 50 years; they serve as a point of pressure relief or cushion whenever there is a sudden change in flow. For vehicle feedlines, however, suppression in a specific frequency range is required; and this requirement necessitates a specially tuned surge system. By selection of the proper damping and spring characteristics, the pressure perturbations in the line can be absorbed by the suppression device and, in effect, uncouple the engine-to-structure feedback loop.

In the Titan II, an entrapped gas bubble was incorporated in the oxidizer-line standpipe to provide a cushion or soft spring for the oxidizer mass in the standpipe to act on (fig. 13); the energy due to pressure oscillations in the feedline can be transferred to this spring-mass system by judicious choice of the volume or height of the entrapped bubble. The fuel feedlines incorporated piston-type accumulators that utilize a mechanical helical-spring-and-piston arrangement to provide the desired soft spring action (fig. 14); the fixed mass of the spring and piston along with the mass of fuel in the accumulator provided the equivalent mass required for a resonant system. The suppression devices were constructed and tuned so that their frequency responses, coupled with the appropriate feedline characteristics, would provide maximum attenuation of pump suction-pressure oscillations that were excited by tank-structure oscillations. The combined system could be optimized for maximum attenuation in a specified frequency range.

Pogo oscillations also occurred in the Saturn V vehicle during boost flight. The frequency involved, 5 Hz, happened to be the natural frequency of the combustion process of the F-1 engines and of the entire Saturn V vehicle including the spacecraft. The vibration increased as propellants were consumed, because the natural frequency of the vehicle increased; the frequency approached 5.5 Hz about 125 seconds after liftoff. While not necessarily destructive, the vibration had to be attenuated because it placed an undesireable acceleration on the crew. The design solution was to deter the two frequencies by placing a pneumatic spring in the liquid-oxygen feedline of each of the five F-1 booster engines. Cavities in the LOX prevalves for the engines provided convenient volumes for introducing gaseous helium, which does not condense at liquid-oxygen temperature, to act as de-tuners (ref. 28).

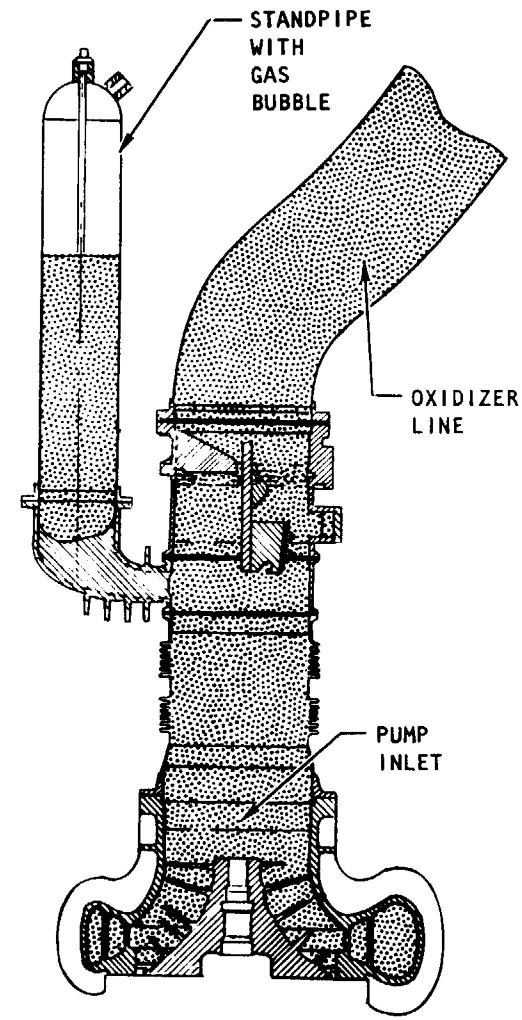


Figure 13. — Standpipe with bubble for Pogo suppression, Titan II oxidizer pump inlet line.

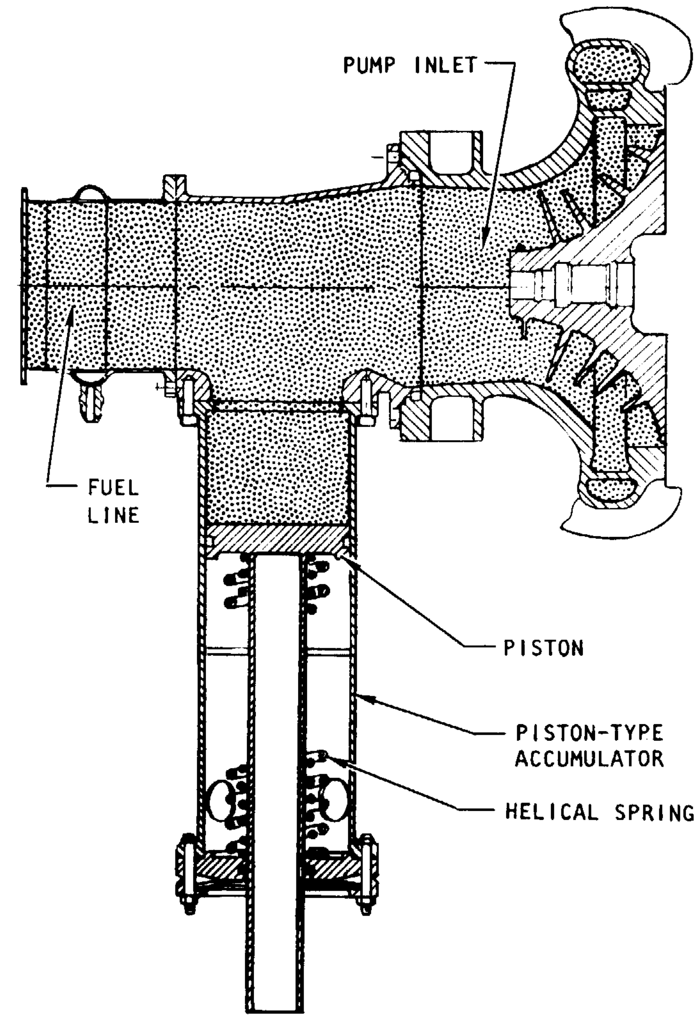


Figure 14. — Spring-loaded accumulator for Pogo suppression, Titan II fuel pump inlet line.

A Pogo suppression system is incorporated in the SSME LOX feed system at the inlet of the high-pressure oxygen turbopump (HPOTP). The system utilizes a gas-filled accumulator to suppress vehicle-induced flow oscillations. Gaseous oxygen tapped off the heat exchanger in the oxidizer-tank pressurization system is used as the compliant medium following an initial helium precharge. The system controls liquid level in the accumulator by means of an overflow line that routes overflow fluids to the inlet of the low-pressure oxygen turbopump (LPOTP).

The SSME Pogo suppression system is shown schematically in figure 15. As noted, the heart of the system is the gas-filled accumulator, which serves as a capacitance in the LOX flow circuit and prevents the transmission of the low-frequency (20 to 30 Hz) flow oscillations into the HPOTP. The system is sized to provide sufficient overflow at the maximum decreasing pressure transient in the LPOTP discharge duct. The engine controller provides valve actuation signals and monitors system operation.

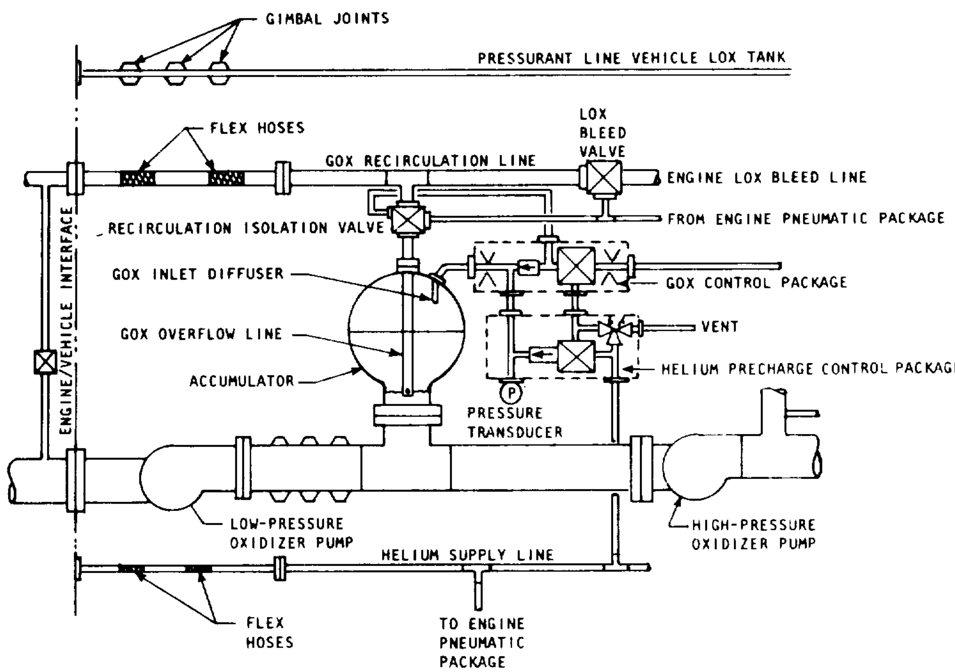


Figure 15. – Schematic of Pogo suppression system in LOX feed system on SSME.

2.1.5 Components

The line assembly as an operational unit is composed of a number of items, each performing a specific function. These components are discussed individually below.

2.1.5.1 SEPARABLE CONNECTORS

To reduce weight and overboard leakage, the number of separable connectors (static-seal disconnect joints) used in the line assembly is kept to a minimum. Orientation of the connectors in the engine or vehicle assembly must provide access for maintenance. Design details of separable connectors are presented in reference 29.

One of the requirements for a successful duct design is the use of connecting flanges or glands that are rigid enough to maintain the integrity of the static seal. The static seals for aerospace ducting are fairly sophisticated and expensive devices necessitated by the requirements for a reliable, lightweight system. Flanges must have the surface finish, radial clearances, and rigidity required by the specific seal used. Without sufficient rigidity, bolted flanges are particularly susceptible to rotation (fig. 16) under operational loads, and thus the effectiveness of the seal can be reduced. Inadequate bolting can cause the same effect under operational conditions. In general, it is desirable to keep the bolt circle of the flanged joint as close as possible to the duct diameter (i.e., to the seal) and to use many small-diameter bolts rather than a few large ones. The toe (outside diameter) of the flange acts as a reaction point to help prevent rotation.

Duct joints with quick-disconnect couplings such as V-bands have not been used successfully for high-pressure (>300 psi) applications. The clamp-type restraint cannot match a bolted

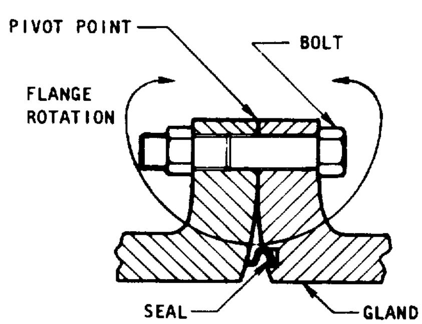


Figure 16. – Rotation of flanged joint as a result of internal pressure load.

flange for rigidity and seal-tight capability. V-band joints were used in the turbine exhaust ducts of early Atlas and Thor rocket engines, but it was difficult to maintain concentricity of flanges on mating ducts. The sheet-metal clamp also was unable to maintain sufficient axial load to prevent flange separation and leakage under operating conditions. These designs therefore were abandoned in favor of bolted flanges.

Not all V-band experience has been negative, however. A modification of the V-band concept was used on the Saturn S-IVB feed-duct-to-prevalve joint. In this design, a fully machined three-piece clamp, with a 300-psi proof level, was restrained with three tangential bolts.

2.1.5.2 MANIFOLDS

A manifold is a duct with one or more flow branches off the main stream (fig. 17). A T and a Y (figs. 17(a) and (b)) are examples of the simplest forms of manifolds; others, with

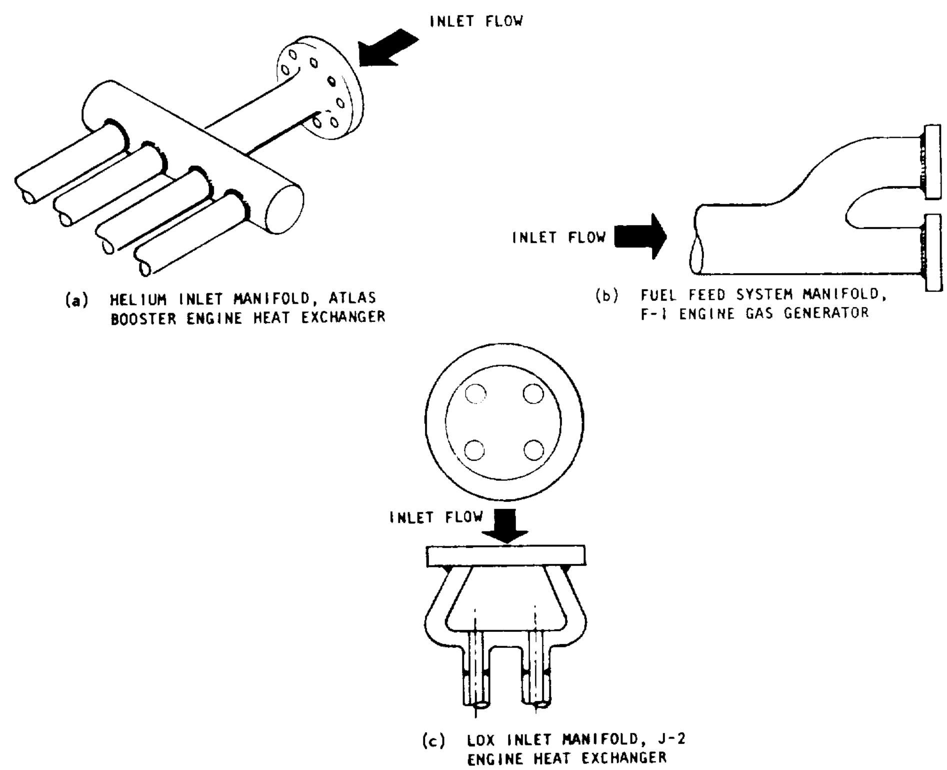


Figure 17. – Three examples of rocket engine manifolds.

interlocking flow passages, are so complex that they do not resemble ducts at all. The main function of a manifold is to distribute flow from one or more inlet passages to one or more outlet passages. The internal flow areas must be designed such that the flow is split among the branches as functionally desired. If the manifold causes an undesirable flow to a particular piece of equipment, it has failed in performing its function.

Design problems associated with manifolds involve fluid flow, structural strength, and cleanability. Since a manifold is a flow-distribution device, careful attention to pressure-drop analysis is paramount to a successful design. Experimental and theoretical data like those available in references 19 and 30 through 35 assist the designer. Laboratory flow tests of mockup or prototype hardware are performed to verify the analysis.

After the optimum flow passages have been established, they may have to be compromised in order to achieve a practical configuration for fabrication. Typical manifolds are welded assemblies, and the geometry of the branches with respect to the mainline does not always permit the ideal type of weld joint; mitered joints frequently are required.

Manifolds with many interconnecting passageways and blind ends are difficult to deburr, clean, and inspect for cleanliness. Angled passageways and blind holes are avoided wherever possible; straight-through passageways that are easily cleaned and inspected are preferred.

A typical turbine-exhaust-gas manifold for a liquid rocket engine thrust chamber is shown in figure 18. After the exhaust gases pass through the engine's pump-driving turbines, the gases

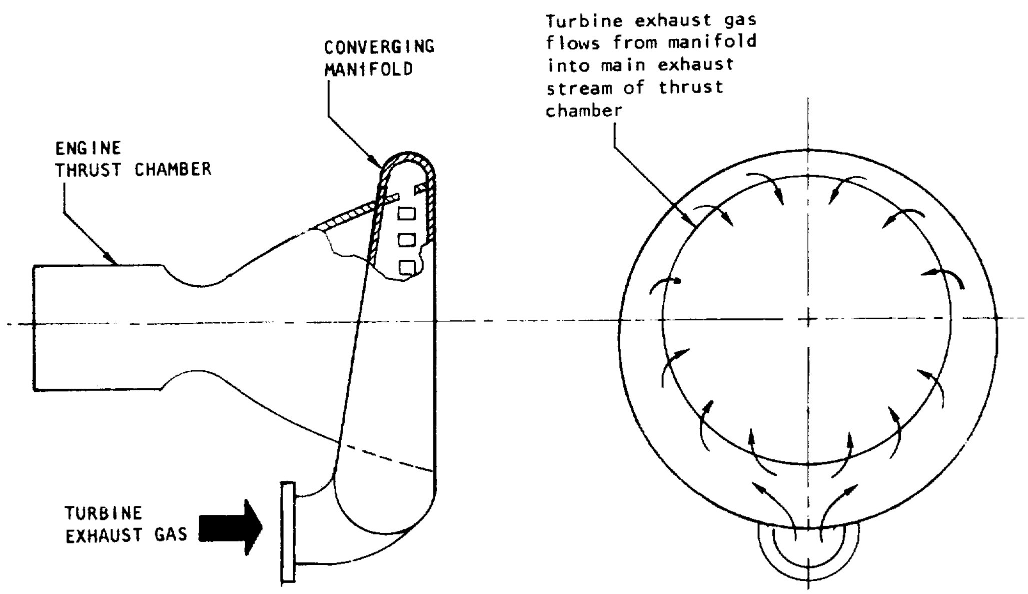


Figure 18. – Turbine exhaust manifold designed for equal distribution of flow.

are dumped into the mainstream exhaust of the thrust chamber through an annular manifold. The cross-sectional area of the manifold decreases as it wraps around the chamber, so that a constant gas velocity is maintained as the gases are bled off through openings in the thrust-chamber wall. The junction of the manifold and thrust-chamber wall does not permit the best structural design, and the condition is aggravated further by thermal gradients and engine vibrations.

2.1.5.3 BRACKETS, BOSSES, AND MOUNTING LUGS

Provisions for line mounting or support, auxiliary-equipment support, and instrumentation bosses are included in most duct assemblies. In assemblies of small lines (tubing, $\leqslant$ 1.25 in. diam.), these provisions are accomplished in several ways. Fittings machined from bars or forgings are butt welded into the tube assembly and can be utilized as brackets, mounting pads, or instrumentation ports. For simplicity, and in areas of low stress, bosses or mounting pads can be machined to fit the tube contour and then fillet-welded to the tube assembly; in all cases, consideration must be given to limiting the amount of weld dropthrough so as not to interfere with the flow of fluid in the line. Flanges also can be machined to include a mounting bracket or attach point, and instrumentaion bosses can be machined into the side of the flange.

In assemblies with line diameters greater than 1.25 in., mounting lugs or instrumentation ports generally are installed on the duct wall by fusion, spot, or seam welding. Spot- and seam-welded doubler-type mounting attachments have been satisfactory in low-stress, low-heat-flux applications; however, structural adequacy in high-heat-flux applications has not been satisfactory. In hot-gas ducts or hydrogen lines condensing liquid air on their outside surfaces, wall thermal gradients introduce shear stresses in welds that can cause wall distortion or weld failure. Doubler-type attachments are extremely undesirable from a contamination standpoint, because they can readily entrap fluids that can promote corrosion. One method of eliminating this condition is to fillet weld around the periphery of the doubler-type attachment.

The most desirable method of installing mounting pads or instrumentation bosses in large-diameter ducts is to machine the bracket or boss to a configuration compatible with the duct contour. These detail parts then are butt welded into the assembly and inspected radiographically to ensure high joint efficiency. In this joint design, sharp transitions in duct wall thicknesses are eliminated wherever possible, and radii of machined transitions are made generous. In large-diameter, thin-wall ducts such as those used in booster vehicles, full rings that match the duct diameters are butt welded into the duct assembly. These rings have mounting brackets or attachment points incorporated into the machining of the ring. Rings can also be utilized in duct assemblies in which one or more instrumentation bosses are an integral part of the machined ring.

Careful consideration is given to the material from which the rings are made. In many instances, bosses, fittings, and rings machined from 321-CRES bar stock leak during leak tests. Leakage is of the fuzz* type and is due to nonmetallic stringers in the parent material that are susceptible to fracturing when subjected to any material movement such as that likely to occur during welding, pressure testing, or actual operation. This leakage problem can be eliminated in several ways: (1) use of vacuum-melted 321 CRES; (2) use of forgings rather than bar stock; (3) use of 304L instead of 321 if lower mechanical properties can be tolerated.

2.1.5.4 INSULATION

Cryogenic feedlines are insulated to prevent heat transfer into the propellants for the following reasons:

- To maintain quality (temperature and density) of the propellants entering the engine pumps so that engine performance is not degraded
- To limit propellant boiloff during tanking and chilldown operations
- To limit propellant boiloff during long vehicle launch-hold periods
- To prevent geysering in the ducts
- To prevent two-phase flow and attendant increase in pressure losses during engine operation.

Cryogenic insulation can be foam, vacuum, vacuum/powder, or vacuum/super-insulation. Uninsulated ducts are also used in certain instances.

Although insulation is used extensively on vehicle propellant feedlines, the use of insulation on the lines of the engines is limited, because (1) the lines are relatively short with little surface area in comparison with long vehicle lines and (2) the mass flows are tremendous once the engine firing commences, so that insulation is of little use. A cryogenic duct that is warm during ground hold might well be left bare, since in space it will have the benefit of the ambient vacuum.

The majority of the engine ducts that are insulated have external insulation attached. The pump discharge and LOX pump inlet lines of the J-2 engines had a foam-type insulation installed after the engine assembly has been completed. Various lines on the F-1 engine were

*Leakage rate in the range 0.27 to 2.7×10^{-3} scc/sec.

equipped with close-fitting nickel-foil/Refrasil blankets to prevent overheating of the line and possible subsequent structural failure due to the heat radiated from the exhaust plume of the five F-1 engines in the Saturn V first stage. The use of external insulation has little effect on the detail design of ducts or lines, but may influence the designer's choice of line routing because of the need to allow space for the desired insulation.

On the Saturn S-II and S-IVB stages, extensive use was made of vacuum-jacketed fuel (LH_2) feedlines of the initially pumped type. The lines jacketed on the Saturn-II stage were the fuel feedlines, the fuel recirculation lines, fuel vent lines, and the LOX feedlines. On the J-2 engine, vacuum jackets were used on the inlet duct of the gimballed fuel pump and the gas-generator fuel bleedline. Both these engine lines were in the form of concentric metal bellows with a cryopumped vacuum.

The main design considerations for the application of vacuum insulation to ducts and lines are as follows:

- Choice of an initially pumped vacuum (lowest heat leak) or a cryopumped vacuum
- Proper ratio of outgassing-surface area to vacuum-chamber volume
- Proper vacuum-jacket cross-sectional area to permit effective pumpdown
- Jacket supports with low-heat-leak paths
- Cleanability
- Provision for pressure-relief devices
- Provision for evacuation valves
- Provision for vacuum measurement

The magnitude of the allowable heat leakage will govern the choice of a cryopumped vacuum or an initially pumped vacuum (lowest heat leak). Cryopumped vacuums in vacuum jackets initially containing air at atmospheric pressure have proved satisfactory for liquid-hydrogen lines when the jackets were leak tight. A pure gas condensible at operating temperatures provides a better cryopumped vacuum than air, since air contains traces of noncondensible gases. Some choices for liquid-hydrogen lines are carbon dioxide, nitrogen, and argon. With a condensible gas, attention has to be given to the effect on heat transfer resutling from possible dislodgement of condensed frost by vibration, and to a method of ensuring that the gas quality is adequate prior to cryopumping.

The most significant problem with vacuum-jacketed engine lines has been controlling the leakage of atmospheric air into the jackets. Until recently, the jackets have been provided with burst diaphragms or relief valves to relieve internal pressure resulting from expansion of air that leaked into the cryopumped vacuum. Burst disks (diaphragms of sufficiently low burst pressure to protect the line) have been fragile and have leaked under engine vibration. Relief valves have leaked and then frozen and failed to protect the line during warmup. Because the safety device itself often has caused the leakage problem it was designed to prevent, current practice has been to seal the jacket with a plug-and-seal combination or a welded plug after a mass-spectrometer leak test has been performed on the entire jacket. Current acceptable assembly leakage rate is 1×10^{-6} scc/sec of helium.

An all-welded jacket construction requires that the cryopumped gas filling the jacket be compatible with the closing welding process. Argon gas has been successfully used in this application to provide an inert-gas backup to the closing weld.

In low-heat-leak applications such as the large-surface-area propellant feedlines of booster stages, initially pumped high-vacuum insulation is used. High-vacuum insulation consists of an evacuated space between two highly reflecting surfaces. Heat is transferred principally by radiation, but there may be some contribution from gas conduction.

Utilization of vacuum insulation requires careful design and precise fabrication techniques. If the vacuum space has the structural duct walls as one of its boundaries, this wall must be tight enough at cryogenic temperature to prevent leakage of contained fluid into the evacuated space. The other boundary of the insulation space usually is exposed to ambient temperature. This wall must be leaktight under all environmental conditions to which it will be exposed. Means must be provided to maintain the vacuum integrity in spite of the differential contraction of the cold and warm boundaries.

Helium leak-detection techniques involving the mass spectrometer are used during fabrication. The allowable leakage rate depends on the degree of vacuum required for optimum insulation performance and the volume of the insulation space. Leakage rates of 10^{-6} to 10^{-7} scc/sec of helium are common and are attainable with materials and welding techniques generally used.

Mechanical joints such as threaded fittings or flanges are not desirable within the vacuum enclosure unless some provision is made to weld the joints so that they are leak tight.

It is desirable to maintain a low ratio (< 4) of outgassing area to vacuum-annulus volume, because this ratio directly affects the rate of vacuum decay. For example, with identical line lengths of annuli equally clean and at the same vacuum level, the vacuum in a line with a ratio of 8 will decay at a rate twice that of a vacuum in a line with a ratio of 4. The part with a ratio of 8 therefore has higher maintenance costs for re-evacuation.

The cross-sectional area of the annulus also is important in that it should be sufficiently large to permit effective vacuum pumpdown within a reasonable amount of time. Also, the structure that supports the jacket shell on the mainline should not restrict molecular flow during the pumpdown period.

In low-heat-leak applications, the heat-leak paths from the jacket to the inner carrier through supports and flanges must be minimized. Supports are required to maintain the outer shell of a vacuum line and to attach the ducting to the structure itself in all cases. The longest possible heat path with minimum contact area is desirable. Stainless steel often is used when both structural strength and comparatively low thermal conductivity are desired; several nonmetallic materials (Micarta, Teflon, Kel-F) are applicable when the lower strength is acceptable in favor of improved thermal resistance.

Cleanability of the vacuum jacket surfaces is an important factor in being able to achieve and maintain a high vacuum. Any material that has been exposed to air will have a layer of gas and vapor several molecules thick on its surface; in addition, the interior of metals contains dissolved or occluded gas that is trapped during solidification. Adsorption on freshly exposed surfaces is very rapid, and the adsorbed gas tends to reach an equilibrium with the environmental pressure; evacuation for a long period of time will desorb the gases. Application of heat during evacuation will speed the process.

Complete elimination of adsorbed gases is very time consuming and expensive. To attain high vacuum for long periods of time, therefore, it is necessary to make use of adsorbents and getters in the vacuum system. Adsorbents such as metallic zeolites, activated charcoal (not compatible with liquid oxygen), or silica gel are used to adsorb gases near liquification at the temperature of the contained cryogen. For this reason, the adsorbent package is located in a low-temperature area.

Hydrogen often is an important constituent of the adsorbed gases. Because of its low boiling point, it is less easily picked up by adsorbents, and therefore even a very small quantity of hydrogen will preclude good vacuum performance. Getter materials are used to remove this hydrogen. The basic difference between a getter and an adsorbent is that a getter cleans up gases by chemical reaction, whereas adsorbents work by surface adhesion. Getters usually will produce very low pressure levels independently of temperature but the chemical-reaction rate is speeded up by elevated temperatures. The practice therefore is to locate the getters in the warmest portion of the insulation system. Common getter materials include active metals, active zeolites, and weak oxides. While getter materials are used largely with jacketed lines in facility or ground support equipment, the concept has direct application in flightweight lines.

Provisions must be made in the duct design for mounting service valves to effect evacuation of the vacuum jacket. These valves usually are welded into the vacuum jacket, as is the case with the S-II stage lines. Problems with Viton and Kel-F O-rings have occurred when these valves were exposed to temperatures lower than -65°F. Metal-to-metal seals usually alleviate this problem.

Provision also must be made with initially pumped vacuum-jacketed lines for vacuum measurement. Thermocouple gages commonly are used to fill this requirement. Data on this accessory are available in reference 36.

2.1.5.5 HOT-TO-COLD DUCT INTERSECTIONS

In some line assemblies, a line carrying a cold fluid (e.g., liquid oxygen at -297°F or helium at -360°F) must intersect and pass through a duct containing a hot fluid (e.g., turbine exhaust gas at +1300°F). One particularly difficult situation arises in turbine-exhaust heat exchangers where supply and outlet lines of the cold fluid pass through the wall of the hot-gas turbine-exhaust duct. Reliability of this interface is of utmost importance when the cold fluid is oxygen, because the leakage of oxygen into the fuel-rich turbine exhaust gases can create temperatures sufficiently high to melt and burn the metal.

Many designs have incorporated either tubes welded directly to edges of holes cut in the duct wall or tubes connected to thick flanges welded flush to the duct wall. Neither approach has been satisfactory for reducing thermal stresses or providing reliable welds. A design used on the J-2S engine incorporates a flanged manifold with stubouts onto which the tubes are welded. The manifold in turn is welded to a somewhat flexible flared portion of the hot-gas duct wall. This configuration provides a pocket of stagnant gas to protect the cold manifold from high heat fluxes and thermal stresses and also provides a flexible interface between hot and cold members. All weld joints are butt welded and are inspectable radiographically.

2.1.5.6 ELBOWS

In low-pressure systems where tube wall thickness is small with respect to tube diameter (tube wall less than 1/20 tube diameter), and where bend radii are small (bend radius less than 4 times tube diameter), large concentrations of bending stress occur in elbows. The stress may be five or more times the direct bending stress. This stress can result in cracking along the mean radius of the elbow unless appropriate design provisions are made (ref. 37).

2.1.5.7 HANDLING-PROTECTION DEVICES

Damage to the bellows in the duct assembly and to the duct walls themselves in the case of thin walls ($\leqslant$ .030 in.) during processing, shipping, and installation is a continuous problem. Damage to the static-seal sealing surfaces of the duct flanges also is a problem, and protective covers must be used.

Ducts having low-spring-rate bellows can easily be abused in handling by overdeflection of the bellows. Since the concern is fatigue life of the bellows, a few cycles of gross overdeflection are equivalent, fatigue-wise, to many cycles of normal operational deflection. Strongback fixtures that attach to the duct on either side of the bellows joint and form a bridge that prevents handling deflection are a good solution for this problem. Figure 19 illustrates a strongback fixture and one type of bellows cover.

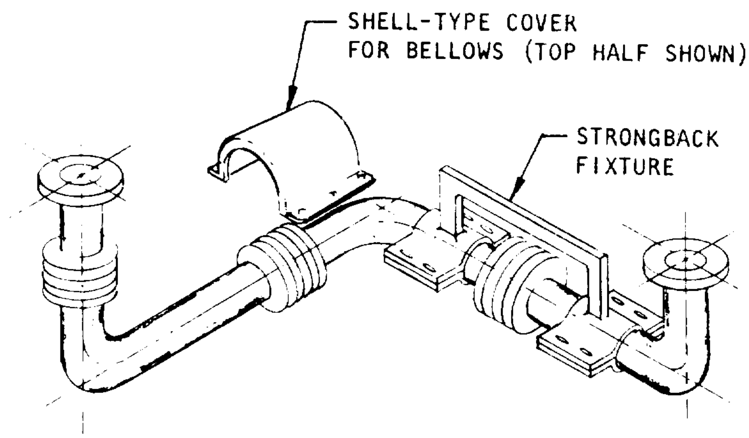


Figure 19. — Handling-protection devices for bellows.

It is preferable to design all protective covers and handling devices so that the parts they are protecting cannot be installed or assembled into the next assembly without removal of the protective device. If such design is not possible, the minimum requirement is that a red flag be used to warn the user that the protective device is installed.

Damage to bellows due to handling during duct fabrication, shipment, and installation into the subsequent assembly is a major contributor to eventual in-service failure. Dents, nicks, and scratches in convolution crowns (the most highly stressed area in a bellows) cause stress concentrations under operating conditions and thus reduce fatigue life considerably. Life-reduction criteria based on coupon and bellows fatigue tests have been developed (ref. 38). These data permit material review action on damaged items in the final product that can result in acceptance of damaged parts that would otherwise be rejected.

2.1.6 Materials

As noted, the 18-8 corrosion-resistant steels are the most frequently used materials for bellows and lines in most rocket fluid systems; over a considerable length of time, these materials have performed well. The nickel-base alloys such as the Inconels and the

Hastelloys are widely used in newer bellows and duct designs because of their higher strengths, greater fatigue lives, and improved corrosion resistance. Aluminum alloys are used sparingly in some noncritical or low-stress applications. Because of the severe forming requirements, only the high-ductility alloys in the annealed condition are used for convoluted sections. Heat-treatable materials are heat treated after forming. Bellows-joint restraint bracketry in cryogenic propellant systems is fabricated from the face-centered-cubic metals to ensure adequate low-temperature toughness. Restraint bracketry in storable-propellant systems is fabricated from high-strength body-centered- or face-centered-cubic steels. The majority of flexible hoses (including wire braid) are fabricated from 321 CRES. Other materials, such as Hastelloy C and Inconel 718, are used for special environmental applications. Table I lists the typical alloys used for the fabrication of the subject parts in various rocket vehicle and engine systems.

Material selection considerations for the entire line or duct assembly include chemical compatibility with environmental fluids, physical and mechanical properties, formability, weldability, and the cost of the materials available.

2.1.6.1 CHEMICAL COMPATIBILITY

The importance of chemical compatibility of line material with system and environmental fluids cannot be overemphasized. For example, a high-pressure gaseous-hydrogen environment is known to embrittle many metal alloys (refs. 39, 40, and 41). The degree of embrittlement is a function of the material temperature, environmental pressure, purity of the gas, exposure time, and applied strain level. Steels (ferritic, martensitic, and bainitic), nickel-base alloys, and titanium alloys become embrittled in pure-hydrogen environments at room temperature; the effects are more pronounced as the pressure increases. High-strength alloys are more susceptible than low-strength alloys. Austenitic stainless steels such as types 310 and 316; certain aluminum alloys such as 6061-T6, 2219-T6, and 7075-T73; pure copper and beryllium copper; and the precipitation-hardened austenitic stainless steel A-286 are only slightly affected. Inconel 718, Inconel X-750, Waspaloy, and Rene 41 are severely embrittled in high-pressure gaseous-hydrogen environments.

The degradation of mechanical properties by embrittlement generally is reflected in a loss of ductility and fracture toughness. A large amount of experimental work currently is being conducted by industry and government laboratories to achieve better understanding of hydrogen embrittlement and its effects. This recognition and concern for hydrogen-environment embrittlement is part of a growing recognition of the sometimes dramatic and insidious effects of chemical environments on material properties and hardware performance. The gross effects of corrosion and the less conspicuous effects of stress corrosion are generally well known, and designers are alert to them.

The presence of hydrogen, water vapor, and other gases can (and does) have a pronounced effect on crack initiation and crack growth rates. This change in turn affects conventional mechanical properties and can drastically reduce fracture toughness. Recognition of this problem is seen in the fracture-mechanics* approach to the design and development of high-performance hardware. The determination of plane-strain fracture toughness under identical proof and service conditions is essential to the realistic application of these concepts.

Chemical compatibility of the line material and environmental fluids (internal and external) thus involves the following factors:

- Corrosion rates of the material (low corrosion rates are desirable, and the nickel-base alloys have performed well).
- Impact ignition sensitivity of the material (in oxygen, titanium can ignite and oxidize explosively).
- Catalytic decomposition of propellants (N_2H_4, Aerozine-50, or MMH will decompose into large quantities of hot gas in the presence of molybdenum, iron, copper, or silver) (refs. 42 through 45).
- Material embrittlement and crack initiation and crack growth rates (as discussed above).
- Acid attack (absorption of water in N_2O_4 forms nitric acid, which will attack aluminum alloys).
- Stress corrosion (stress-corrosion cracking has been observed in 321-CRES bellows and is attributed to chlorides from cleaning fluids or unknown sources deposited on the convoluted surfaces; titanium alloys are susceptible to stress corrosion in uninhibited (brown) N_2O_4).

2.1.6.2 PHYSICAL AND MECHANICAL PROPERTIES

Physical and mechanical properties of importance are primarily high strength (ultimate, yield, and fatigue), high elongation capability, and low density. A low elastic modulus is helpful when hard lines require some flexibility. In high-heat-flux applications, high thermal conductivities and low thermal expansion coefficients produce the lowest thermal stresses. Also, joints of dissimilar materials have the lowest thermally induced stresses when the materials have similar thermal expansion coefficients. Forming limits (minimum bend radii) are important considerations in the design of bellows-convolution shape. Fatigue data are most useful for design of convoluted elements, since the rest of the line usually is designed for stresses under yield. Some reduction of air-test fatigue allowables is made for aluminum alloys and corrosion-resistant steels used in N_2O_4 (ref. 46). Cryogenic temperatures reduce

*A brief discussion of fracture mechanics appears on page 46.

the toughness of most body-centered-cubic materials. Data on many alloys are available (ref. 47) and are useful in determining the acceptability of alloys that suffer large reductions in fatigue strength. Table III presents data on representative alloys used in flexible lines.

Material selection considerations for the convoluted portions of bellows joints or flexible-hose elements are more extensive than those for the remainder of the line. The most important factors are material formability requirements, corrosion properties, and low-cycle-fatigue properties. The metal-working operation in forming (convoluting) a bellows or a flexible-hose innercore requires a material of high ductility, more so than for the forming of the tubular portion of the line. Since the convoluted portion is difficult to clean and usually is made of thinner material (typically 0.005 to 0.030 in.) than the rest of the line, corrosion resistance is important if the section is to remain free of pits, holes, cracks, or other leakage paths. Corrosion-resistant coatings or platings have not been effective because coverage is inadequate. Coating integrity is difficult to maintain during forming, and uniform application is difficult after forming. Low-cycle fatigue properties are important in that the bellows are designed for a given number of flexural cycles, while the rest of the line is designed to keep alternating stresses below the endurance limit of the material. A sophisticated and expensive way to improve corrosion resistance and fatigue life of bellows is to electropolish them prior to forming.

Material characteristics desirable for use in hard lines are (1) high strength-to-weight ratio, (2) good weldability, and (3) good elongation. The following materials have the desired characteristics and provide minimum weight for the given categories of rigid lines:

- Inconel 718 (high-pressure lines > 1.25 in. OD)
- Armco 21-6-9 (high-pressure lines < 1.25 in. OD and low-pressure lines > 0.75 in. OD)
- 321 and 347 CRES (all lines < 0.75 in.OD)

Inconel 718 is used in the high-pressure, large-diameter lines of the SSME primarily because of its high strength-to-weight ratio. Inconel 718 may be used in lines that carry hydrogen and operate at cryogenic temperatures, where hydrogen embrittlement is not a problem.

Armco alloy 21-6-9 can be used for the intermediate-size lines, since these lines are gage limited. This alloy is highly resistant to hydrogen embrittlement. The elongation capabilities of Armco 21-6-9 are comparable to those of Inconel 718: 10 percent at cryogenic temperature and significantly better (40 percent) at room temperatures. This alloy is annealed and does not require heat treatment after welding.

321 and 347 CRES can be used in the small lines, which are also gage limited. Hydrogen is not a problem with these alloys. These materials have elongation capability equal to or better than Armco 21-6-9 and require no heat treatment after welding.

Table III. – Typical Properties, Forming Limits, and Relative Cost of Representative Alloys Used in Line Assemblies

Parameter	Alloy					
	6061-T6	321	Hastelloy C	Inconel 625	Ti-6Al-4V, annealed	Inconel 718, age-hardened
Mechanical properties at 80°F						
Ultimate, ksi minimum	42	75	110	120	130	180
Yield, ksi minimum	36	30	50	60	120	150
Elongation, percent minimum	10	40	40	30	10	12
Reduction of area, percent minimum	45	40	12	35	20	15
Fatigue strength, 10^7 cycles, $R = -1$, ksi	20	23	42	45	64	46
Strength/Density ratios at 80°F						
Ultimate/Density, 10^3 in.	428	262	340	394	810	604
Yield/Density, 10^3 in.	367	105	154	197	750	504
Physical properties at 80°F						
Modulus of elasticity, 10^6 psi	9.9	28.5	29.8	29.8	15.5	29.6
Thermal expansion coefficient, 10^{-6} in./(in.- °F)	13.0	8.5	6.3	7.1	4.9	7.1
Thermal conductivity, Btu/(hr - ft. - °F)	96.0	9.0	5.0	6.0	4.0	7.0
Density, lbm/in.^{3}	0.098	0.286	0.323	0.305	0.160	0.298
Forming limits, annealed condition, minimum radii						
Bend radius, 0.060-in.-thick sheet	1.5t	1.5t	2.5t	2.5t	6t	2.5t
Bend radius, 3-in. D_o x 0.060-in.-wall tube	1.5D_o	1.5D_o	2.3D_o	2.3D_o	6D_o	2.3D_o
Relative line cost						
6 - ft. long x 3 - in. D_o, two flanges, four bends	1.0	1.5	3.5	3.7	4.5	3.5

$R = -1$: imposed strain in fatigue testing is equal in both directions from neutral

t: thickness of sheet, in. D_o: outer diameter of tube being bent, in.

Properties at cryogenic temperatures are given in reference 62.

Properties at elevated temperatures are given in reference 63.

Bolt and nut materials used in aerospace applications are inherently corrosion-resistant, the most frequently used materials being A-286, Inconel 718, Inconel X-750, K-Monel, and Rene 41. Material selection is based on the strength level required for the coupling and the service temperature. All of the above materials except Rene 41 are used in applications ranging from cryogenic temperature to 1200°F. Rene 41 is normally used at temperatures in the 1200°F to 1600°F range because of good strength levels, but high cost restricts it from being used at lower temperatures. Additional details on material selections for threaded fasteners is provided in references 48, 49, and 50.

Fracture-mechanics properties. -- Conventional structure analyses account for stress, material strength, and stress-concentration factors at fillets, corners, and holes, but they do not consider flaws and minute cracks inherent in all materials. When ductile materials such as those used in line assemblies are stressed, they deform plastically around the tips of the cracks and absorb local overloads. But when a material cannot behave in a ductile manner -- for example, when it is near or below its transition temperature, when strain rate is high, or when the stress system is complex and the material is physically constrained -- it can fail at a relatively low stress level. For these conditions, the concepts and techniques of fracture mechanics can be utilized for analysis.

The concept of fracture mechanics recognizes discontinuity and nonhomogeneity in materials and provides a numerical method to account for minute cracks. The fracture-mechanics design parameters are a toughness (energy-absorption) factor, applied stress, size of crack, operating temperature, and the "state of stress", which is often far more complex than first assumed.

Discussion of fracture-mechanics theory and practice is beyond the scope of this monograph. More detailed information on the fundamental theory and its application to ducting and filter design, including material selection, selection of allowable working stress, evaluation of fabrication methods, evaluation of inspection and testing methods, and design of the proof test, may be obtained from references 51 through 60.

2.1.6.3 FORMABILITY

Formability (high ductility) of materials is important in that most of the line assembly, with the exception of flanges, bosses, or linkages, is formed from sheet metal. The least severe forming operation is rolling of the tubular elements, followed in increasing severity by forming of (1) bends, (2) fittings and transitions, and (3) the bellows convolutions. To date, corrosion-resistant steels and nickel-base alloys have been excellent materials for all formed elements.

2.1.6.4 WELDABILITY

Weldability of materials is an important consideration, since welding is the most often used method of permanently joining line-assembly elements. In several aerospace programs (e.g., Apollo), brazing is used for permanent line connections, and the braid of flexible hoses frequently is brazed to the end attachment joint. Most materials in current use are effectively welded by tungsten-inert-gas methods. Electron-beam welding has been used effectively with both Hastelloy C and 347 CRES. Dissimilar materials often are welded (e.g., Hastelloy C to 347 CRES, Inconel 718 to A-286, Inconel 718 to 321 CRES, and 321 CRES to A-286).

2.1.6.5 LUBRICANTS

Sometimes a lubricant is required on the bellows or on other surfaces. Molybdenum disulfide dry-film lubricants have been used, but often chemically attack corrosion-resistant steels when moisture is present. Nickel-base alloys offer the best resistance to attack. A corrosion-inhibiting type of molybdenum disulfide coating, satisfactory on corrosion-resistant steels, recently has been developed (ref. 61).

Platings and lubricants are used on bolts and nuts to prevent thread galling. Silver is the most widely used plating, although others are used when required to solve fluid compatibility problems. The standard thread lubricant used on the Saturn engine systems was a phosphoric-acid-bonded dry-film lubricant. One characteristic that influenced the selection of this lubricant was its impact ignition compatibility with liquid oxygen. Useful temperature range for this dry-film lubricant is from cryogenic temperatures to +300°F. Platings are used at higher temperatures.

2.1.7 Testing

Testing usually is divided into two categories: (1) the initial qualification testing, which is later repeated on a sampling basis during production runs; and (2) acceptance testing, which consists of nondestructive tests made on all production pieces. Typical tests and the categories for which they apply are given in table IV.

2.1.7.1 TEST REQUIREMENTS

Experience has shown that component testing is extremely beneficial in locating design and manufacturing weaknesses early in the development cycle, thus permitting redesign prior to actual engine use where failures would be much more costly and time delaying. The component tests are designed to simulate engine or operational conditions as closely as

Table IV. – Typical Test Requirements for Line Assemblies

Type of test	Qualification requirement	Sampling plan requirement	Acceptance requirement	Test description
Proof pressure	Yes	Yes	Yes	Apply hydrostatic pressure to the part for predetermined number of cycles, cycle durations, and pressure levels. No leakage or permanent distortion allowed.
Leak test	Yes	Yes	Yes	Apply pneumatic pressure to part at predetermined pressure level and duration. Leakage measured must be within predetermined tolerable leak rate.
Examination of product	Yes	Yes	Yes	Inspect for conformance to dimensional, material, functional, and processing requirements set forth on drawing.
Movement verification	Yes	Yes, if considered critical	Yes, if considered critical	Verify design movements by moving through maximum excursion.
Spring rate (force/deflection)	Yes	Yes	Yes, if considered critical	Determine force or moment versus deflection (angular, axial, lateral, or torsional) characteristics.
Buckling stability	Yes	Yes, if considered critical	Yes, if considered critical	Apply proof pressure with flex joints in extreme deflected position.
Flexural endurance	Yes	Yes	No	Deflection cycle under operational pressure and temperature conditions for a predetermined duty cycle without fatigue failure.
Mechanical vibration	Yes	Yes	No	Vibrate bellows or assembly mechanically through engine or vehicle vibration spectrum and life specification under operational pressure, temperature, deflection, and environmental conditions. No failure allowed.
Flow calibration	Yes, if considered critical	Yes, if considered critical	No	Measure fluid flow head loss due to friction and inertia losses at rated flowrate conditions.

(continued)

Table IV. – Typical Test Requirements for Line Assemblies (concluded)

Type of test	Qualification requirement	Sampling plan requirement	Acceptance requirement	Test description
Flow fatigue	Yes, if considered critical	No	No	Flow end-use fluid (or fluids) through part over operational flowrate range for engine or vehicle specified life. No failure allowed. See note 1.
Pressure impulse	Yes	Yes	No	If part is subjected to pressure oscillations or impulses in operational service, apply a predetermined number of cycles; no failure permitted.
Sectioning and thickness measurement	Yes	Yes	No	Cross section part after completion of tests; measure conformity with drawing dimensions, particularly wall thinning in bellows.
Burst	Yes	Yes	No	Apply hydrostatic pressure to part at up to specified burst-pressure level. Yielding permitted, plastic distortion permitted, but no leakage.

Notes:

1. Unjacketed flight bellows flowing cryogenic fluid should be tested in an environment precluding condensation or air liquification, because either condition could provide viscous damping of vibration.
2. For economic reasons and for better simulation of operational use, some of these tests may be conducted simultaneously For example, the flow calibration may be done during the flow-fatigue test for practical reasons, and mechanical vibration tes s may be combined with flow-fatigue tests.

possible so that all failure modes can be identified. The flow-induced vibration failure of Saturn AS-502 flexible hoses in flight (ref. 64) is a case in which the lack of vacuum-environment simulation during component testing was the reason that a significant failure mode was not discovered prior to flight.

The mechanism for triggering flow-induced vibration in a bellows or hose is a function of the fluid heat-transfer characteristics, momentum, and bulk modulus. No dimensionless parameter for correlating data from one fluid to another is known; the behavior of a bellows flowing a fluid other than the end-use fluid may be entirely different from the behavior with the actual fluid. For this reason, the current practice is to use actual end-use fluid for flow testing.

There are two general schools of thought on the number of samples to be tested in the initial qualification of a new design. The NASA requirement is that a minimum of two samples go through all of the planned tests in sequence, decathlon style. Industry practices have sometimes been to run separate samples for each major test (vibration, flexural endurance, etc.), but with more time or cycles applied than required in the NASA method. The samples usually would be tested to destruction after the minimum requirements had been met, thus giving a measure of the part's margin of safety for each type of critical test. A discussion of test requirements and conditions is presented in reference 16.

The effect of environmental temperature on fatigue life of metals (in standard coupon shapes) is well known. In general, fatigue life is reduced at elevated temperatures and increased at cryogenic temperatures. For example, the life of 321-CRES bellows at liquid-nitrogen temperature (−320°F) is 4 to 5 times greater than that at room temperature for a given strain level; at liquid-hydrogen temperature (−423°F), life is 10 to 100 times greater. However, very little fatigue testing at high or low temperatures actually has been performed on bellows for design-evaluation or production-comparision tests. Usually, acceptability is based on results of a room-temperature test that gives little more than an indication of actual life at operating temperatures. For example, for a Hastelloy-C bellows on a J-2S heat exchanger, the fatigue life at 800°F, under the same deflections, was approximately one-third the life at room temperature. Testing at operational temperatures during initial qualification tests provides a good basis for assurance that the later quality-control sampling tests at room temperature are adequate.

For bellows intended for aerospace application, manufacturing procedures normally require proof- and leak-test certification of every piece in a production lot. Proof testing is primarily a structural integrity test. Water rather than gas commonly is used as the test medium because, in case of a burst, much less energy is released from water. Distilled or deionized water is employed, because tap water can deposit chlorides on the bellows surface and promote cell corrosion or intergranular corrosion within a very short time (refs. 65 and 66). For bellows or ducts that operate at elevated temperature, it is common practice to perform the acceptance proof-pressure test at room temperature. The test pressure level is increased to compensate for the effect of elevated temperature on material properties. The proof pressure simply is multiplied by the largest ratio of the room-temperature value to elevated-temperature value for yield stress, ultimate stress, or elastic modulus; this pressure then is used as the test pressure.

The mass spectrometer is a generally accepted device for verifying bellows leak tightness. The mass-spectrometer leak detector (MSLD) is capable of detecting leakage rates (i.e., has a leak sensitivity) of about 10^{-9} scc of helium/sec. The MSLD test is preferred for accuracy over the test with gas-pressurized bellows submerged in water, because the surface tension of the water is capable of sealing minute pinholes or cracks in the bellows wall, and at small leakage rates the water can dissolve a leak bubble as it is formed; leak sensitivity with bubbles is 10^{-2} scc/sec. Soap or detergent leak-detection fluids have been found to cause stress corrosion of bellows materials; however, a neutral aqueous solution of surfactants (per ref. 67) has been used safely and has a sensitivity of 10^{-4} scc/sec.

The end-use configuration is simulated as closely as possible in test setups so that all conditions to which assemblies will be subjected can be evaluated. For example, both the J-2 and F-1 engines have numerous fluid-carrying lines that must cross the interface between engine and vehicle. The lines are flexible to accommodate gimballing motion. On the J-2 engine, all the lines are flexible hoses and are clamped together to assist in maintaining their relative positions. This clamping affects the distribution of the line assembly deflection among the individual flexible-hose sections in the line and adds to their overall stiffness. On the F-1 engine, the flexible hoses cross the gimbal plane and are attached to nongimballing flexible hoses on the engine; this configuration influences their deflection. Good practice, then, is to test all lines clamped together as a bundle.

Acceptance tests are performed for quality assurance on all welded components. Circumferential weld dropthrough can cause an increase in pressure drop of hose assemblies that is sufficient to affect engine performance. Small-diameter (< 1.25 in.) lines are especially sensitive and are checked during acceptance by passing a ball gage completely through the assembly.

Testing of aerospace ducting involves extremes of temperature, large pressure forces, high-pressure gas generation from cryogenic fluids, combustibility, and toxicity; considerable caution therefore is exercised in the design and operation of test programs. References 16 (p. 30) and 68 present details on test programs.

2.1.7.2 TEST INSTRUMENTATION

Typical parameters measured on lines, bellows, and flexible hoses during development and qualification testing include static and total pressure, temperature, flowrate, strain, and vibration response. Pressure and temperature measurements are obtained with standard low-frequency pressure transducers and thermocouples. Orifice meters, venturi meters, and turbine flowmeters are utilized for flowrate measurements. Turbine meters must be used with caution in cryogenic service where overspeeding can occur during chilldown of the system, when the gas velocity is much higher than the liquid velocity for which the meter is design; blade or bearing failure can be the result. Pressure, temperature, and flow measurement techniques outlined in reference 69 are extremely useful.

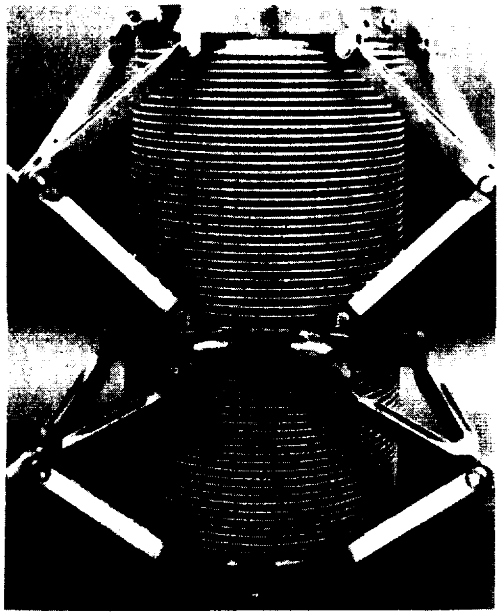
One of the most recent developments in instrumentation, unique to flexible ducting, has been the technique for measuring frequencies of flow-induced vibration and "g"-forces on bellows. Methods evaluated involved microphones, radiography, strain gages, and miniature accelerometers. The accelerometers proved most effective for braided flexible hoses, whereas strain gages, mounted on the crowns of bellows convolutions, were most effective for bellows. The attrition rate on gages during flow tests is high (up to 50%), especially under cryogenic conditions. Strain gages have been used on bellows inside wire-braided flexible hoses, but installation of gages requires great care, whether they are installed before the braid is assembled to the hose or afterwards, when the braid wires must be compressed locally to form a window for the innercore; the braid window is retained by local soldering to permit access for affixing strain gages to the convolution.

Vibration frequencies of bellows, particularly those of pendulum mode in braided hoses, can be as high as 60 kHz under cryogenic conditions. Frequencies of this magnitude normally are filtered out in the measuring circuit of the standard, high-frequency, rocket-engine-vibration instrumentation. By removing the filter and taking advantage of the voltage-vs-frequency curve to complement the voltage-vs-temperature curve, a usable "flat response" output within approximately 10 percent of the mechanical/electrical resonance of the mounted accelerometer can be obtained.

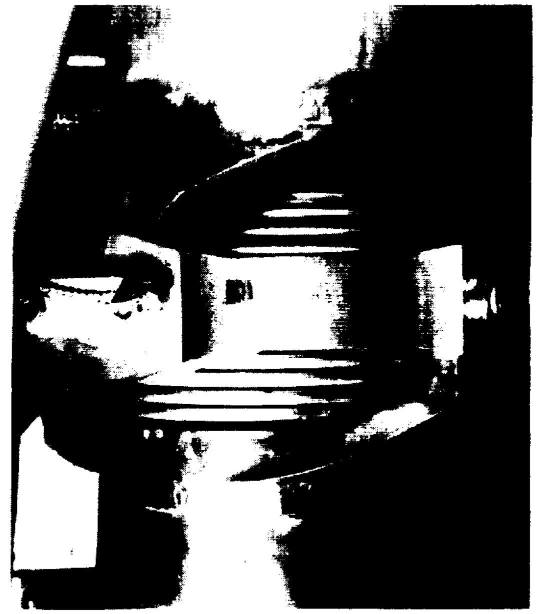
2.2 BELLOWS JOINT

A bellows section (or joint) in a line generally consists of three parts: a flexible pressure-carrying convoluted tube (the bellows), a restraint on the bellows to handle the pressure separating load, and a provision for attaching the bellows to the line. In most cases, a fourth part – a flow liner for the bellows – is incorporated. Photographs of typical bellows joints are presented in figure 20.

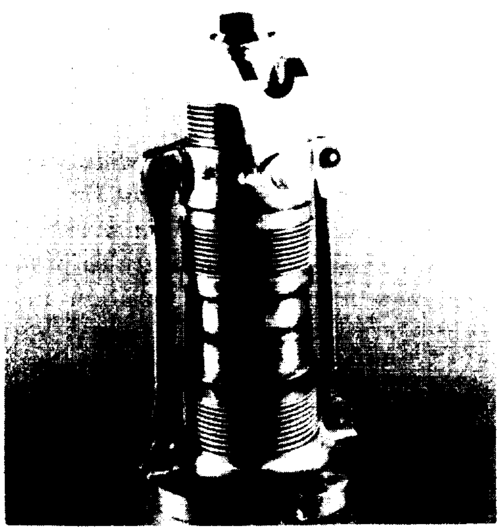
A bellows joint in ducting can absorb motions (deflections) with much smaller reaction loads than those that develop in a hard line. A free bellows can absorb four kinds of motion applied to its ends: (1) axial (extension or compression), (2) offset (with end planes parallel), (3) angular (about its center), and (4) torsional (about its axis). Typical applications may impose one, all, or any combination of these end-point motions (deflections) on a bellows. Various restraining devices or linkages are designed to limit a



(a) Compression-type bellows externally restrained against buckling



(b) Externally restrained, double-shear gimbal-jointed bellows



(c) Externally restrained tension-tie bellows

(d) Internally restrained bellows: internal tripod with integral ball and socket

Figure 20. – Four examples of bellows joints.

bellows to absorb only certain motions (e.g., a gimbal ring linkage allows a bellows to receive only angulation). These end-point motions and the number of motion cycles that a bellows joint is designed to absorb are accurately determined in the line-assembly design phase.

Unfortunately, a bellows must absorb vibratory motions that are either flow induced by the internal fluid motion or mechanically induced by motion of the engine or its components. These bellows motions are not related to those described above; they usually occur as motions of the convoluted section in a variety of modes with the ends of the bellows fixed. The magnitudes of these motions are difficult to predict, and the frequencies are sometimes several thousand cycles per second. Motions at these frequencies that produce amplitudes and attendant stresses above the fatigue limit of the bellows material can cause failure in only a few seonds.

The pressure separating load on a bellows in a duct system is developed, as if the bellows were a pressure vessel with end caps, by the duct pressure reacting against a projected area across the duct bore (e.g., an elbow, an injector face, or any other reaction point in the system). Bellows have low spring rates and thus cannot withstand very high pressure separating loads without changing length; therefore, restraint mechanisms to prevent this change must be used to enable the bellows to perform its function in a duct assembly.

Information on axial-restraint design and stress analysis is provided in references 70 through 74. Information on the use of bellows in general design is presented in references 75 through 81. Basic considerations involved in the successful design of a bellows joint are presented in the sections that follow.

2.2.1 Bellows

Bellows have been used successfully in vehicle and rocket engine ducting in a wide range of sizes under a variety of operating conditions. However, success is dependent on careful design, as bellows generally operate in the plastic range of the bellows material. This condition is necessary so that the wall thicknesses can be reduced sufficiently to produce acceptable end-restraint reaction loads with the bellows in a deflected position and to reduce weight.

Formed, straight-side-wall bellows are the predominant type used in aerospace ducting. Welded-disk bellows, the other major type, are not used in any aerospace ducting application; their use is limited to rotary-shaft face-seal applications, where long strokes in relatively short lengths are required. The crown and root weld locations create a blind root notch that increases stress. Minor flaws have a tendency to propagate rapidly unless low-magnitude stresses are maintained.

2.2.1.1 PRESSURE CAPABILITY

A bellows must cope with several kinds of operational pressure:

- Normal operating pressure
- Surge pressure
- Proof pressure [(normal operating pressure + surge pressure) X safety factor]
- Burst pressure (proof pressure X safety factor).

These pressures produce hoop stresses in the bellows and bulging stresses that tend to bulge the convolution sidewalls. The bellows must be stable and resist column buckling under the proof-pressure application. While subject to the burst pressure, the bellows is permitted to deform and take permanent set but not to develop leaks.

The calculated operational stresses are startlingly high, exceeding the yield strengths of the materials, but in reality are only index stresses and must be treated as such. The limiting bulging stresses and allowable motion stresses (bending stresses) of frequently used bellows materials are given in table V. Cycles are full stress reversals at a frequency of 30 to 60 per minute. Stresses are calculated with the equations presented in reference 16 (p. 72); different types of motion (axial, lateral, angular) can be converted into equivalent axial motion with equations given therein (p. 73).

Table V. – Limiting Bulging Stresses and Allowable Motion Stresses of Frequently Used Bellows Materials

Material	Material Condition	Limiting bulging stress*, psi		Allowable motion stress, psi	
		$t \leq 0.012$ in.	$t > 0.012$ in.	1000 cycle life	10 000 cycle life
321, 347 CRES	Cold worked as formed; R_c 10 to 40	140 000	120 000	120 000	74 000
A-286	Heat treated; R_c 29 to 40	200 000	160 000	200 000	120 000
Inconel 718	Heat treated; R_c 38 to 45	200 000	160 000	210 000	135 000
Inconel X-750	Heat treated; R_c 30 to 37	200 000	160 000	165 000	105 000
Hastelloy C	Cold worked as formed; R_c 10 to 40	150 000	130 000	185 000	105 000

*Reduction in material properties for thicker sheet is due to poorer surface finish and greater variation in thickness than in thinner sheets.
R_c = hardness on Rockwell C scale

In determining additive bending stresses in the convolution crowns and roots caused by simultaneous application of pressure and deflection, where the maximum stress is that which could be applied if only one of the effects were present, the following expression is used:

$$\left(\frac{\sigma_b}{\sigma_{b,\,max}}\right)^2 + \left(\frac{\sigma_B}{\sigma_{B,\,max}}\right)^2 \leqslant 1 \qquad (1)$$

where

σ_b = bending stress (motion stress)

σ_B = bulging stress

The major problems with bellows used in ducting on large liquid-propellant rocket engines have been the following:

- Fatigue failures (due to mechanical and flow vibration and possibly contributed to by handling abuse)
- Bucking stability (internal and external)
- Corrosion
- Manufacturing difficulties
- Handling damage (sec. 2.1.5.7)

Other causes of bellows failures that have occurred less frequently are implosions in LOX or high-temperature service, and bursting due to pressure surge.

2.2.1.2 FATIGUE LIFE

The end-point motions of a flexible-duct assembly impose the largest deflections on the bellows joints within the duct. These motions are predictable in the design phase, as is the anticipated duty cycle. With the use of this information and the S-N (alternating stress vs number of cycles) properties for the particular materials involved, the bellows then can be designed for an adequate low-cycle fatigue life while operating under very high stresses. Since aerospace design emphasizes light weight, bellows are characterized by thin walls. Thin walls reduce the weight of the bellows itself and also reduce spring rates, so that the reaction loads on attaching members are low and their weight thus is kept to a minimum. This thin-wall feature dictates that the bellows operate under stresses near or in the plastic range, and fatigue life in the low-cycle range is a result. Reference 82 presents low-cycle fatigue data for bellows made from 321 CRES and nickel-base alloys.

Other motions imparted to the bellows come from vibrations that are induced mechanically or by the fluid flow. The bellows must be designed such that the stresses incurred are kept below the endurance limit of the material (i.e., no reduction in fatigue life is caused by vibration).

Methods for predicting flow-induced vibration inputs are presented in references 83 through 90; these techniques utilized in bellows design can enable the designer to develop a bellows completely free from vibration or at least free from unacceptable levels of vibration.

Prediction of mechanical vibration inputs is more difficult because of the dynamic complexity of the structure to which the flexible duct is attached. One method is to extrapolate known vibration inputs from similar installations on other engines and compare them with calculable mechanical resonant frequencies of the duct design; a more accurate but costlier method is to build prototype hardware and test it on development engines while the vibration environment is measured. If a bellows operates in a vibration environment containing excitation frequencies matching any of the natural structural frequencies of the bellows, its fatigue life can be exceeded within a short operational time period. The most frequent bellows failure mode is the development of a fatigue crack that permits leakage through the convolution. The cracks usually are circumferentially located in the crowns or roots of the convolutions, since these are the areas of maximum bending stress. Analysis of past fatigue failures indicates they have been caused by flow-induced vibration (sec. 2.2.1.2.2), mechanical vibration (sec. 2.2.1.2.3), or a combination of the two (sometimes aggravated by overdeflection handling abuse). Successful bellows are those designed with adequate fatigue margins that allow for all of these influences.

2.2.1.2.1 Vibration Susceptibility

Structural integrity must be maintained during and after the operational periods when many types of dynamic loads may be applied to a bellows. The types of loads include those generated by the enclosed fluids (e.g., pressure surging [Pogo, water hammer, pump oscillations] and flow-induced vibration) and those imposed by mechanical environments (e.g., sinusoidal vibration, random vibration, shock, noise, and steady-state acceleration).

There are three primary modes of bellows vibration (fig. 21): (1) the axial or accordion mode, (2) the lateral mode, and (3) the mode in which the individual convolutions rotate back and forth at the inner diameter while pivoting about the outer diameter. The individual-convolution mode (fig. 21(c)) is especially noticeable in flow-induced vibration of braided hoses, where the convolution outer diameter is restrained by the braid friction and thus provides a fixed pivot point for the convolution being excited by flow forces.

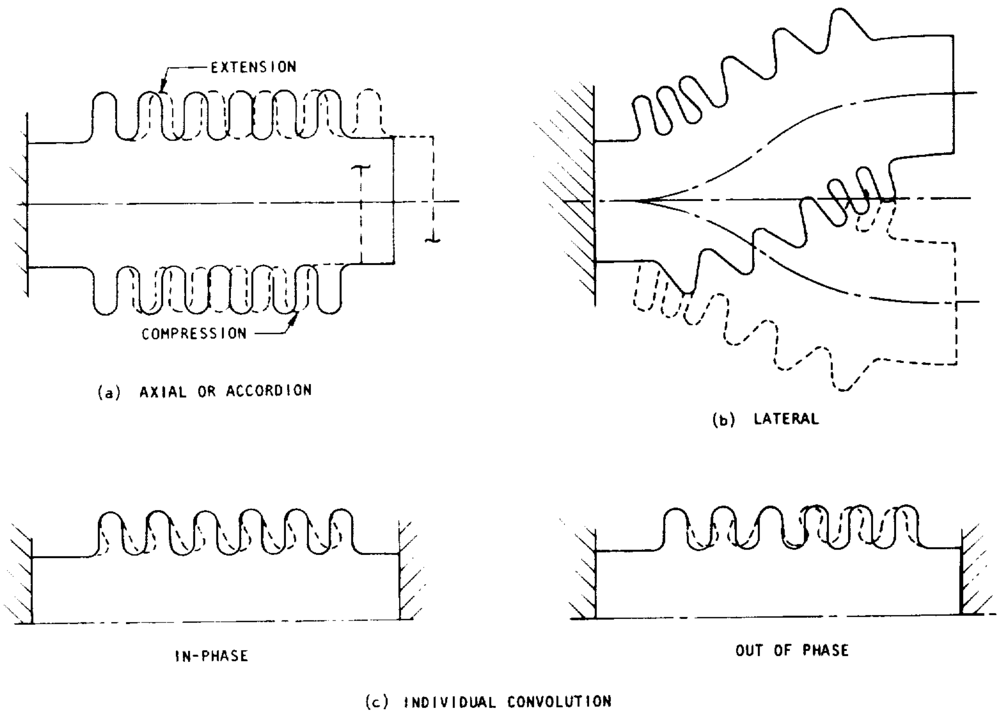


Figure 21. – Three primary modes of bellows vibration.

For a given bellows design, the frequencies of these modes can be predicted with reasonable accuracy (refs. 83 through 94). The difficult problems are determining (1) the frequencies and amplitudes of flow or mechanical vibratory forces that may excite these modes and (2) the possible response characteristics of the bellows to these forces.

2.2.1.2.2 Flow-Induced Vibration

Flow-induced vibration of flexible-line bellows with resultant fatigue failure is a frequent problem in lines containing fluids flowing at high velocities. In bellows, an internal flow liner (sec. 2.2.4) has always resolved the problem by preventing impingement of the flow stream on the convolutions; however, the installation of the liner usually presents complications in design and fabrication and makes the bellows more difficult to clean. The liners themselves have failed, apparently as a result of vibration. Flow liners appear to be the best "quick fix" or empirical solution to flow-induced vibration, but not necessarily the best solution. References 95 and 96 give examples of quick-fix solutions to some Saturn ducting problems.

Not much is known about the actual physical phenomena involved in flow-induced vibration; however, the recent experience with flexible-line failures on the Saturn AS-502 vehicle instigated an intensive investigation that greatly advanced the state of the art in understanding and preventing vibration failure (refs. 28 and 97). The Saturn AS-502 failure investigation culminated in an analytical method (ref. 90) for determining whether a flexible line is susceptible to flow-induced vibration. The analysis evolved from the results of a test program utilizing more than 50 different flexible line and duct specimens (refs. 64 and 98).

The analysis determines the dynamic response and fluid excitation required to cause fatigue failures associated with the individual-convolution vibration mode. The essential dynamic consideration is that the response characteristics for the individual-convolution mode are the result of coupling of the structural dynamics of the convolutions and the dynamics of the fluid within the convolutions. The excitation due to fluid flow within the flexible line is described as being capable of exciting a band of response frequencies where the boundaries of the excitation frequencies are proportional to velocity. The low-frequency excitations generally are associated with vortex shedding within the turbulent boundary layer. The high-frequency (and potentially much stronger) excitation is associated with a convection frequency in the turbulent boundary layer. The convection frequency is defined as the characteristic frequency associated with the fluid velocity past equally spaced wall irregularities (the convolutions).

Acoustic loading can increase the number of degrees of freedom of a bellows system by coupling the bellows to the acoustic modes. When this type of coupling occurs, very significant pressure fluctuations can be propagated throughout the fluid system. These acoustic resonances, in addition to producing noise, cause a significant increase in the flow-induced stress levels. This effect apparently is the result of the acoustic-resonance flow and pressure fluctuations coupling with the vortex shedding process to produce a force amplification. This phenomenon is discussed in detail in references 85 through 89.

If a resonance can exist, the analysis provides a technique for estimating the energy levels associated with the resonance; the energy available can then be translated to a stress level in the part. When the expected stress level is known, a quantitative evaluation then can be made to determine if the stress level exceeds the alternating stress that the part is capable of withstanding without high-cycle fatigue failure.

The detailed analytical methods for determining flow-induced vibration excitation and response dynamics are presented in reference 90. Useful response information also appears in references 91 through 94. These analytical methods provide for determining, with a high confidence level, the flow-induced vibration susceptibility of a given flexible-line design for a given set of operating conditions while still in the design phase; these techniques thus eliminate the need for expensive cut-and-try flow testing of prototype hardware.

Before the sudden surge in activity on flow-induced vibration of bellows brought about by the failure of flexible lines on the Saturn AS-502 vehicle, it was commonly believed that the multi-ply bellows, because of the friction damping between plies, was a solution to the flutter problem. Many experiences (e.g., those involving the bellows on the Thor pump discharge line, and bellows on the helium inlet duct on the F-1 heat exchanger) proved that multi-ply in itself was not the solution. The dynamics technology after the Saturn AS-502 flexible-line failures proved this conclusively. The multi-ply does provide some damping over the single ply (refs. 84 through 88), but if the natural structural frequency of the convolution shape involved coincides with exciting frequencies, failure is likely to occur.

2.2.1.2.3 Mechanically Induced Vibration

The vibration-response characteristics for a given mechanical excitation input are calculated in the same manner as those for flow-induced vibration. It is relatively simple to predict flow-induced excitation inputs, but it is extremely difficult to predict the mechanical-excitation amplitudes and frequencies of the components or system in which the bellows will be installed. Experience in the analysis and experimental correlation of similar systems is an invaluable asset in this endeavor, and the best analysis possible is made prior to design release.

The excitation frequencies and acceleration forces sometimes are not accurately identified until accelerometer measurements are made on the actual hardware in engine testing during development. An analysis of the effect of these inputs on the flexible ducting and bellows is then made, and design changes are incorporated, if necessary, before committing the design to production. In some applications, only frequencies of potential exciters are known (e.g., functions of known pump speeds). In this case, bellows are deliberately designed to keep natural-response frequencies from matching excitation frequencies.

Double-bellows-spoolpiece arrangements (fig. 22) with an undamped or unsupported span are susceptible to vibration of the long unsupported mass from both flow-induced and mechanical excitation.

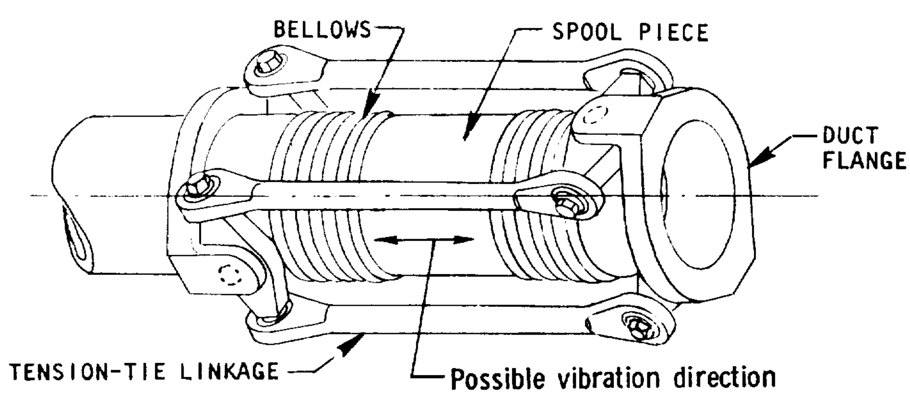


Figure 22. – Example of double-bellows-spoolpiece configuration susceptible to vibration.

The initial design of the discharge duct on the pump for the F-1 engine incorporated this configuration; fatigue cracks attributed to vibration occurred in the bellows. The condition was intensified by stress raisers in the form of wrinkles found on the inside radius of the convolutions. These wrinkles were attributed to an overly small ratio of bellows convolution radius to thickness. As noted, these ducts were changed to hard lines of aluminum. This vibration mode also was observed in the inlet ducts on the J-2S pump. Although vibration was present, the amplitudes were not large enough to cause failure.

2.2.1.3 BUCKLING STABILITY

Buckling of the bellows can result from excessive internal or external pressure. Internal-pressure buckling takes the form of column instability, whereas external-pressure buckling appears as crushed convolutions.

If the bellows is designed properly initially, buckling can occur only when operating conditions beyond the original specification requirements are imposed on the part. For example, maximum surge pressure may occur in service with the bellows in its most elongated position, a condition for which the part may not have been designed; or the rigidity of the end restraints, a very important parameter in buckling, may be less than was assumed in the design calculations.

Buckling stability analysis for internal pressurization is well founded (refs. 99 through 102). The calculation of critical buckling pressure for an externally pressurized bellows is similar to that for a thin-wall tube. An equation for the bellows analysis can be found in reference 2 (p. 377). If all the operational conditions are well considered when the design requirements are established, sound application of the analytical methods presented in the cited references can produce a trouble-free product.

2.2.1.4 CORROSION RESISTANCE

A commonly used material for aerospace bellows is 321 CRES. It is readily available, has good forming properties, and has high ultimate strength in comparison with yield strength. However, it has been demonstrated repeatedly in the Atlas, Thor, and Saturn engine programs that while the material is, as its name implies, corrosion resistant, it is not corrosion proof. Most of the classical corrosion modes such as stress corrosion, pitting corrosion, and intergranular corrosion have occurred (ref. 103).

The high percentage of iron in 321 CRES makes it susceptible to corrosion. Bellows have residual stresses as a result of cold working; being thin walled, bellows thus are particularly vulnerable to eventual leakage caused by stress corrosion. Chlorides, deposited on the bellows surface by everything from contaminated water to perspiration from a sweaty palm, typically have been the corrosion promoters. Chlorides used in cleaning agents can be entrapped in geometric crevices such as close-pitch convolutions. Cryogenic fluids cause moisture to condense and freeze on the outside surfaces of engine ducting and then act as an electrolyte with chlorides or other contamination deposits to promote corrosion. This problem is dealt with on existing hardware in service by periodic leak checks and inspections, with replacement of unfit hardware, and by continuous efforts to maintain a moisture-free controlled environment.

The storable (noncryogenic) propellants used in the Titan II, Apollo Command Module, and LEM ascent and descent stages do not create copious amounts of condensate, and the hardware has not been subject to corrosion. Many storable propellants, however, are very corrosive, and selection of compatible materials for service in contact with these fluids is an important design problem (refs. 42 through 45).

2.2.1.5 MANUFACTURING

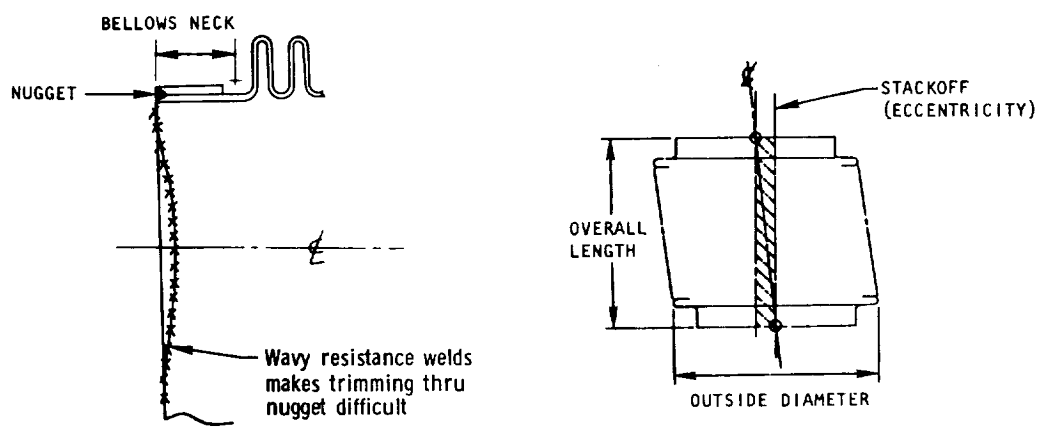
The main problems in the manufacture of formed, straight-side-wall, convoluted bellows have involved material thinning, proper heat-treatment control, proper welding, and convolution stackoff.

Thinning of the convolution crowns during the forming process is a common occurrence with roll-formed bellows. With this type of forming process, thinning 10 percent below the nominal wall thickness is attained. Hydraulic forming processes have less trouble with thinning, and thinning 5 percent below nominal wall thickness is attainable. The consequence of thinning is that more deflection takes place in the thinned section because stiffness is reduced, and therefore high stresses and early fatigue failure can occur. Thinning can be controlled by dimensionally examining sample pieces in a production run and by subsequently adjusting initial tube thickness, tooling, or production techniques if variation from the norm appears.

Proper control of heat treatment requires constant quality-assurance surveillance, because (1) bellows performance is degraded by improper heat treatment and (2) bellows usually are heat treated in lots; improper heat treatment can then be costly in requiring reprocessing or replacing the entire lot. One typical problem has involved maintaining the proper atmosphere in the furnace or retort during the heating process. Circulation of the atmosphere must be maintained to ensure coverage of all bellows surfaces; the spaces between convolutions are particularly restrictive to good flow coverage. Result of improper heat treatment can be lower-than-design strength and possible pressure or fatigue failures. Surface effects such as oxidation and "orange peel" also can take place and reduce fatigue life because of the notch stress raisers created.

Welding problems stem largely from lapses in quality control and workmanship. Ideally, all welds are inspected radiographically. However, some welds such as resistance welds and lap fusion welds inherently have the gap between the materials being joined show up on the X-ray film as cracks, which are difficult to interpret. These welds can be controlled by periodic sectioning and examination of production parts and control of the welding processes and operators.

A typical weld problem involves the resistance welding of plies at the neck of the bellows (fig. 23(a)). Prior to fusion butt welding in the next assembly, this weld is trimmed through the nugget. If the resistance weld is not maintained within the same dimension from the end of the bellows (i.e., if the weld is wavy as it progresses around the circumference), trimming with a normal cut will not always pass directly through the nugget center and may even cut through the weld completely in localized sectors. This condition makes it very difficult to make the fusion weld to the next assembly, and considerable grinding out and rewelding are required to achieve an acceptable radiograph. The problem can be resolved by either controlling the distance of the weld from the end of the bellows or widening the weld nugget.



(a) BELLOWS-NECK RESISTANCE WELD (b) CONVOLUTION STACKOFF

Figure 23. – Two examples of problems in bellows fabrication.

Convolution stackoff is the eccentricity of the convolutions of the bellows with respect to each other. Since the condition is inherent in the tooling of the fabrication method, all of the eccentricity is accumulated in the same direction (fig. 23(b)). This accumulation can be controlled by rotating the tube after the forming of each convolution. Such control is important in fabrication, because stackoff results in difficulty in aligning the bellows for welding into its next assembly. Shear forces must be applied to the part to align it in a straight position, and once the restraining forces are relieved, the bellows will assume its natural stackoff position. Stackoff is more noticeable in large-diameter (> 6 in.) bellows, but can exist in any size.

Careful manufacture can reduce the amount of stackoff, but it must always be considered. Stackoff becomes more noticeable and more significant when the bellows is in the compressed mode and when the bellows has a small ratio of length to diameter. In this respect, the rapid increase in lateral stiffness as this ratio becomes smaller contributes to the significance of stackoff.

2.2.1.5.1 Interply Lubricants

The tubes forming the multi-ply construction are slipped together concentrically during fabrication of the bellows; a very tight fit is required and sometimes a lubricant is used. Because of the tight fit, it is difficult to remove the lubricant after the tubes are engaged. Thus, the lubricant must be chemically compatible with the service fluid, particularly with LOX if the part is to be used in LOX service; it must also be nonvolatile at operating temperatures to prevent interply pressurization.

Failures have occurred in multi-ply bellows used in LOX and in high-temperature ducts when contaminants trapped between the plies vaporized during operation (or reacted violently when they came in contact with LOX that entered through a crack) and caused sufficient pressure to burst the bellows (ref. 104). The word "implosion" is used to describe the failure mode, because the rapid pressure buildup causes the inner ply to burst into the flow stream.

Solid dry-film lubricant (e.g., molybdenum disulfide) has been utilized successfuly as an assembly aid; since it is a solid, it creates no volume-expansion problem when heated.

A successful method for removing lubricant or moisture is to bake the bellows after forming and before the plies at each end of the assembly are circumferentially welded together. If a particular bellows-forming process necessitates welding the tubes circumferentially prior to forming to prevent ply slippage, vent holes are located adjacent to the weld in each of the plies; the vent holes permit outflow of trapped vapors during the baking process after forming. A second circumferential weld is then made inboard of the vent holes, and the bellows neck is trimmed to eliminate the hole areas. This procedure prevents contamination from enterring between the laminations once the part is in service.

Because of the failures that have resulted from interply lubricants, the practice of using lubricants has been largely discarded by bellows suppliers. The ply tubes are nested concentrically without lubricant, and the additional clearance required between tubes is taken up by hydraulic sizing prior to forming the convolutions.

2.2.1.5.2 Weld Necks

The success or failure of a bellows joint often depends on the detail design involved in the integration of the bellows-to-duct interface. In preferred configurations, the weld neck inner diameter (ID) is equal to the bellows ID. Since the bellows wall thickness is always less than that of the adjoining duct, the transition piece must provide the proper thickness ratio for a sound weld joint. Doublers are often used to build up weld-neck thickness for a more favorable ratio of bellows-to-duct weld thickness. However, if a doubler is permitted to touch the end convolution, the stress raiser thus created can promote early fatigue failure when the bellows is deflected or subjected to vibration. Also, the weld attachment is kept away from the end convolution, because (1) the flexural stress at the convolute root dissipates rapidly with distance from the convolute and (2) short necks make welding difficult.

The spring-rate loads of the bellows transmitted to the duct at the weld neck are factors in neck design. Failure of the weld joint or the neck can occur if the stress applied at the bellows-to-duct interface exceeds the design strength.

Although the geometry of the outside-neck attachment may have appeal, designs that require this configuration are evaluated carefully, because fabrication is difficult. The outside-neck attachment always increases cost of manufacture and may downgrade the bellows performance, as certain convolute configurations may be inhibited.

2.2.2 Bellows Restraint

2.2.2.1 MECHANICAL LINKAGES

The most commonly used restraint design is the gimbal-ring joint (or universal); other concepts include the hinged joint, ball joint, and braided sheath. Reference 16 (pp. 76-78) presents a broad spectrum of designs.

The gimbal-ring joint is the "workhorse" of bellows restraints. It can be designed to withstand extremely high pressures and temperatures, is capable of angular deflection in all planes, and will withstand imposed torsional loads. Stops to prevent excessive bellows angulation are readily adaptable to this design. The load-carrying pivot pins may be designed for either single or double-shear support, the choice depending on the separating load. The same comments apply to the hinged joint, which provides angulation in one plane only.

A subtlety of gimballed bellows, sometimes overlooked even by experienced duct designers, is the fact that a gimbal (or universal) joint, when deflected about both of its axes simultaneously, produces a torsional deflection (refs. 105 and 106). Motion is not transmitted with a constant angular-velocity ratio unless the intersection of driving and driven components lies in a homokinetic plane (ref. 105). Gimbal-ring linkages do not have this type of geometry; therefore, they induce a torsional deflection. Because the bellows normally is very rigid in torsion, high torsional stresses are produced.

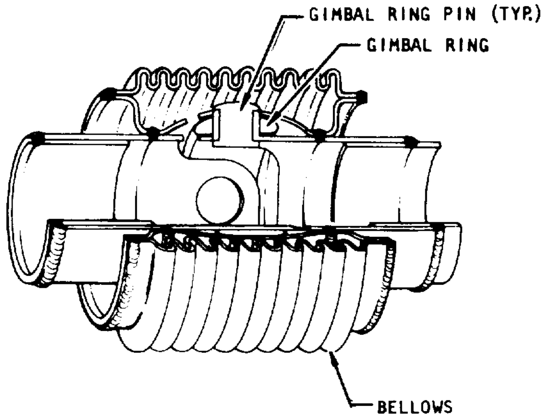
One of the most popular and inexpensive methods for an external, tension-tie restraint is the use of a braided wire sheath, as in the case of a flexible hose. This type of joint can absorb both angular and shear deflections, and acts as a vibration damper for the bellows inside. The braid restricts angular deflection under high pressure by offcenter pivoting, which results in increased bending moments.

Friction between the braid and bellows can be minimized by an adapter, which provides clearance at the end convolutions, and by the application of solid dry-film lubricant on the outer surface of the bellows and inner surface of the braid. Excessive friction and eventually possible galling of the journal-type pin bearings on the linkages is a problem that also can be alleviated with solid dry-film lubricants (refs. 107 and 108). LOX-compatible lubricants must be used between bellows plies and on the bearings of restraint linkages utilized in LOX systems. Data on LOX compatibility of lubricants are utilized in making a selection; if data are not available, as on a new product, LOX impact tests are performed. Reference 107 provides additional information on the use of lubricants for linkage bearings.

Once a gimbal-ring type of bellows restraint has been selected, the question of internal versus external mounting (fig. 24) must be resolved. In general, the selection of internal or external gimbal is left to the discretion of the designer after design tradeoff studies for his particular application have been made. Each type has its advantages and disadvantages. The internal gimbal is more difficult to clean, because all of the mechanisms comes in contact with the flowing fluid and therefore requires cleaning. Both gimbals can be "sleeved" to reduce flow losses; however, the conventional internal-gimbal-ring design has a higher loss coefficient than the external-ring, sleeved-bellows design.

If the diameter of an internal gimbal ring is equal to or greater than the duct inside diameter, a larger diameter bellows is required to encompass the linkage. The larger bellows causes a greater separating load and consequently an increase in linkage weight to carry this load. A comparision of the weights of internal and external gimbal designs for a given diameter and pressure cannot be generalized and must be evaluated for each individual design case.

The external-ring design affords some handling damage protection to the bellows, whereas the internal ring design leaves the bellows exposed. Double versus single shear in the clevis pin of the gimbal linkage is a function of the load that the joint can carry, and either type is adaptable to both the internal and external gimbals. Experience on the Centaur stage has



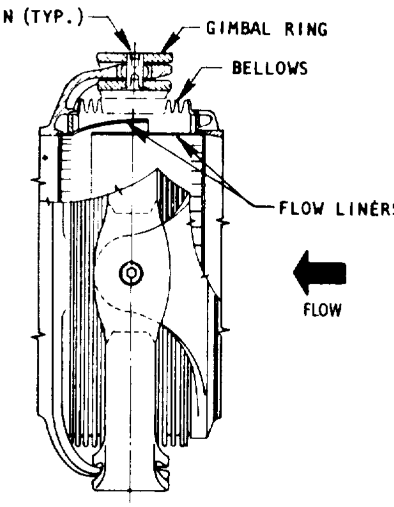


Figure 24. – Internal and external mounting of gimbal rings with double-shear clevis pins.

shown that the double-shear design is highly desirable. A single-shear design was used for the propellant feedlines in the early Centaur development, but was abandoned in favor of the double-shear design. The double-shear design proved to be much stronger and improved the reliability of the joint. The mechanism of failure in a single-shear design is popping or stripping of the heads from the pins restraining the joint under load.

In the F-1 engine, in ducts equipped with external-gimbal-ring bellows joints, the gimbal ring came in contact with the bellows' outer diameter in an overdeflected condition and caused damage. The ducts were small (1.5 in. diam.) and therefore easily overdeflected during shipment, handling, and installation. Normally, protective covers are installed to prevent damage and overdeflection of the bellows joint, but in this instance the covers were not installed. To solve the overdeflection problem, the gimbals were redesigned so that the ring would bottom out on the rigid tube structure of the duct instead of on the bellows.

The internal-tie link joint is an economical version of the gimbal joint for small or medium pressure-separating loads, and weighs less than the equivalent gimbal. It has the disadvantage of high loss factors due to the frontal-area restriction to flow. This joint is not suitable for use in systems with high-velocity fluids, but frequently is used in the low-pressure, turbine-gas duct systems of rocket engines. Link joints also have low torsional strength.

A peculiarity of the chain-link joint (fig. 25) is that the pivot point of the linkage is not fixed but depends on the plane in which the joint is angulated. Since the angular deflection is evenly distributed over the live length of the bellows only when that length is centered about the pivot point, this change in pivot-point location with plane of angulation will reduce fatigue life.

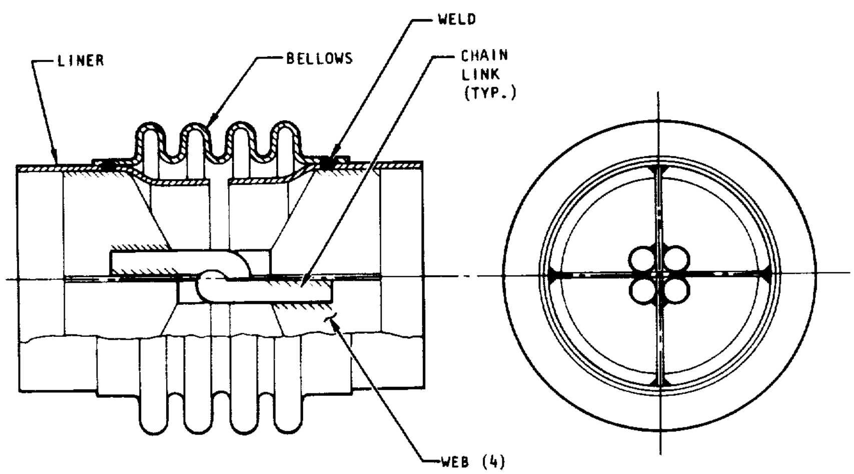


Figure 25. – Chain-link restraint joint with internal tie.

A case in point is the chain-link joint design used in the duct from the fuel turbine to oxidizer turbine on the J-2 engine. This joint developed a history of being unable to pass consistently the production-lot comparison fatigue tests. It was found that the manufacturers' tooling was not locating the chain link accurately on the center of the bellows live length. The links of the chain were 0.625-in. in diameter, and the bellows live length was a relatively short 2.4 in.; consequently a slight shift in live length had a great effect on bellows deflection and fatigue life. Results of flexural endurance tests showed a 20-percent reduction in fatigue life (800 cycles vs. a normal 1000 cycles) with a 0.100-in. axial shift in pivot point.

Internal linkages are a requirement for nuclear engines because external inkages act as heat sinks for radiation. The linkages are cooled by liquid-hydrogen propellant.

A common problem with bellows restraints, both gimbal and hinge types, is that the bellows pressure separating load produces a reaction in highly localized areas of the duct. Loads concentrated in a point on a member are distributed across the cross section of the member, as described by a 30° to 45° half-angle. When the joint is adjacent to a flange, duct length is insufficient to allow the concentrated loads to redistribute circumferentially into the walls of the duct. Consequently, localized loads applied to the bolted flange can overload the bolts in the load area or cause deflection in the flange, possibly allowing static-seal leakage.

The ball-bearing ball-joint restraint (fig. 26) was used in the inlet line to the F-1 LOX pump on the engine test stand. The line has an inside diameter of 16.75 in., and a proof pressure of 400 psig. Because of the large pressure separating load, the duct overall length limitations and gimbal angular-deflection requirements dictated that the bellows joint utilizing this restraint be located immediately upstream of the pump inlet flange. An uneven, circumferential loading of the pump flange could not be tolerated structurally by the pump, and the ball joint offered a design capable of providing gimbal-type restraint and motion while applying a uniform circumferential load to the pump flange. The problem was resolved on the flight ducts by gusseting the flange to the duct wall.

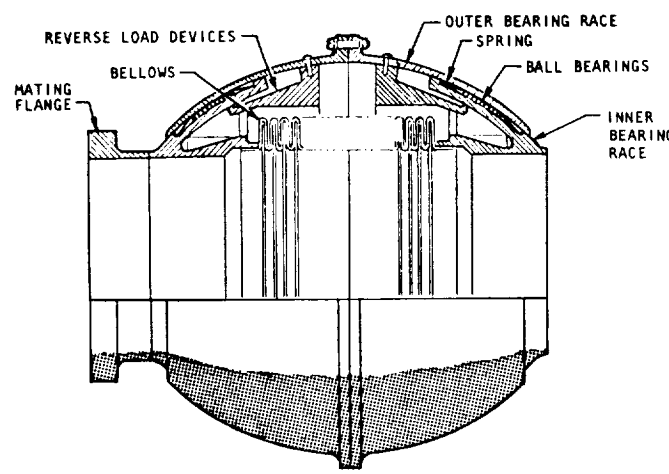


Figure 26. – Ball-bearing ball-joint restraint used on F-1 engine oxidizer pump for engine test.

The SSME flexible joints incorporate internal tripod and external gimbal-ring linkages. The internal type (fig. 27; cross-sectional view presented in fig. 17(d)) is used in the low-pressure pump discharge ducts where the pressure loss can be tolerated and the overall joint envelope must be kept as small as possible. The external type (fig. 28) is used on small-diameter (2.0 to 2.7 in.) high-pressure ducts where the pressure losses associated with internal ties are not acceptable. The extremely high pressures and gimbal angles (up to ± 13.5°) of the externally tied joints required the use of bellows' live lengths that were unstable in column buckling. This difficulty necessitated the development of a novel linkage (fig. 28 and ref. 109) that

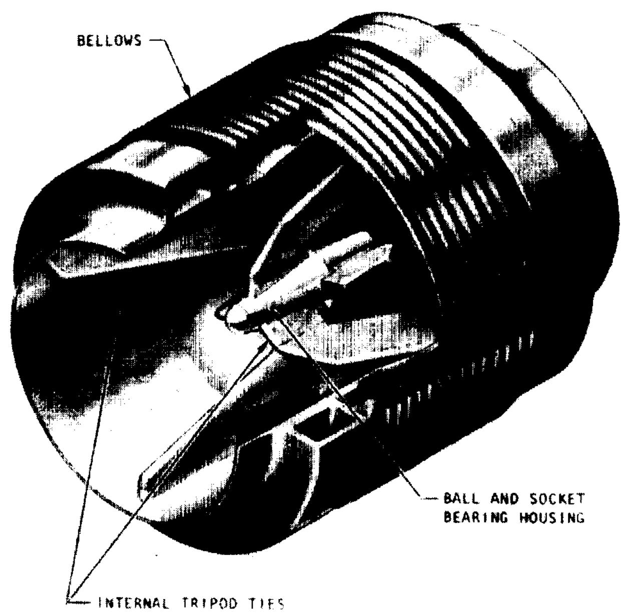


Figure 27. – Internally tied tripod flex joint used on discharge duct of SSME low-pressure pump.

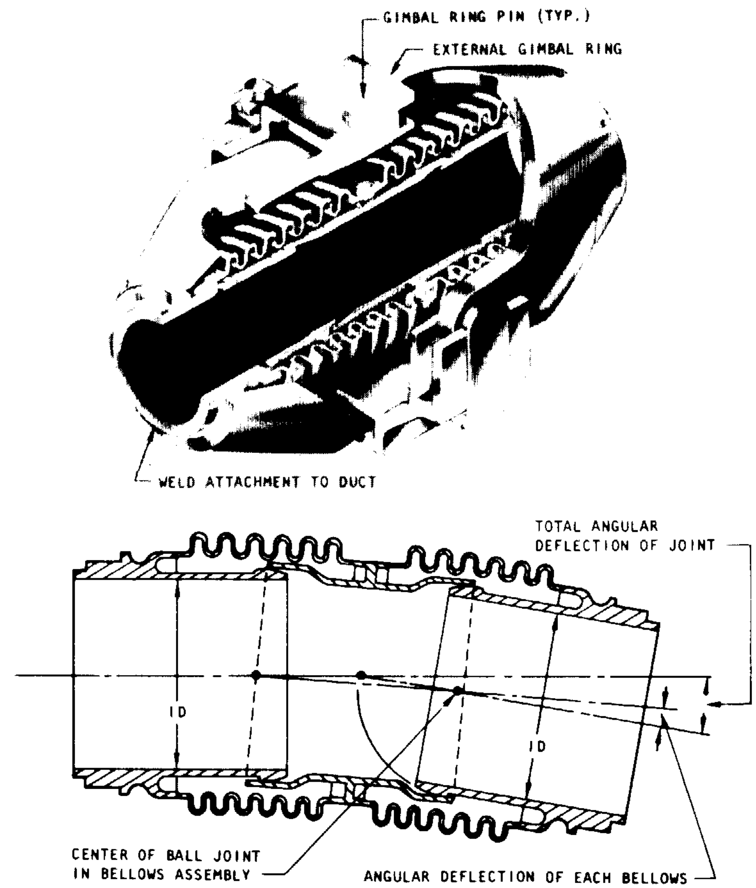


Figure 28. – Externally tied gimbal ring flex joint used on SSME high-pressure lines.

provided lateral support at the mid-span of the bellows live length and thereby eliminated the buckling-stability problem. The geometry of the design in figure 28 ensures that the angular deflection of the joint assembly is distributed equally between the two bellows.

The "Gimbar"* linkage-restraint type of flex joint (fig. 29) is used in the fuel and oxidizer fill and drain ducts of the Shuttle Orbiter (table I). This configuration offers a lightweight design (lighter than an external or internal gimbal ring) for large-diameter ducts carrying low-velocity fluids where the pressure loss associated with the structure protruding into the flow stream can be tolerated. This design provides torsional load-carrying ability through the linkage; the tripod or chain-link configurations do not.

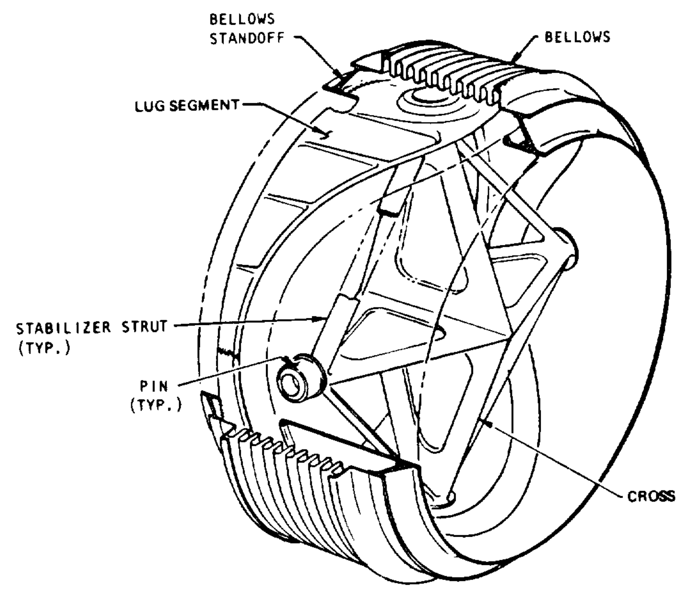


Figure 29. – Typical "Gimbar" restraint assembly used on fill and drain lines of Space Shuttle Orbiter.

The bending moment loads required to angulate flexible-joint assemblies can be a significant factor in the design of a duct assembly and the load reactions at the duct installation and attach points. A gimballing duct, for example, can add a considerable force requirement to the actuators that apply the moments to gimbal an engine.

*Coined word for gimbal ring with crossed bars for structural strength.

2.2.2.2 THRUST-COMPENSATING LINKAGES

The use of a thrust-compensating bellows to offset the thrust of a primary bellows in a duct system (fig. 30) can be considered as a form of a pressure-separating-load restraint mechanism (i.e., a thrust-compensating linkage). This type of bellows assembly allows for axial deflection in the same manner as an unrestrained bellows.

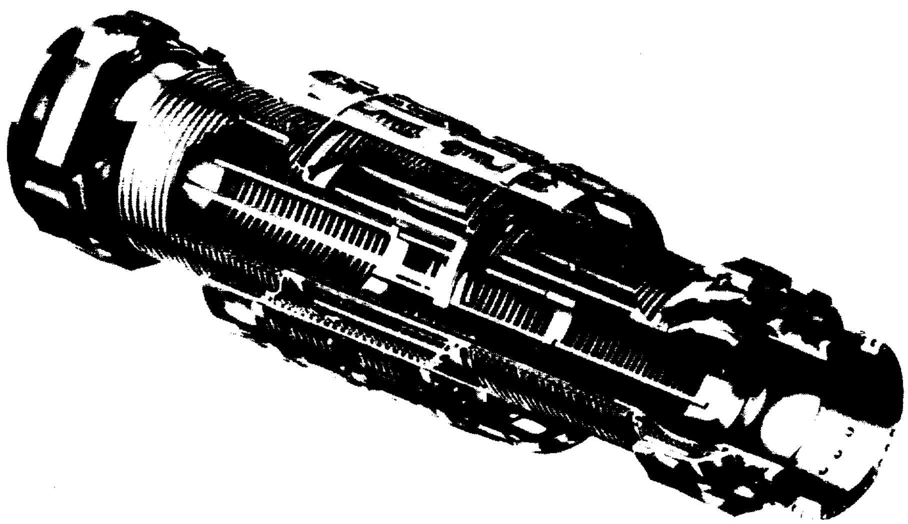


Figure 30. – Thrust-compensating linkage employing a thrust-compensating bellows (PVC joint).

The term "pressure-volume-compensating" (PVC) duct has come into use in describing this type of joint. This terminology is misleading, however; no pressure compensation is performed by the duct. A more nearly correct description is "thrust compensating," because it is the axial thrust caused by the pressure separating force that is balanced by the compensating bellows. It is axiomatic that, in order to compensate for thrust, the volume must also be compensated; so this portion of the PVC description is correct. The primary use of the compensating joint is to retain a tension-type system in areas where limited space exists and axial travel cannot be absorbed by angulation. By proper selection of the cross-sectional area of the balance bellows, the thrust force may be undercompensated, fully compensated, or overcompensated, so that a compressive, balancing, or tensile load is imposed on the installation restraints.

The large (± 50%) volumetric changes incurred in straight-run gimballing ducts correspondingly incur pressure perturbations that could be detrimental to engine operation. The thrust-compensating design eliminates these pressure variations. This design concept was used in all the ducts to the pump inlets of the five F-1 engines of the Saturn S-IC vehicle. As the engines respond to thrust vector control, these ducts are required to have an axial stroke of as much as ± 50 percent of the live length of the compensator bellows. Without the PVC feature, the volume change of the system could cause corresponding pressure pulses and variable loading on the pump inlets. Consistent engine performance depends on uniform supply of propellants to the pumps. The PVC design maintains a constant system volume under all engine gimballing conditions.

In the F-1 pump inlet duct, a bellows external to and concentric with the main line bellows offsets the separating force of the main bellows (fig. 31, a schematic of the joint shown in fig. 30). The annular chamber is vented to internal duct pressure, and axial-tension rods tie across the bellows, as indicated, alternately around the circumference. Telescoping flow sleeves can also be provided. This design permits wide-open flow area throughout its length and thereby adds very little additional pressure loss over a straight hard line.

The thrust-compensating design shown in figure 32 utilizes axial bellows of the same cross-sectional area; operating pressure is applied to the outside surfaces of both bellows. The internal volume of the bellows second from the left in the figure is vented to atmosphere. The outer shell over both center bellows acts as the linkage that restrains the forces separating the two. Flow direction is indicated by arrows. The joints on either end of the duct are of the internally tied, angulating type and complete the utility of the duct assembly. This externally pressurized configuration permits the use of axial bellows with lower spring rate than can be used with pressure applied internally to bellows, because the critical buckling pressure of a given bellows design is higher with external pressure than with internal.

The "finger compensator" design shown in figure 33 features a thrust-compensating chamber installed external to and concentric with the main duct bellows. The chamber is vented to internal duct pressure and, since the annular area AA is equal to the main duct bellows area A_1, an axial force equal and opposite to the pressure-separating force of the main duct bellows is created. The entire joint is prevented from separating axially by the overlapping fingers alternately tying across the joint from each free end. Sufficient clearance is provided between fingers to permit the required axial and angular motion. This design provides compactness, because the bellows overlap concentrically and in addition are not limited to axial stroke alone.

The design in figure 33 was used in the pump-discharge ducts of the early J-2 engines. In later engines, the design was changed to the external-tie-bar design shown in figure 34, not because of performance problems but because of lower cost. It is interesting to note that this classical design used for many years in the petrochemical industry is still practical for the more sophisticated aerospace industry.

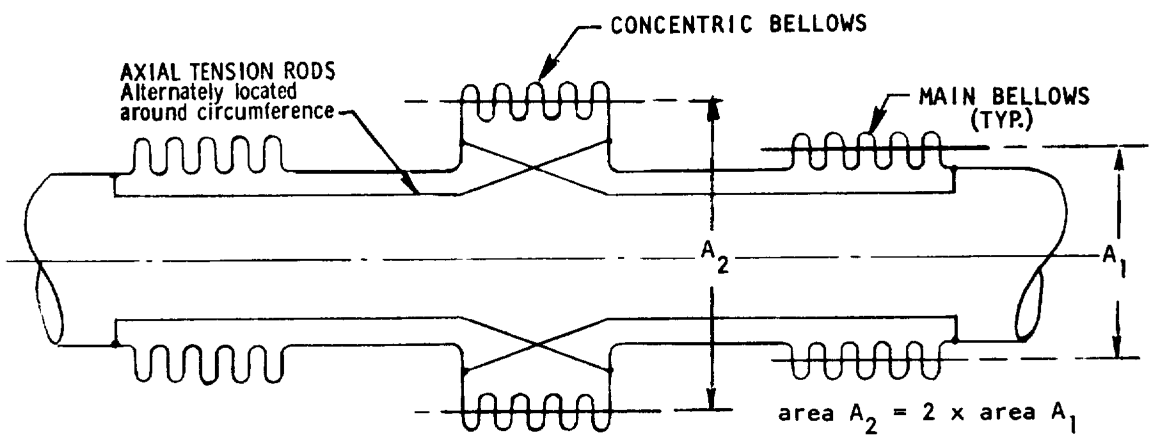


Figure 31. – Thrust-compensating linkage, internal pressure
(F-1 pump inlet line).

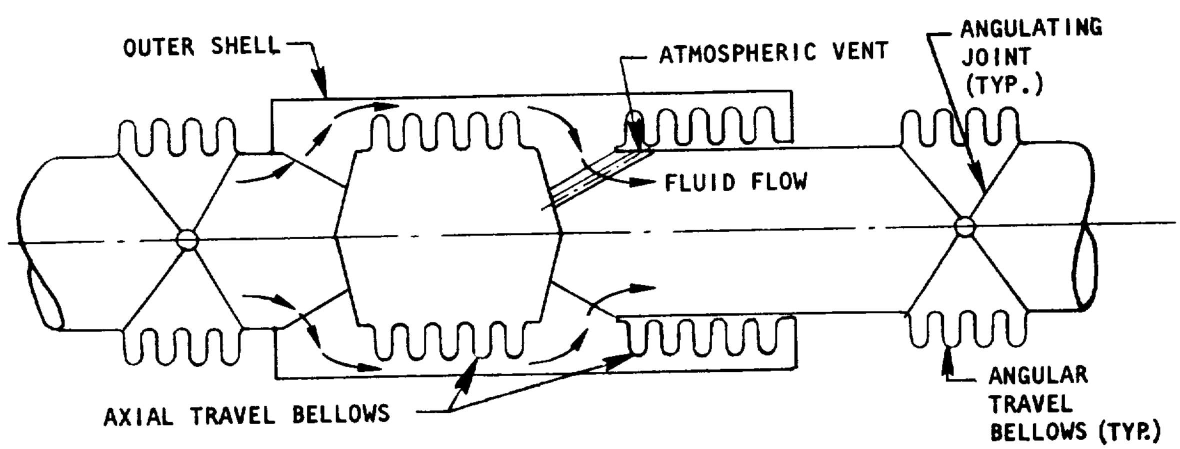


Figure 32. – Thrust-compensating linkage, external pressure.

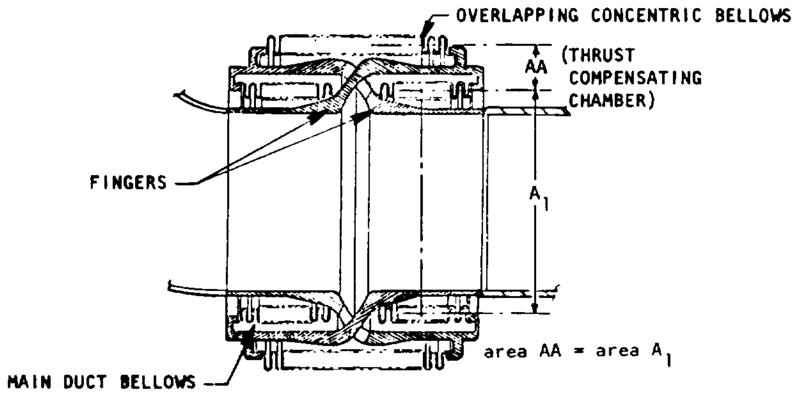


Figure 33. – Finger-compensator joint with thrust-compensating chamber.

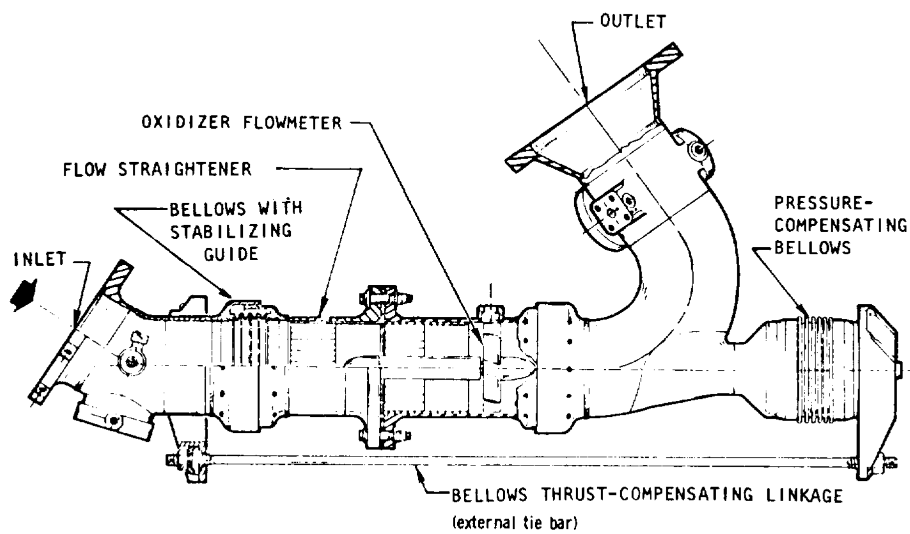


Figure 34. – Thrust-compensating linkage with external tie bar
(J-2 oxidizer high-pressure duct).

2.2.2.3 COMPRESSION SYSTEM

A compression system is one in which free bellows, without tension-tie linkages, are used to absorb the operational deflections in a duct system. The pressure separating and spring-rate forces of the bellows then must be reacted by the mating attachment structure of the engine or airframe; this reaction places the ducting in compression.

In ducting systems with compression-restrained bellows (e.g., low-pressure systems), the engine or vehicle support structure can be overloaded with moments (shear deflections in bellows) caused by eccentric pressure separating loads from the bellows. This problem can be minimized by aligning the thrust vector of the bellows with the reaction points of the attaching structure. If the operational deflections of the bellows are predictable and repeatable, the bellows can be installed in an opposite deflection position such that it moves toward a neutral (and lower stress) position while in operation. This type of installation is shown in figure 35.

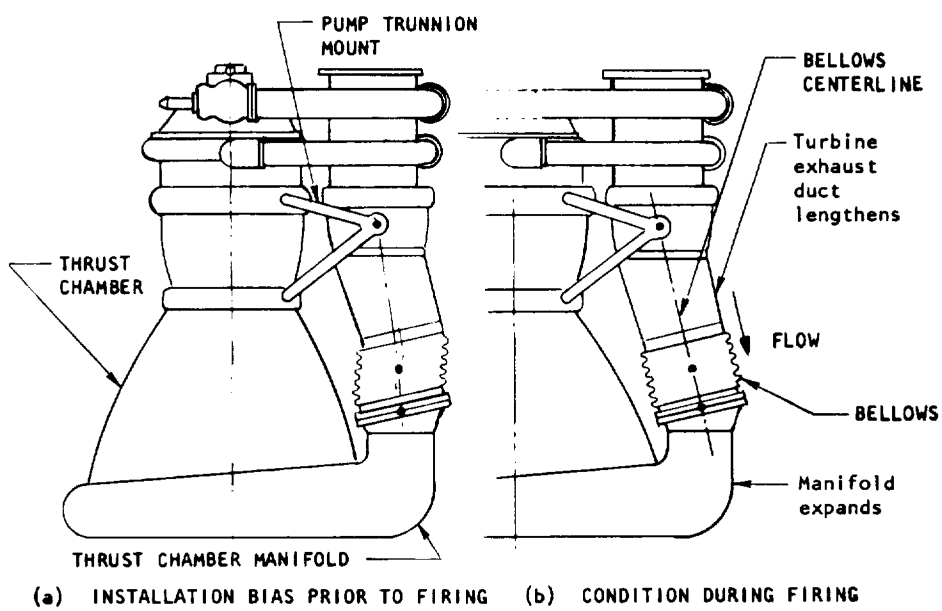


Figure 35. – Installation of compression bellows to minimize loading on support structure.

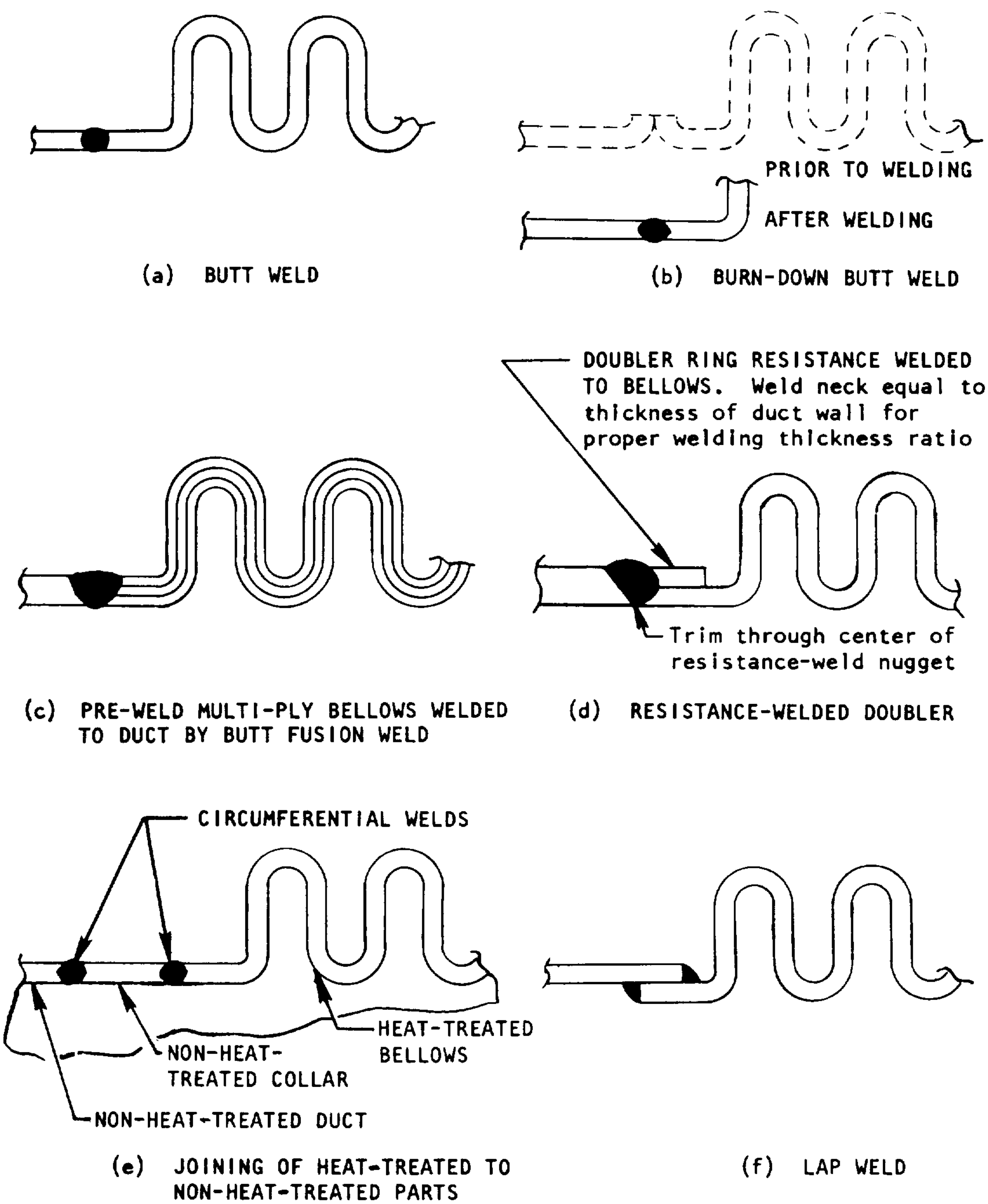
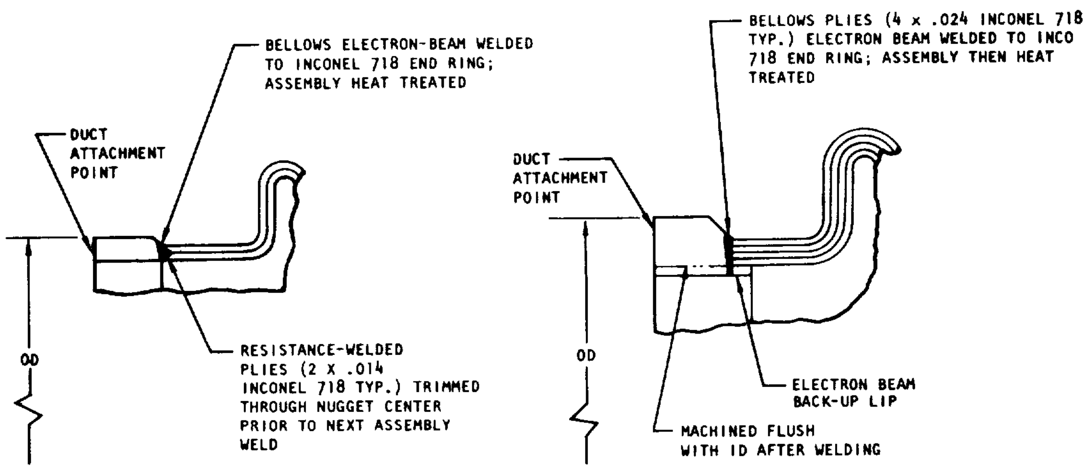


Figure 36. — Typical bellows-to-duct attachments.

2.2.3 Bellows-to-Duct Attachment

The present practice of joining bellows, which are made of thin, sometimes laminated, and often dissimilar materials, to adjacent, more massive parts requires careful manufacturing techniques. Weld fixturing for proper fitup, use of chill blocks, and automatic welding where possible – all must be carefully planned, designed, and applied. Electron-beam welding and diffusion bonding offer advantages for joining dissimilar metals and widely different wall thicknesses and in addition minimize the effect on heat-treat properties because of confined heat input. Heat-treatable-alloy bellows are welded to non-heat-treatable-alloy ducts without disturbing the heat treat condition in the bellows material. This attachment is accomplished by welding a transition section of the duct alloy to the bellows before heat treating so that the final weld to the duct involves identical alloys. Typical bellows-to-duct attachments are shown in figure 36.

The bellows-to-duct attachment welds for the SSME are basically of two types: one for low-pressure, relatively-thin-wall bellows, and the other for high-pressure, relatively-thick-wall parts (fig. 37). The necks of the thin-wall bellows are resistance welded together, then trimmed around the circumference through the center of the nugget. This practice essentially makes the bellows a single ply at its next-assembly weld interface and



(a) LOW-PRESSURE (<500 PSI) BELLOWS

(b) HIGH-PRESSURE (>4000 PSI) BELLOWS

Figure 37. – Detail design of bellows-to-duct attachments on SSME.

improves weldability. The bellows is then electron-beam welded to end rings, and the resultant assembly is heat treated. This subassembly technique allows the structurally important bellows neck weld to be brought up to heat-treat strength whereas, because of the fabrication sequence, the weld cannot always be heat treated when it is welded directly into a final assembly.

The welded junctions of bellows to ducts (usually thicker walled than the bellows) can result in high discontinuity stresses. Butt-welded connections, which are as thick as the bellows wall at the weld and gradually taper to the tube thickness, appreciably reduce stresses resulting from thermal shock. This type of welded connection also reduces discontinuity stresses resulting from radial pressure. The H-1 turbine exhaust cobra-head hood (fig. 38(a)) is an example of fatigue failure in a weld joint that resulted from thermal cycling with mechanical vibration superposed. Fatigue cracks and eventual leakage repeatedly occurred with the sandwich-joint design used (fig. 38(b)). Thermal and stress analyses of the design indicated that high cyclic thermal stresses at engine start and cutoff due to the large temperature gradient across the laminated structure were causing the failures. The butt-welded forged Y-ring design (fig. 38(c)), in which laminations of material in a high-heat-flux region are eliminated (to reduce the thermal gradient) and all welds are Class I, proved to be the design solution, as demonstrated by successful test evaluation; however, the H-1 engine was phased out, and the new design was not accepted for retrofit of existing engines (ref. 110).

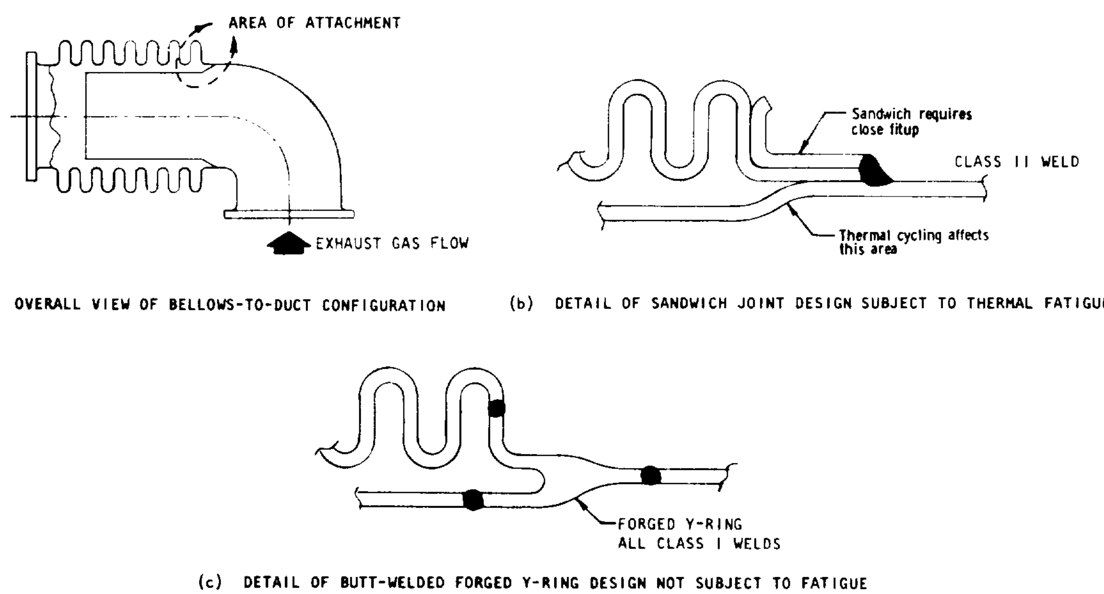


Figure 38. – Design change to preclude fatigue failure of bellows-to-duct attachments on H-1 turbine exhaust.

Resistance seam welding is the least expensive and quickest way to join thin-wall bellows to thin-wall ducts ($\leqslant$ 0.090-in. wall). This method of joining does present some problems, however. During the circumferential seam welding operation, the clearance between the mating parts is gathered into a single large gap as the roller electrodes approach the 360° closeoff point (fig. 39). If this gap is excessive, the electrodes will not be able to force the two parts together for an adequate weld.

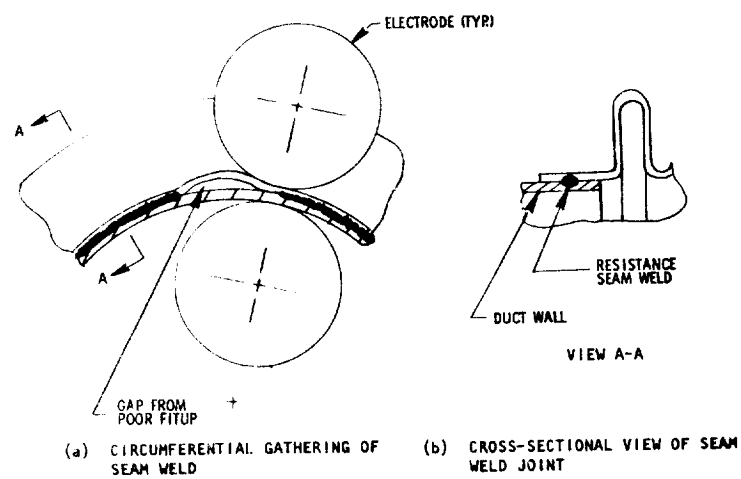


Figure 39. — Fitup problem with resistance seam welding of thin-wall ducts.

Close fitup (i.e., careful sizing of the parts to be joined) and intermittent resistance spot welding around the circumference of overlapping parts prior to welding constitute a fundamental requirement for a structurally acceptable joint. Lap resistance welds also have the disadvantage that there is no conclusive nondestructive test method for proving their adequacy. Radiography can be used, but the film is difficult to interpret because of the built-in cracks adjacent to each side of the weld nugget.

Fusion butt welds or burndown welds are preferable to resistance welds, but require very accurate jigging and tooling to align the butting edges. Butt welds can be inspected radiographically, and they are structurally superior to resistance welds, since the weld is loaded in tension, not in shear as in the resistance type.

A method of joining bellows to ducting, not preferred but sometimes unavoidable, is the induction-brazing technique; this method is useful when the joint is inaccessible for the usual welding process.

2.2.4 Flow Liners

Internal liners (sleeves) for bellows have been successful practical solu: cns to the problem of bellows fatigue failures brought about by flow-induced vibration Liners also are used to effectively reduce the pressure loss of a bellows joint. They have been used as structural members supporting tube coils in the J-2 and J-2S engine heat exchangers. The liners themselves are not without problems, however. Liners must be designed such that, with the bellows at the maximum limits of excursion, they will not bottom out or bind on the bellows or ducting; and they also should be designed with drain holes located as near as possible to the weld attachment to provide for removal of cleaning fluids and contaminants. Typical flow-liner configurations are shown in figure 40.

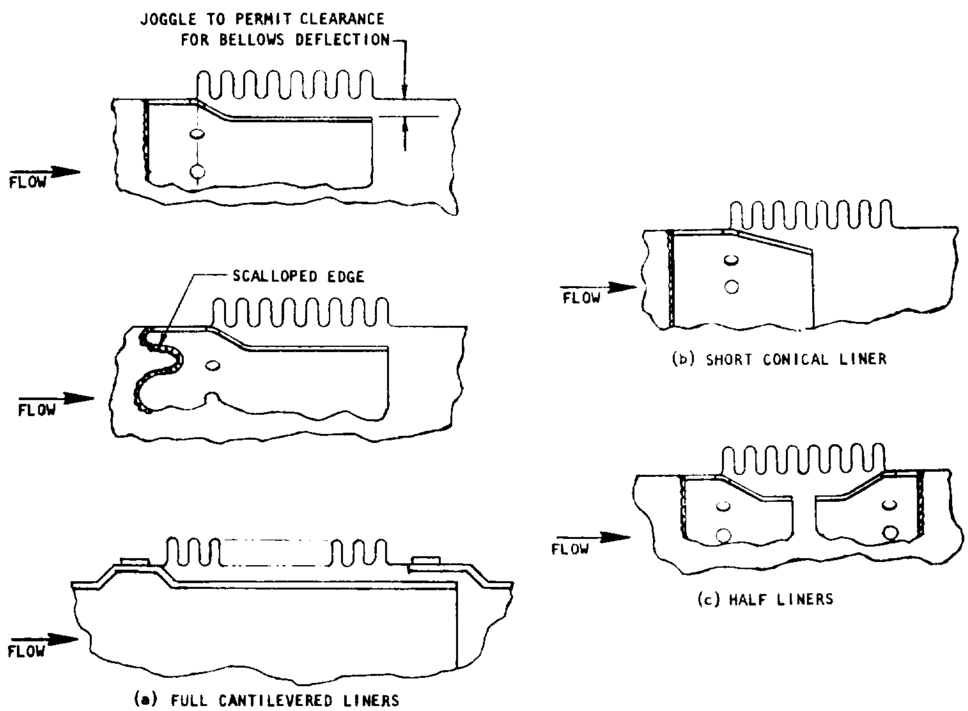


Figure 40. — Typical flow-liner configurations.

Fatigue failures of liners in which cracks appeared in the trailing edge and in the weld joint have occurred. These failures were attributed to vibration modes of the cantilevered end of the sleeve. Increasing the wall thickness, which thereby increased the rigidity of the liner until it successfully withstood the vibration inputs, solved this problem.

Collapse of liners as a result of pressure on the outer diameter of the liner exceeding that on the inner diameter also has occurred (ref. 111). Collapse has been attributed to sudden changes in flowrate that cause a differential static pressure across the liner wall. Since liner diameter usually is smaller than the attaching duct diameter (fig. 40), the liner section behaves like a venturi, and rapid changes in flowrate cause static pressure changes in the liner section. If the static pressures on either side of the liner do not stabilize quickly, a momentary differential can be induced on the liner wall and cause collapse. In the case of fill and drain lines, the stagnation pressure in the drain flow direction (against the OD of the liner) adds to the difference in static pressures and can cause liner collapse (ref. 111). Vent holes in the liner can relieve some of this pressure differential, or the liner can be strengthened to withstand the applied load.

2.3 FLEXIBLE HOSE

The industrial experience in braided flexible hose provided a starting point for advancing the technology to the state required for space-vehicle applications. However, as in the case of bellows joints, pressure ratings in the sizes needed were unheard of. The procurement of commercial braided metal hose of 3- and 3½-in. diameter made of 321 CRES for 750 psi operational pressure in the original Atlas booster engine design is an example. This pressure level was greatly in excess of the commercial rating and left little or no margin of safety. These hoses, however, were used for initial engine tests, and pressure ratings eventually were increased by modifying the design to meet aerospace requirements. Thousands of hoses in this size (3 to 3½ in.) have been used for both cryogenic-oxidizer and hydrocarbon-fuel applications in the Navaho, Atlas, Jupiter, and Thor engines, a very large population for space-vehicle hardware.

Little was known about the pressure losses in hoses at the Reynolds number used in rocket engine operation ($Re > 10^6$). Fatigue life and cryogenic-temperature properties of the materials available were sketchy. These voids eventually were filled to permit successful operational designs.

Coiled tubing, an alternate design for small-diameter bellows-type flexible hoses, is not used on any engines of the Saturn or earlier programs. However, rocket engine manufacturers do use coiled tubing consistently in tank-pressurization heat exchangers. The primary purpose is not to achieve line flexibility, a secondary benefit, but to provide the maximum heat-transfer surface area possible in a minimum of space.

In the Centaur stage, a coiled tube is used for the pneumatic lines to the main engines; also, flexible bends are used in propellant recirculation lines and tank pressurization lines (all $\leqslant$ 1 in. diam.).

2.3.1 Routing

The same practices for routing ducts or lines generally apply to assemblies of braided flexible hose, except that all systems are of the tension type. A minimum number of flexible sections is used in an assembly, and they are strategically located to achieve the maximum line motion possible with minimum motion of the flexible section. References 112 through 118 provide design guides on use of flexible hose.

A typical configuration for flexible-house assembly that displays sound design practice is shown in figure 41. Note that as opposed to the layout shown in figure 5 the entire wraparound lies in the gimbal plane rather than passing through it. The assembly is for a

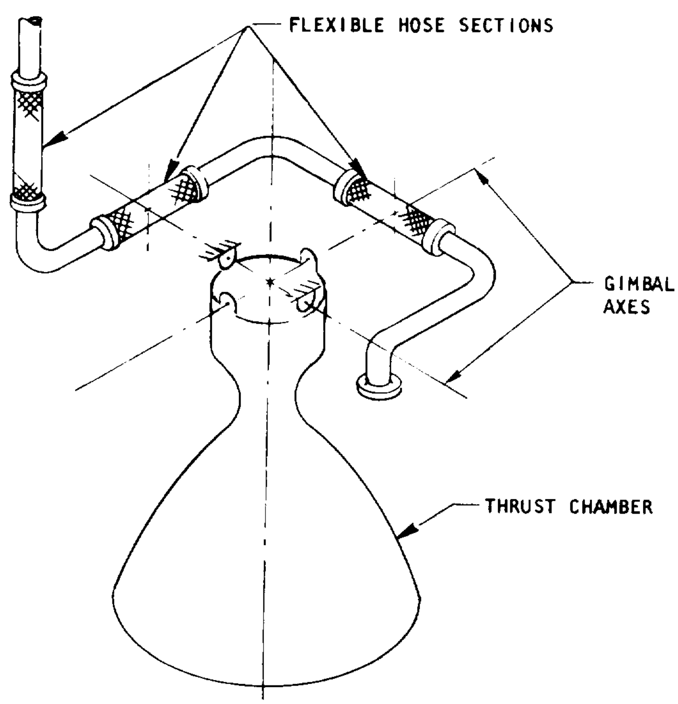


Figure 41. — Gimbal-plane wraparound hose configuration.

rocket engine with a thrust-chamber head-mounted gimbal used for thrust vector control. For two of the three flexible sections, the longitudinal centerlines are located in the gimbal plane. The center of the live length of each of these two flexible sections is located on each of the gimbal axes. A third flexible section is located in the vertical leg to provide universal motion of one end of the line with respect to the other. This method of locating the flexible sections limits their motions to the gimbal angle only, thereby minimizing bending stresses and attendant fatigue.

When the hoses are not inherently stiff enough to resist motions imparted by mechanical vibration or undesirable displacements due to acceleration loads, then clamps and support brackets, strategically located, are used as dampers.

Other configurations have been used for crossing the gimbal plane when lack of available installation space prevented the wraparound approach. The engine-to-vehicle interface flex lines of the J-2 engine (fig. 42) are a case in point. These lines were arranged in a U-shaped routing configuration with a section of braided flexible hose in each leg of the U.

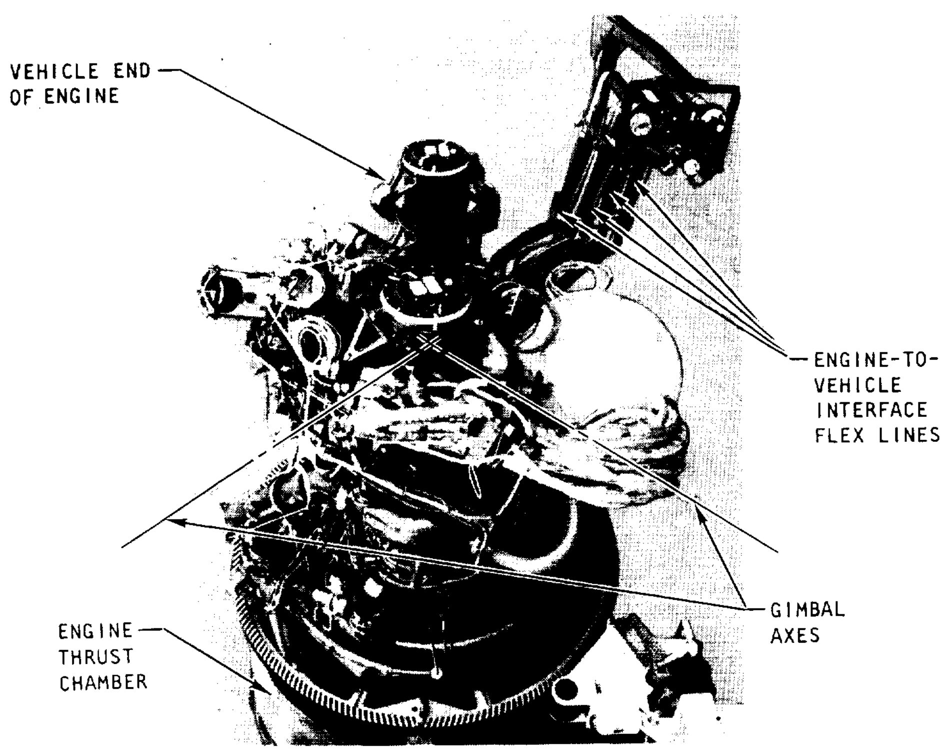


Figure 42. – Off-center hose configuration crossing gimbal plane used in engine-to-vehicle lines of J-2 engine.

A considerable amount of test data has been accumulated on the flow losses through flexible metal hoses (refs. 119 through 125); these data are used in the design phase of a new application. The pressure drop in a convoluted innercore is 10 to 15 times that for a comparable-diameter hard tube. Bends in flexible-hose elements are avoided where possible. The references show that, for comparable geometries, the ratio of the pressure-loss factor for a convoluted elbow to the pressure-loss factor for straight convoluted section is much greater than the ratio of pressure-loss factor for plain-tube elbow to the pressure-loss factor for a straight tube. In addition to the pressure-drop disadvantages, the bent flexible configuration maintains a steady-state deflection stress on the part in an installed position; this stress can adversely influence fatigue life.

2.3.2 Sizing

Current practices in sizing flexible hose involve the same general considerations used for sizing lines (sec. 2.1.2). A practice on most flexible-hose designs is to keep velocities of gases below a Mach number of 0.3 in order to avoid pressure loss and flow-induced vibration. This approach is not supported by either analytical results or experimental data. Analyses of pressure loss (refs. 119 through 125) and flow-induced vibration (refs. 84 through 90) are better methods for avoiding problems with either phenomenon.

Standard design practice for manufacturers of flexible metal hose is to make the innercore inside diameter greater than the attaching-tube inside diameter. This procedure introduces expansion and contraction losses that must be traded off with hose pressure losses for an acceptable total loss.

2.3.3 Innercore

Analysis of flexible-hose innercore for flow-induced and mechanically induced vibration is the same as for bellows (sec. 2.2.1.2), with the exception that the braid restrains the convolutions to vibrate individually. Other considerations such as pressure capability, corrosion resistance, manufacturing practices, and handling protection for flexible hoses are similar to those for bellows (secs. 2.2.1.1, 2.2.1.4, 2.2.1.5, and 2.1.5.7).

2.3.3.1 OPERATIONAL CAPABILITY

Temperature extremes to which components are subjected in a rocket engine system usually will exempt a nonmetal hose from consideration. Even if the hose is suitable for conveying a particular fluid, then proximity to hot or cold running components, placement in view of the exhaust plume, exposure to exhaust blowback, a test-stand fire, or exposure to cryogenic temperature could cause failure of a nonmetal hose. These adversities seldom pose problems with metal hoses.

Annularly convoluted metal innercore is the predominant type of pressure-carrying member in aerospace flexible hoses. The methods of stress analysis for bellows are applicable for braided-hose innercore; however, it cannot be assumed that the total live length will be active in bending. The "Chinese-finger-trick" effect of the braid, which causes a compressive load on the innercore OD, plus the bottoming out of the braid wires in bending, do not permit a smooth arc (constant radius) deflection of the innercore.

Helically convoluted metal innercore, although more readily available and less expensive than the annular type, is not generally used in aerospace applications. The scarfed end of the innercore presents a difficult joint at the end-fitting attachment. Puddles of braze or weld are required, with resultant poor control of quality. Torsional deflection imposed on the hose during assembly creates a preload that tends to loosen end fittings if they are flared-tube threaded connections. This same twisting effect is created when the line is pressurized.

Although some flexible-hose users insist that seamless tubing be used, no evidence has been found in J-2, F-1, H-1, Atlas, or Thor programs to indicate that this practice is necessary. Seamless tubing is much more expensive than rolled and welded tubes, and since bellows are rarely made from standard tubes, the lead time for special mill runs can be excessive. Rolled and welded tubes, fabricated properly with good fitup and welding, have proved to be very successful. The welds are X-rayed for quality verification and put through a torturous ordeal in the bellows forming process. If no leaks occur after forming, the assurance of high reliability is excellent. Also, the uniformity of wall thickness for rolled and welded sheet is better than that achieved with seamless tubing because of ID/OD eccentricities inherent in tube forming.

Nonmetal innercore such as rubber or tubular or annularly convoluted Teflon has not been used extensively in rocket engine systems. Atlas, Thor, and F-1 engine experience with tubular Teflon included problems of kinking, cracking, leaking, and burning (pin holes).

Kinking, the most prevalent problem in Teflon hoses, results in increased differential pressure and possible fracture of the innercore. Kinking problems have been resolved by the addition of an internal helical spring throughout the hose ID, mechanically attached to the end fittings. Another solution is convoluting of the innercore; however, the convoluted Teflon type has limited pressure-carrying capability.

Cracking because of embrittlement of the Teflon has occurred. Curing of the Teflon liner is critical, because if incorrectly accomplished, the liner is subject to longitudinal and radial cracking.

Leakage of swaged end fittings used on all nonmetal hoses is a problem because of inherent design weaknesses. The mechanical seal arrangement in swaged fittings is dependent on initial application of pressure during fabrication to seal properly. Torquing of this joint is critical, and if the torque is lessened either accidentally or during disassembly, the joint will leak.

Pinhole burning, presumably caused by electrostatic discharge from the liner to the braid, has occurred with Teflon liners. This problem supposedly has been solved by impregnating the innercore with carbon, thus making it conductive and presumably preventing accumulation of static charge.

Despite all these problems, Teflon has been used successfully in some applications. The Titan III engines use both normal (white) and carbon-impregnated Teflon flex hoses in sizes ranging from 3/8 in. to 1 in. in diameter. About half of these are carbon impregnated; changes to the carbon type have been made only for new designs, since it has never been conclusively proved that carbon impregnation is the solution to the problems of pinhole burning caused by electrostatic discharge.

On the S-IC stage of the Saturn vehicle, Teflon hose was used successfully with RP-1 at 2200 psi operating pressure. Teflon innercore has been used successfully for the airborne hydraulic system on Centaur.

Rubber hose is limited to noncryogenic or non-high-temperature applications. Rubber is subject to age problems because its quality deteriorates with time. It also will swell when subjected to trichloroethylene flushing; the swelling restricts flow and can cause failure of the end-fitting joint.

2.3.3.2 BENDING MOMENT

The bending moment required to bend a flexible hose is a consideration because of the loading applied to adjacent components. For example, reduction of bending moments of flexible hoses that cross the engine gimbal plane reduces the actuator loading required to gimbal the engine. Lubricants applied to braid wires and the outer surfaces of the innercore reduce bending moments. Compressing the innercore axially before installing the braid results in more convolutions per inch and a lower bending stiffness, at the expense of increased weight and material cost. Innercore compression also improves the ability of the hose to withstand pressure impulses.

2.3.3.3 BUCKLING STABILITY

Buckling stability, which is of concern in the design of a bellows, is not a source of concern in the design of a flexible hose. The support that the braid provides for the innercore prevents buckling unless end attachments (on installation) prevent the braid from pulling taut when the hose is pressurized.

2.3.4 Braid

The braid consists of a woven tubular covering that restrains the innercore against elongation while giving it lateral support and protection. Since the braid absorbs the entire separating load, its strength largely governs the burst pressure* of a hose; stiffness and yield strength govern the length or stretch. The braid covering on the majority of hoses used in rocket propulsion systems is woven of stainless-steel wire. Tubular braid may be woven directly on the innercore. For bellows-type innercores, however, the braid generally is woven in continuous lengths or sheaths that are cut as required and slipped over the bellows during assembly. Sections of braid are cut at a length slightly greater than that of the convoluted innercore. A tensile load then is applied to the ends of the braid section, and the section is attached to the end fittings by brazing or welding.

The braid-weave pattern must be such that the wires will not bind or bottom out within the angular-deflection design limits of a particular flexible section. If the wires within a braid bottom out against each other when the hose is angulated, further deflection will be distributed over a shorter live-length section, the result being higher stresses and shorter fatigue life.

Tubular wire braid is available in a variety of constructions. In addition to variations in numbers and diameters of wires, there are variations in the form or grouping of the wires, the weave, the braid angle, and the number of braid layers. The most commonly used form of wire is the "beamed" type in which each carrier bobbin of the braiding machine is wound with a group of parallel wires called a "pick," or an "end," or a "carrier." In another type, each pick consists of a number of wires that have been first braided into a flat braid before being wound on the bobbin. The size and number of wires per carrier are selected to obtain flexibility and strength within the machine capabilities.

Two weaves are commonly used: plain weave, two-over/two-under; and diamond weave, one-over/one-under. The term "basket weave" is loosely applied to any tubular braid. Plain weave permits the greater number of wires and therefore the greater coverage. (Coverage is critical in electrical shielding applications, but is of secondary importance in metal hose as long as the mesh remains tight enough to retain the innercore.) Large braids are limited to plain weave. Diamond weave is the tighter construction. The cost of either weave is about the same. Typical braid configurations are shown in figure 43.

The braid angle or helix angle is the angle whose tangent is the pitch divided by the braid circumferential length per pitch, the pitch being the axial distance taken by any given wire to make one complete turn. An angle of approximately 45° is presumed to give maximum flexibility consistent with good end strength and burst resistance for hose with metal innercore.

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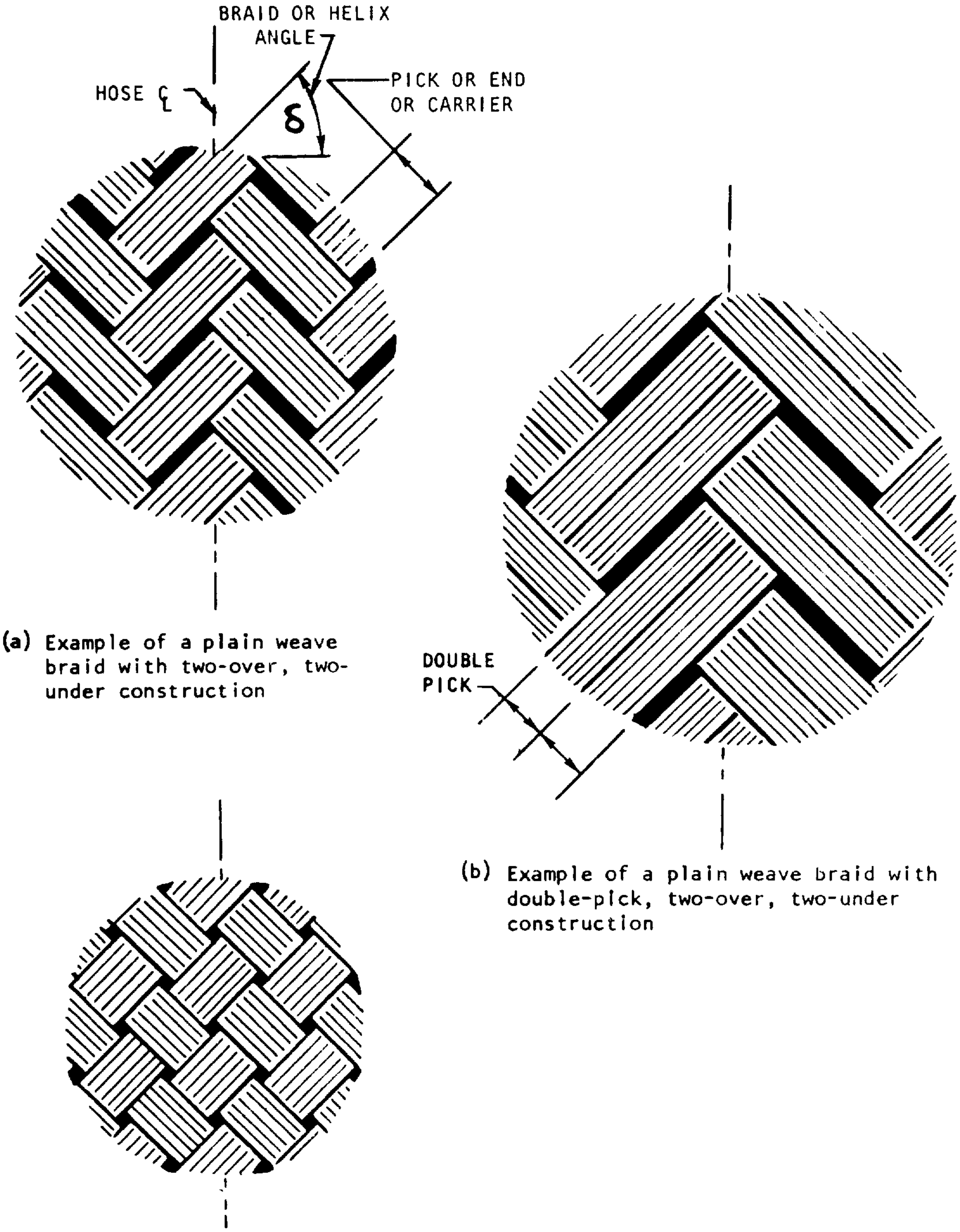
*Maximum pressure the hose can retain without losing pressure or fluid.



(a) Example of a plain weave braid with two-over, two-under construction

(b) Example of a plain weave braid with double-pick, two-over, two-under construction

(c) Example of diamond-weave braid or one-over, one-under pattern

Figure 43. – Typical weaves used in construction of tubular wire braid.

For most efficient use of braid, the wires should make a traverse of at least 360° along the length of the hose. Less of a traverse results in high bending loads and excessive bending of individual wires at the attach points, thus causing fatigue failure. Sufficient wire cross-sectional area, plus a suitable margin, is supplied to resist the pressure separating load of the bellows.

Multiple layers of braid may be used to achieve greater strength within wire-handling capacity of the braider and without too great a sacrifice in flexibility. Because of the difficulty in obtaining perfect load distribution between layers of braid, it is reasonable to assume that the second layer is only 80-percent efficient.

Tubular braid is readily available in a variety of metals and sizes up to 18 in. in diameter. The tensile strength of stainless-steel braid wire is about 120 000 psi, although high-tensile (up to 270 000 psi) braid is obtainable. Tubular braid is identified or described as follows: ID X number of wires per carrier X number of carriers X diameter of wire X kind of material.

The end strength of a braid may be calculated from the relation

$$F_B = n\ F_w\ B_e \sin \delta \tag{2}$$

where

F_B = braid end strength, lbf

n = total number of wires in braid

F_w = strength of a single braid wire, lbf

B_e = braid efficiency (a factor providing for stress intensification in the braid and in the braid attachment)

= 0.93 for annealed wire

= 0.85 for hard-drawn wire

= 0.80 for a second layer

δ = braid angle as defined previously (fig. 43), deg

The burst pressure (as limited by braid end strength) of a bellows-type metal braided hose may be found from the expression

$$P_b = \frac{F_B}{A_{eff}} \tag{3}$$

where

P_b = burst pressure, psi

A_{eff} = effective area of bellows, in.^{2}

$= \dfrac{\pi D_m^2}{4}$, where $D_m = \dfrac{OD + ID}{2}$

The restraining force contributed by the bellows is practically negligible.

The approximate elongation of the braid when the hose is pressurized may be calculated from the expression

$$\xi = \frac{P_i \, \ell \, A_{eff}}{n \, E \, A_w} \tag{4}$$

where

ξ = braid elongation, in.

P_i = internal pressure, psi

ℓ = total length of one braid wire between end connections, in.

E = Young's modulus for a braid wire, psi

A_w = cross-sectional area of each wire, in.^{2}

It should be noted that this expression includes no allowance for slack in the braid. It also assumes that the braid stress is within the elastic limit. References 126 and 127 present further information on braid design.

Braid wire is subject to corrosion; reference 128 presents a summary of a corrosion incident. Because of the experience with corrosion of 300-series-CRES braid materials and the exceptionally long life requirement of the SSME (100 missions), all SSME braided hoses are made of nickel-base alloys.

2.3.5 End Construction

Two types of end construction (i.e., juncture of innercore, tube adapter, and braid), evolved after many years of experience, were used extensively on the J-2 and F-1 engines and on the S-II stage. These constructions are (1) a welded and brazed design for use at temperatures less than 400°F (fig. 44(a)), and (2) an all-welded design for high-temperature (> 400° F) applications (fig. 44(b)); welded construction is required for high-temperature applications because of the loss of strength of the braze material. These designs have been used successfully for lines up to 3 in. in diameter.

A construction different from those described above is used on large-diameter (3½- and 4-in. ID) hoses developed for the Atlas and Thor gimballing feed systems. The tube ID is large enough to permit a lap weld of the pressure-carrier neck to the tube adapter (fig. 44(c)). This design is different from the brazed design used in the previously mentioned smaller sizes because it was developed in working with different suppliers. It is believed that the designs for the smaller sizes can be extrapolated to larger diameters; however, there has been no practical experience with the larger diameters.

Many other end constructions have been used successfully in rocket engine and vehicle applications. Some of these are shown in figure 45.

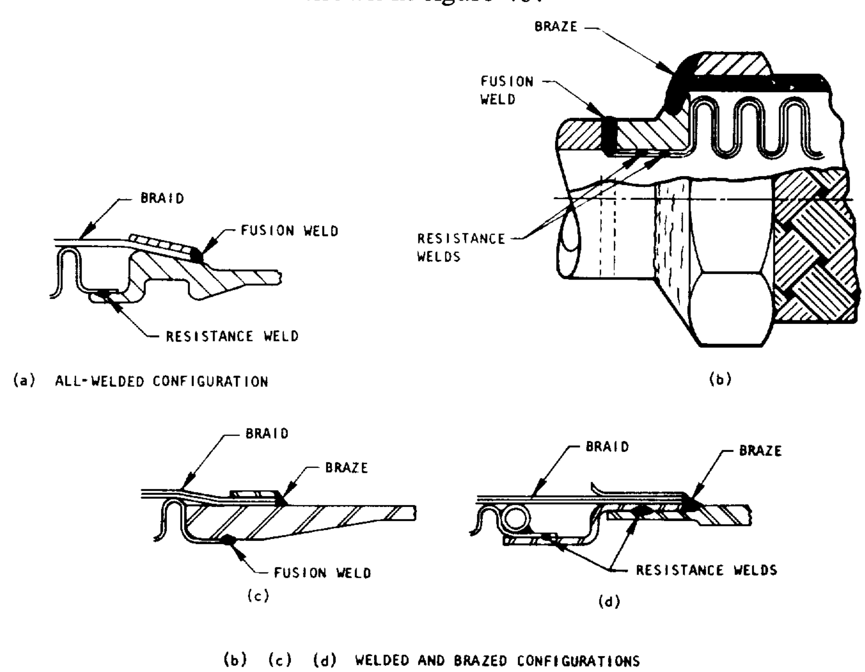
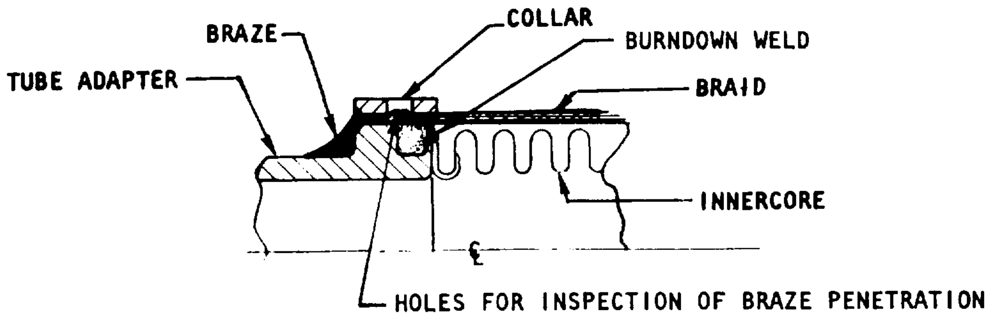
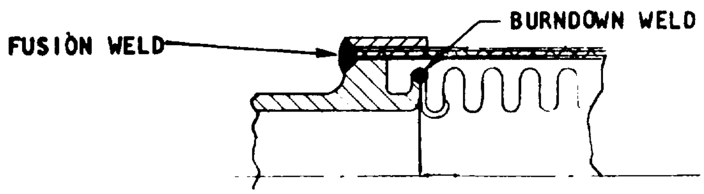


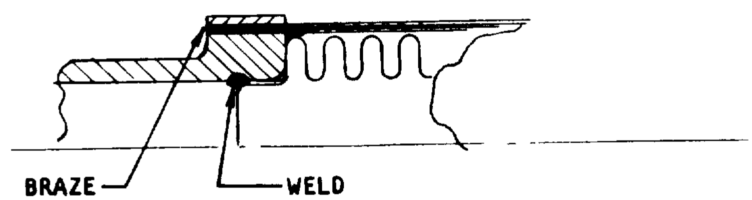
Figure 45. — General-purpose end constructions for braided hose.



(a) End construction for small-diameter braided metal hose (temperatures from cryogenic to 400°F)



(b) End construction for small-diameter braided metal hose (temperatures above 400°F)



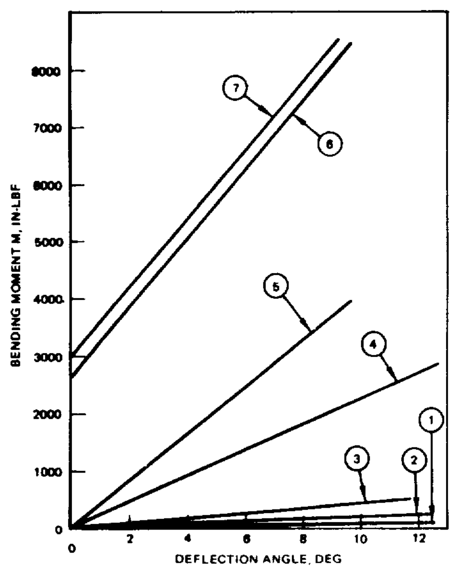
(c) End construction for braided metal hose 3-in. ID and over (temperatures from cryogenic to 400°F)

Figure 44. – Braided-hose end construction for specific applications.

A common problem, especially with small (¼-in. diam.) hoses, is the overangulation and subsequent braid bulging at the ends of the hose. A solution widely used with commercial elastomeric hoses is the placing of external coil springs over the braid for a few inches at each end of the hose; this practice precludes exceeding the minimum bend radius and damaging the hose or braid near the braid rings.

2.3.6 Bending Moment

The moments required to deflect braided metal hoses are a function of hose geometry and construction (e.g., diameter, innercore thickness, number of plies, live length, number of layers of braid, number of wires and diameters) and operating pressure. Data on bending moments can be obtained from the manufacturer or by test. Figure 46 presents some typical values.



CONFIGURATION											
ITEM	ID	OD	NO. PLIES	t/PLY	PLY MTL	BRAID LAYERS	BRAID MTL	BRAID	LIVE LENGTH	OPER PRESS*	NC
1	.500	.790	1	.008	321	2	321	32X7X.012 48X6X.012	4.50	2000	72
2	.756	1.080	1	.013	321	1	321	48X6X.012	4.50	2000	59
3	1.70	2.18	3	.010	321	2	321	48X9X.016 48X11X.016	13.00	0	72
4	1,70	2.18	3	.010	321	2	321	48X9X.016 48X11X.016	13.00	900	72
5	3.50	4.00	3	.008	321	2	321	48X13X.020	10.50	850	40
6	4.00	4.50	3	.012	321	2	321	96X6X.025	12.00	1080	54
7	4.50	5.00	3	012	321	3	321	96X6X.025	12.40	1080	56

NC = NUMBER OF CONVOLUTIONS
ALL DIMENSIONS IN INCHES
*PSIG

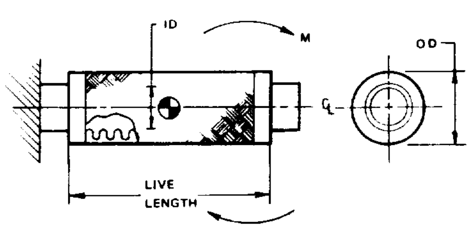


Figure 46. — Typical bending moments for various configurations of braided metal hose.

2.4 FILTERS

The function of a filter in a fluid system is to reduce the amount of contamination in the system to a level at which the remaining particles do not affect the performance characteristics of contamination-sensitive components. The filter must be of a size capable of removing the necessary amount of system-generated, built-in, or environmental contamination throughout the required service life without excessive increase in differential pressure across the filter element (refs. 129 and 130). Typical filter applications are summarized in table II.

2.4.1 Filter Element

Mechanical filter elements remove contamination from a fluid system by physically trapping the particles as they pass through the filter material. The filter element can be either a surface or a depth type. A surface filter performs its function by trapping particulate contamination on the upstream side of the filter surface at the inlets of the filter material capillaries. A depth filter performs its function by trapping particulate contamination within the intricate capillary system of the filter material; it has the ability to remove particles with one dimension significantly greater than either of the other two dimensions (e.g., fibers). The significance of longest or largest dimension for the particle as compared with the diameter of a spherical particle is emphasized in reference 131. In rocket systems, most surface filters have been woven wire cloth and most depth filters have been stacked etched metal disks. Reference 12 contains excellent descriptions of various types of filters and filter materials.

Surface filters have the following characteristics:

- Good control of maximum particle size
- Easy to clean
- Thin cross section (necessary for design of maximum area configurations)
- Free from filter-material (media) migration and release of initial element contamination.

Depth filters have the following characteristics:

- Longer life than surface type for equal area
- Thicker cross-sectional area than surface type, hence greater weight for equal area
- Generally poor control of maximum particle size, because of nonuniform pore size
- More efficient removal of fine particles for equal large-particle control

- Tend to erode

- Prone to filter-material (media) migration and release of initial element contamination.

The majority of filters in use are of the surface type, primarily because they provide maximum surface area for minimum weight. Selection of the optimum filter design for a given application requires a definition of the following parameters:

(1) Largest particle size that can be tolerated by the operating components

(2) Amount and type of contaminant that will be encountered during service life, including all cleaning and test fluids

(3) Maximum pressure drop that can be allocated by the system

(4) Space that can be made available within the system

Of these four basic variables, the first can be defined with a fair degree of accuracy, since the allowable particle size can be determined by a review of the clearances or the sizes of capillaries in a contamination-sensitive component. A review of system-performance requirements reveals the degree of severity of possible component malfunction due to the entrapment of a particle on a valve seal or other critical surface. The second variable, however, is more difficult to define because of the following factors:

- The amount of contamination remaining after component cleaning and system flushing can be calculated only to an approximate minimum value.

- The amount of contamination that is environmentally introduced during service and assembly is almost impossible to predict.

- During laboratory tests, the fluids and temperatures used for evaluation often vary considerably from those encountered by the filter in service.

The third and fourth variables can be defined by performing a systems analysis and by making a trade study to determine whether it is more advantageous to have the filter externally mounted or to have it installed as an integral part of the component.

2.4.1.1 FILTER RATING

The filter rating is intended to specify the largest particle that can pass through the filter; the rating thus determines the degree of protection for downstream components. Several different ratings are used in the filter industry; some tend to be misleading if not fully understood by the user. The most definitive of the various ratings is the maximum particle size rating (MPR), which controls the maximum (longest) dimension of any particulate contaminant allowed downstream of a filter. Other filter ratings commonly used are "absolute" and "nominal". The absolute rating applied to a filter unit requires the unit to retain all spherical particles equal to or greater in size than the absolute rating. This rating controls only the second largest dimension of contaminant allowed downstream of the filter. The absolute rating is often referred to as the glass bead rating (GBR). The nominal rating applied to a filter unit requires that the unit retain a certain percentage (by count or by weight) of spherical particles or graded dust equal to or greater in size than the nominal rating. The nominal rating is always less than the absolute rating. The nominal rating is less well defined than absolute rating, and it is a relative indication of filter capability only. The use of nominal rating in the filter industry is becoming obsolete because of the ambiguity of its definition.

The maximum particle size rating can be determined by test. The test involves the use of a readily identifiable contaminant that contains particles of various sizes and shapes. A test procedure for determining maximum particle size is described in reference 132.

The absolute rating (GBR) of a filter also can be determined by test. Tests are based on filtration of an artificial contaminant (glass beads) under specified test conditions. Effluent containing artificial contaminant that has passed through the test filter is collected. The artificial contaminant is passed through a membrane filter and the particles held on the membrane are scanned for the largest particle (ref. 133). Since approximately 60 percent of the beads required for testing a 15-μ filter fall in the 25-μ range or smaller, they tend to coagulate on the membrane and obscure beads that are larger. The test procedure most often utilized is well defined in reference 133, but provisions for preventing coagulation are lacking.

A means of determining nominal (average) pore size is the mercury intrusion method, which is not in general use because of the toxicity of the mercury and is applicable only to filter cloth in the flat state. A detailed explanation of the mercury intrusion test is presented in reference 12 (sec. 15.8.5.2).

Although the use of absolute rating (GBR) is more prevalent, the maximum particle size rating (MPR) is a truer indication of the protection afforded to downstream components. A correlation between MPR and GBR has been established for both twilled double Dutch weave (TDDW) and plain Dutch single weave (PDSW) wire cloth (ref. 132), viz., the maximum particle size rating (MPR) is approximately 2.5 times the absolute (GBR) rating. As an example, a filter element using TDDW wire cloth having an absolute rating of 10μ will permit the passage of particles up to 25μ into the downstream system. The correlation between MPR and GBR for other types of filter elements (e.g., etched disk) needs to be established by test for each particular type. Reference 132 lists the correlation factors for a limited number of filter materials and kinds of construction.

Maximum-particle-size tests are not practical for acceptance testing of production filters. The glass bead test is a destructive type of test and is never used as an acceptance test; however, correlation with a nondestructive bubble-point test permits verification of absolute rating for production filters. The bubble-point test consists of wetting the filter with a liquid of known surface tension and then determining the gas pressure required to force a bubble through the wetted pores. The larger the pore, the lower the pressure at which a bubble or stream of bubbles will form. The detailed procedure is described in reference 134.

2.4.1.2 SYSTEM CONTAMINATION/FILTER AREA

The primary function of a filter is to prevent damage to contamination-sensitive components in the fluid system. To ensure reliable performance of a system, it is necessary to reduce the contaminants that are entrained in the operating fluid to a level at which the remaining particulate matter will not cause damage to the critical operating components. The determination of how much filter area is needed is based on accurate prediction of the amount of system contamination that can be expected for any given application. Several companies and universities have been working on the problem of accurately predicting system contamination, but as yet no real solution is in sight. There appears to be no correlation between the actual and predicted amounts, sizes, and types of contaminants; the reason may be the variation in techniques utilized in manufacturing and building systems and components. A detailed description of some of the problems encountered may be found in references 135 through 138.

The lack of a realistic basis for determining the amount or makeup of contaminant that is likely to pass through a filter during its service life has led to some arbitrary compromises. In order to demonstrate that sufficient filter area has been provided for a particular design, a specified amount of contaminant is required to be retained by the filter without exceeding some maximum differential pressure (contaminant tolerance). The most frequently used test contaminants are AC-Fine and AC-Coarse dusts.* The two dusts may not ideally represent the true makeup of system contamination, but they do provide a baseline material for evaluating the ability of a filter to retain particles presented to the upstream side under flow conditions. The amount of dust to be presented to the filter is also an arbitrary compromise. Reference 132 provides an excellent source of information on tests that were performed to determine the contaminant tolerance of various filter materials for both gaseous and liquid flow.

A good example of a filter problem related to unpredictable system contamination occurred on the SSME. The main pneumatic filter had satisfactorily demonstrated its ability to retain a specified amount of AC-Coarse dust without creating excessive differential pressure. When engine testing began, several filter failures were caused by the sudden accumulation of fine contaminants (smaller than the absolute rating (GBR) of the filter) that plugged the pores

*These test dusts are Natural Arizona Dust supplied by General Motors Phoenix Laboratory and classified to specific particle-size distributions by the AC Spark Plug Division of General Motors Corporation.

and greatly increased the filter differential pressure. As a consequence of the higher pressure, several filter elements ruptured. The source of the contaminants was the test facility plumbing. To resolve the problem, the quality of filtration in the test facility was made better than the quality of filtration called for in the engine system.

2.4.1.3 RESIDUAL FILTER CONTAMINATION

Residual contamination is the amount of contaminant remaining in the filter after final cleaning that can potentially slough off and pass into the downstream system. Contaminant particles are accumulated by filters during the various stages of manufacturing. A last-stage cleaning of the filter removes some percentage of the particles, the amount depending on how rigorous the cleaning process is. To facilitate final cleaning, it is desirable therefore to limit the amount of contaminants accumulated by the filter during manufacture. Cleaning processes are expensive, and when a filter has accumulated a great deal of contaminant the cost of final cleaning can often exceed fabrication costs by several orders of magnitude. To limit the accumulation of particles during fabrication, assembly, and testing of filters, it is necessary to control the environment during these stages of manufacturing. All filters used in the Space Shuttle flight system must conform to the requirements of reference 131 to minimize residual contamination. The cleanliness level (residual contamination) actually is a measure of the particle release rate of the filter and therefore is greatly affected by the intensity of the sampling method. Normal practice is to test only the effluent side of the filter for particle release while subjecting the element to a sonically-induced cavitational field (ref. 139). It should be emphasized that cleanliness levels specified for filters are only an indication of particles that can potentially slough off. If the operating environment is not as severe as the sampling method, the potential for sloughing particles decreases proportionately.

2.4.1.4 FILTER MATERIAL AS CONTAMINANT

Media migration is a commonly used term for the presence in the system fluid of particulate contaminant identifiable as originating in the filter element or filter-supporting structure. Some filter materials are subject to media migration more than others because of the method of manufacturing. Filter materials subject to migration are sintered porous metal, pressed paper, matted fiber, glass fiber, sintered plastic, fired porcelain, bonded carbon, and bonded stone (ref. 12).

Methods of testing for media migration vary considerably, and standards for verification of migration vary even more. In general, the most accepted method involves a combination of thermal-shock and vibration tests. A filter is exposed to its lowest service temperature for a period of time and then is rapidly transferred to its highest service-temperature environment for an equal period; this procedure is repeated for several cycles. After the thermal shock tests, the filter is subjected to a vibration environment commensurate with its service environment for a specified period of time. On completion of testing, the filter is flushed with a specified amount of fluid (usually 500 ml of Freon TF); the fluid is collected and passed through a 0.45μ membrane pad. The membrane pad then is analyzed for particles identifiable as filter or supporting-structure material. Unfortunately, the sample collected is composed not only of possible media particles but also has residual-contamination particles. In the simplest case, if the particles are all nonmetallic and the filter is constructed entirely of metal, it can be concluded that no media migration has occurred. If the particles are metallic, further analysis is required, and filter material must be distinguished from residual contamination.

Individual particles from a representative sample of the pad can be identified with equipment such as an electron microprobe X-ray analyzer. Comparison of the chemical analysis of a representative number of the particles with the filter-material chemistry provides the basis for determining if media migration has occurred. This kind of analysis is expensive and time consuming and is resorted to only when a visual microscopic examination of the particles does not suffice.

It has been the practice on occasion to utilize the cleanliness-level (residual contamination) requirements as the basis for media-migration verification. The basis is acceptable so long as the actual residual-contamination particle counts fall well below the maximum particle counts specified. If the cleanliness-level requirements are stringent, the above method is not used. For example, the hydraulic filter for the SSME satisfactorily demonstrated no media migration. The pneumatic filter for the same engine had more stringent requirements, and the post-test particle counts exceeded the cleanliness allowable counts. It was necessary to resort to electron microprobe X-ray analysis in order to verify that the particles were not comprised of filter or structure material.

2.4.1.5 PRESSURE DROP

Prediction of the differential pressure created by a given flowrate presents one of the more difficult problems in filter-element design. Generally, in the case of a surface-type element, the pressure drop across the filter medium for either fine or coarse mesh is relatively small when compared with the loss created by the element entrance or exit configuration. The accepted practice is to generate flowrate-vs-differential-pressure curves by actual test after the element is fabricated. Empirical equations for calculating basic flow resistance for various filter materials are presented in reference 132. The equations provide an excellent means for evaluating filter area requirements; however, they do not consider entrance and exit losses. The slope of the curve for pressure-drop buildup is a function of flow velocity, type of fluid, and filter material. The collapse pressure* of an element must be greater than the maximum transient differential pressure expected under flow conditions.

*Collapse pressure is the maximum differential pressure that the filter element must be able to withstand without any permanent deformation that results in rupture or degradation of performance that releases contamination into the downstream system.

System-transient differential pressures that exceed the nominal can result in deformation of the filter element. The deformation can change the flow resistance of the filter, particularly in the case of a tightly pleated woven-wire-cloth configuration. The pleats can become pinched together, thus reducing the effective area of the filter. In cases where high system-transient differential pressures exist, a coarse backup screen is used to support the woven-wire-cloth element.

A good example of a problem with filter pressure drop occurred in development of the LEM ascent engine. A coarse filter upstream of the injector was found to be inadequate because fine particles contained in the propellant that passed through tended to plug the holes in the thrust chamber injector. A fine pleated-wire-mesh filter was installed between the coarse filter and the injector. The new filter, however, caused a new problem: after a period of time, performance shifts during engine operation were noted. The shifts in the performance were found to be due to a change in the resistance of the filter caused by pressure surges that distorted the pleated elements. A redesign of the element to eliminate the pleats resolved the problem.

2.4.1.6 TESTING

Filters are not acceptance tested with the actual propellants used in engines because of the difficulties (e.g., facility requirements or safety considerations) in testing with cryogenics or highly toxic or corrosive propellants. Qualification tests generally are not performed with actual propellants unless adequate safety precautions can be taken. At present, test fluids include Freon TF, hydraulic oil, trichloroethylene, water, isopropanol, and denatured ethanol. During evaluation or performance-demonstration tests, it is essential that the fluid and contaminants used produce test data that can be related to the system requirements.

2.4.2 Filter Case

The filter case functions as the structural supporting member of the filter element. Space available, serviceability, pressure drop, fluid compatibility, operating temperature, and operating pressure – all enter into the filter case design. The size and shape of the case influence the filter capacity and service life. The minimization of flow restrictions and abrupt changes in flow direction is an important parameter. The critical area – the smallest cross-sectional area of the flow passage – must be identified by the designer. All other flow passages naturally will have a greater area than the critical area. The flow transition from one chamber to another is made as smooth as practical.

In-line cases with axial-design cylindrical or conical filter elements offer the least resistance to flow and are lightest in weight. The case may be either permanently assembled by welding, or it may be capable of disassembly to facilitate cleaning or replacement of the element. Because filter elements are designed to have a service life commensurate with the life of the engine system, case designs that permit removal and replacement of the element without actually breaking into the fluid system are seldom used in spaceflight applications. Interface connections depend upon the particular philosophy in engine-system plumbing; they may be either threaded, welded, brazed-union, or flanged. Threaded connections (straight thread) are the least desirable type in that contamination is generated as the case is installed in the system; pipe thread connections are extremely undesirable and are rarely used. Welded connections provide a leaktight system, but are difficult to inspect; electron-beam welding or other high-quality welding techniques commonly are used to achieve leaktight connections. Welded filter assemblies are used wherever appropriate. Flanged connections result in the cleanest system assembly, but some external leakage may occur at the interface if the seal is inadequate.

An important consideration in filter-case design is to ensure that the filter element cannot be inadvertently left out. This can be accomplished by making the element an integral part of one end of the case or by making the installation such that omission of the filter element results in failure to pass a test designed to detect that omission. A less desirable approach is to ensure by visual inspection that the filter element has been installed.

3. DESIGN CRITERIA and Recommended Practices

3.1 LINE ASSEMBLY

3.1.1 Routing

3.1.1.1 CENTERLINE GEOMETRY

3.1.1.1.1 Flexible Lines

The centerline geometry of the line shall minimize the number of flexible joints required to accommodate the design deflections.

Until the development of the SSME, no analytical method existed for achieving this objective; however, reference 140 presents a method for a three-joint, one-plane system that gives some insight into the subject. Generally in a multi-leg duct design, one flex joint is located in each leg for maximum flexibility. See figure 5 for the typical wraparound, gimbal-plane configuration.

Kinematic design layouts locating the flex joints in their optimum positions are recommended as the best technique for providing initial inputs into a space-frame program like that used on the SSME; the program then can be used to "fine-tune" the locations. Once the joints are located, verify that the loads they create are within the capabilities of the attached engine or vehicle structure.

3.1.1.1.2 Hard Lines

The centerline geometry of the line shall be such that the line does not impose excessive strain on attached structure.

Prior to laying out the centerline geometry of a new hardline design, consider the structural limitations of the components to which the line will be attached. Line installation tolerances in the engine or vehicle and the loads they would cause the hard line to induce on the attaching components must be evaluated. The routing of the line then should be such as to provide required flexibility without producing excessive strains on the attaching structure. Figure 4 depicts some typical routings that provide installation flexibility. Utilize the modulus of elasticity and wall thickness of the hardline material to assist in providing necessary flexibility.

Whenever practical, adjacent lines should be routed in banks or rows to achieve mutual support in bracketing and add to aesthetic appeal.

3.1.1.2 DEFLECTION LIMITATION

The routing of ducts shall preclude the requirement for a given flexible joint to absorb more than one type of deflection (angular, axial, or shear) simultaneously.

Locate the flexible joints in the duct so as to make maximum use of their deflection capability. Figure 5 gives a practical example for achieving this condition. Locate the centers of the bellows on the gimbal axes, so that the bellows are required only to angulate to accommodate gimballing; if they are located off-center, both shear and angulation will be applied.

3.1.1.2.1 Maximum Bellows Joint Deflection

The sum of all deflections simultaneously imposed on a bellows joint shall not cause bending stresses in excess of those commensurate with the design fatigue life.

See reference 82 for S-N data on typical bellows materials. When a bellows is subjected to more than one mode of deflection (e.g., angulation and shear) simultaneously, the sum of the stresses created by all modes must not exceed the total allowable.

If a particular bellows is subjected to combined motions and attendant stresses that do not permit the part to meet the design fatigue life, the motions must be reduced or additional bellows introduced into the system to share the deflections. For example, the duct shown in figure 8 was required to accommodate a torsional deflection that caused excessive strains in the axial bellows. To resolve the problem, a special bellows was inserted between the two axial bellows to absorb only the torsional deflections and thus relieve the torsional strain imposed on the other bellows.

All the different deflection modes that a bellows can absorb result in the application of the same type of bending stresses in the crowns and roots of the convolutions. Specific information on stress calculations corresponding to different types of deflections is presented in references 141 through 151. Methods given in reference 16 may be used to convert the various modes into axial equivalents for use in stress calculations.

3.1.1.3 TORSIONAL DEFLECTION

Duct assembly design shall preclude or minimize torsional loading on the ducts.

Use attachment flanges with slotted bolt holes in rocket engine applications where a duct must be closely connected, and therefore short in length, between two end mounts that are rotationally misaligned as a result of manufacturing tolerances and operational deflections. The slotted holes in the attachment flanges can in some instances accommodate the manufacturing misalignment. This is, however, not a desirable method of absorbing operational deflections, because a dynamic fluid seal is required.

Dog-leg ducts with angulating joints in each leg can absorb torsional deflections by angular deflection. A recommended method for making a short, straight duct torsionally flexible is to incorporate a bellows limited to torsional deflection and having a low spring rate. Such a bellows can be designed to allow a few degrees of motion within the allowable stresses of the material. A typical example is shown in figure 8.

3.1.2 Sizing

3.1.2.1 FLOW AREA

The duct flow area shall be consistent with the system pressure loss allowable.

Calculate the line pressure drop, utilizing data from tests on the individual components or on close-coupled components if typical of the duct design. Select the line inner diameter to provide the flow area required. Confirm the calculations by pressure-drop/flow tests on mockup or prototype hardware as soon as the final system configuration is achieved. In establishing flow velocity, take into consideration the water-hammer effects of sudden stoppage of the fluid.

3.1.2.2 WALL THICKNESS

The duct, while being of minimum weight, shall withstand all predicted loads.

Make a careful stress analysis of all duct elements in the design phase. The methods described in references 12 and 16 are recommended. Include consideration of the loads induced by flow effects and mechanical vibration.

To verify the analysis, the ducts should be subjected to adequate proof-pressure tests and external loading and vibration tests as early in the prototype phase as possible.

Recommended test practices are as follows:

(1) Subject the duct to proof and burst testing. Stress levels of 1.2 to 1.5 times maximum expected operating levels for proof testing and 2.5 times maximum operating levels for burst testing are recommended (table IV).

(2) When thermal-gradient tests or tests at operating temperature are inconvenient, simulate the reduction in material strength at elevated temperature by additional pressure or loads at ambient temperature.

(3) Perform vibration testing of the duct with the operating fluid flowing at maximum operating condition whenever possible; this test condition will account for fluid mass and possible coupling of flow-induced and mechanical vibration.

(4) Perform system start- and shutdown-transient flow tests to simulate fluid pressure surges (water-hammer effects).

3.1.3 Control of Pressure Drop

3.1.3.1 FLOW-DIRECTION CHANGE

The configuration for a change in flow direction (e.g., a bend or tapoff branch) shall minimize the pressure loss that accompanies a direction change.

Where routing permits, use bends with R/D values that produce the lowest loss coefficients. An example of a graph for determining pressure loss in a circular 90° elbow is shown in figure 47 and discussed in reference 8. Friction loss in bend length must be added to bend loss to obtain total loss.

Flow guide vanes (fig. 9) are recommended for reducing losses in short-radius bends. The selection of the final configuration requires a tradeoff between flow efficiency and producibility. Analyze vane flutter to ensure that no vibration during operation is induced by system fluid oscillations.

Pressure loss in tapoff branches can be minimized by aligning the tapoffs toward oncoming flow (fig. 48(a)) rather than at a right angle to the flow (fig. 48(b)). The low-loss configuration results in a structurally less desirable joint, the effects of which must be traded off against the pressure gain.

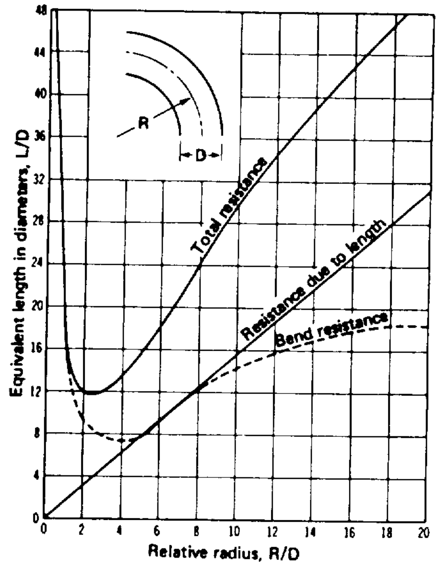


Figure 47. – Graph for determining pressure loss in a circular 90° elbow.

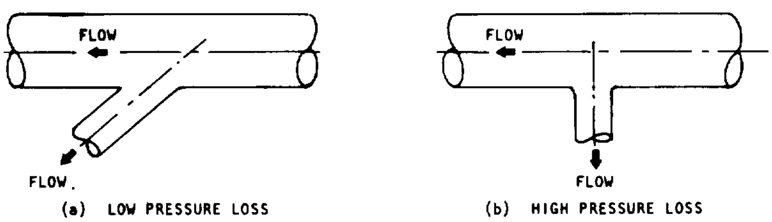


Figure 48. – Pressure loss in a tapoff branch related to configuration of the branch.

3.1.3.2 FLOW-AREA CHANGE

The configuration for a change in flow area shall minimize system pressure drop that accompanies an area change.

Use smooth, gradual area transitions, particularly in entrances to lines or in contracting sections where rounding of intersecting edges can improve flow efficiency. Expansions should have a gradual taper (total included angle of 10°); pressure recovery depends on the length of divergence available, but even short lengths of divergent section are beneficial. Consult reference 22 for some guidelines in diffuser design.

3.1.3.3 FLOW DISTRIBUTION

Propellant flow into and out of a system component shall be evenly distributed.

When engine-vehicle interfaces do not permit a straight-in approach for the inlets, direct the flow into the component through vaned elbows (fig. 10). Velocity profile downstream of the elbows is kept flat through the turning vanes, so that the desired inlet condition for best pump performance is achieved. Other devices that should be used when necessary to improve flow distribution include the egg-crate straightener (fig. 11) and the flow splitter (fig. 12).

Potential problems with thermal-expansion loads due to flow straighteners can be minimized by design. The configuration in figure 49 is recommended for minimizing thermal-expansion loads to the wall of the duct.

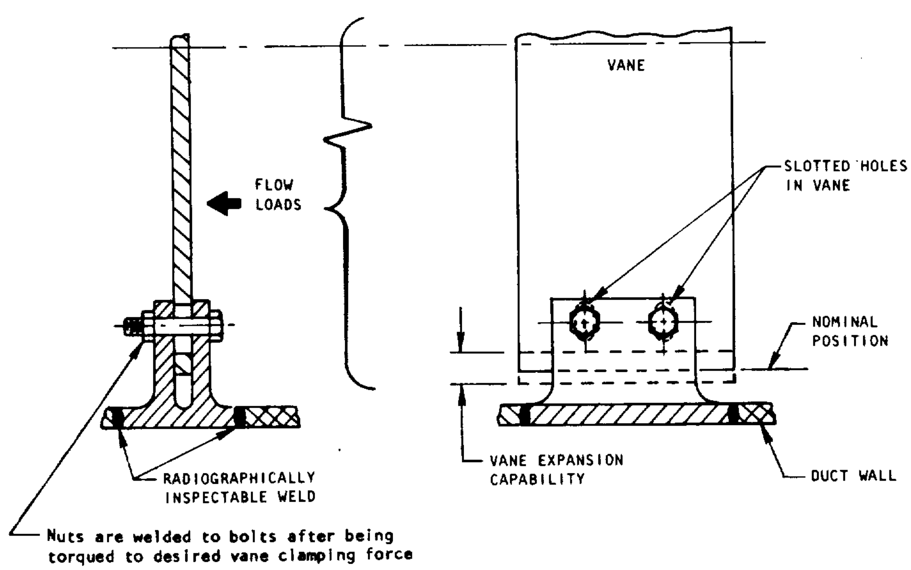


Figure 49. – Recommended flow-straightener design for relieving thermal-expansion loads on a duct wall.

3.1.3.4 FLOW RESISTANCE

The duct inner surface shall offer minimum resistance to flow.

Grind or polish rough wall surfaces of castings and remove weld protrusions where a reasonable or desirable improvement in flow efficiency will result. A tradeoff of the cost of reducing flow losses versus increased efficiency of the engine should be made for high-performance systems.

Hydrodynamicists should be consulted in the design phase of internal-tie linkages for advice on streamlining the structure so as to minimize energy losses. In the case of low-pressure ducts (e.g., the inlets of pumps) where the vapor pressure of the fluid may be near the static pressure, the hydrodynamicist should also analyze the design for possible cavitation that could affect the pump performance or cause mechanical damage with cavitation-bubble collapse.

3.1.4 Control of Pump-Inlet-Line Vibration

The dynamic response characteristics of the engine pump inlet lines shall not contribute to a regenerative feedback interaction between the propulsion system and vehicle structure.

The interaction of the vibrational characteristics of a liquid rocket engine and the vehicle in which it is installed should be analyzed in the design phase for possible Pogo effects. Utilize the technology developed on this subject (refs. 152 through 155) in conducting a dynamics analysis of the excitation and response interrelationship of engine and vehicle. Modify the inlet-line configuration as necessary to preclude or damp instabilities; examples of successful designs are shown in figures 13, 14, and 15.

3.1.5 Components

3.1.5.1 SEPARABLE CONNECTORS

The flanged joints of the ducts shall not be subject to static-seal leakage under operation conditions.

Provide seal integrity of the flanged joint by designing the flange to be rigid enough to prevent seal-unseating rotation under operational pressures and loads. The bolt-circle diameter should be as small as practicable; utilize small bolts closely spaced rather than a few large ones. A recommended design for static-seal flanged joint is shown in figure 50. Consult reference 29 for additional information on the design of static-seal joints.

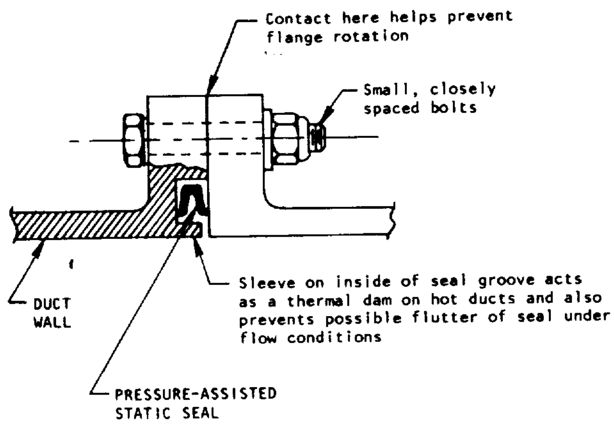


Figure 50. — Recommended design for a flanged joint.

3.1.5.2 MANIFOLDS

3.1.5.2.1 Flow Distribution

The manifold shall distribute flow evenly with minimum energy loss.

Recommended practice is to design the cross section of the flow passage in relation to the desired distribution of flow using contours that produce a minimum of pressure energy loss in the working fluid. Use the empirical and analytical data in references 30 through 32 to assist in these calculations.

3.1.5.2.2 Structural Adequacy

The manifold shall be structurally adequate for the mechanical and dynamic loads applied during operation.

Because of the intricacies of their flow passages, manifolds often do not permit application of the best structural design and fabrication techniques. The classic pressure-carrying shapes such as spheres or cylinders many times must be compromised into partial spheres or cylinders intersecting with other shapes in order to achieve the fluid-flow design goal of the manifold. When necessary, reinforce the pressure-carrying chambers with internal tension-tie splitters or external ribs to make them structurally acceptable.

The complexity of the intersecting structure at branch points can be analyzed by the finite-element or NASTRAN computer methods to determine stresses or deflections. An alternate method is to combine simplified analyses and experimental results. In either case, any new design should be subject to design-verification testing.

3.1.5.2.3 Cleanability

The manifold design shall allow for cleaning and cleanliness inspection.

Design the manifold such that all inside surfaces (1) can be reached with deburring tools and cleaning materials and (2) are accessible for visual inspection to verify cleanliness. Use a combination of lights, dental mirrors, and fibre-optic instruments for inspection when necessary. Avoid angled passageways and blind holes in design.

Pockets where fluids could be entrapped should drain freely by gravity, although it may be necessary to rotate the part and stop in several attitudes to assure complete drainage. Use halogen sniff detectors to determine the presence of any residual cleaning fluids containing halogens as a final acceptance test.

3.1.5.3 BRACKETS, BOSSES, AND MOUNTING LUGS

Brackets, bosses, and mounting lugs attached to the duct wall shall withstand service loads without cracking or distorting.

Design the attachment to provide a smooth wall-thickness transition from the duct to the attachment. Avoid fillet-welded or resistance-welded sandwich or doubler-type attachments, including identification attachments. Use butt-welded construction where possible to (1) avoid stress concentrations and traps for cleaning residuals and (2) allow complete radiographic inspection capability. Figure 51 illustrates both recommended and unacceptable design practices.

Weld-on attachments should be located on line assemblies in areas that are free of applied or induced stresses due to pressure loads, flex-joint-linkage reactions, vibratory loads, or thermal strains.

3.1.5.4 INSULATION

3.1.5.4.1 Thermal Resistivity

Insulation shall provide an adequate barrier to the flow of heat into lines carrying cryogenic fluids.

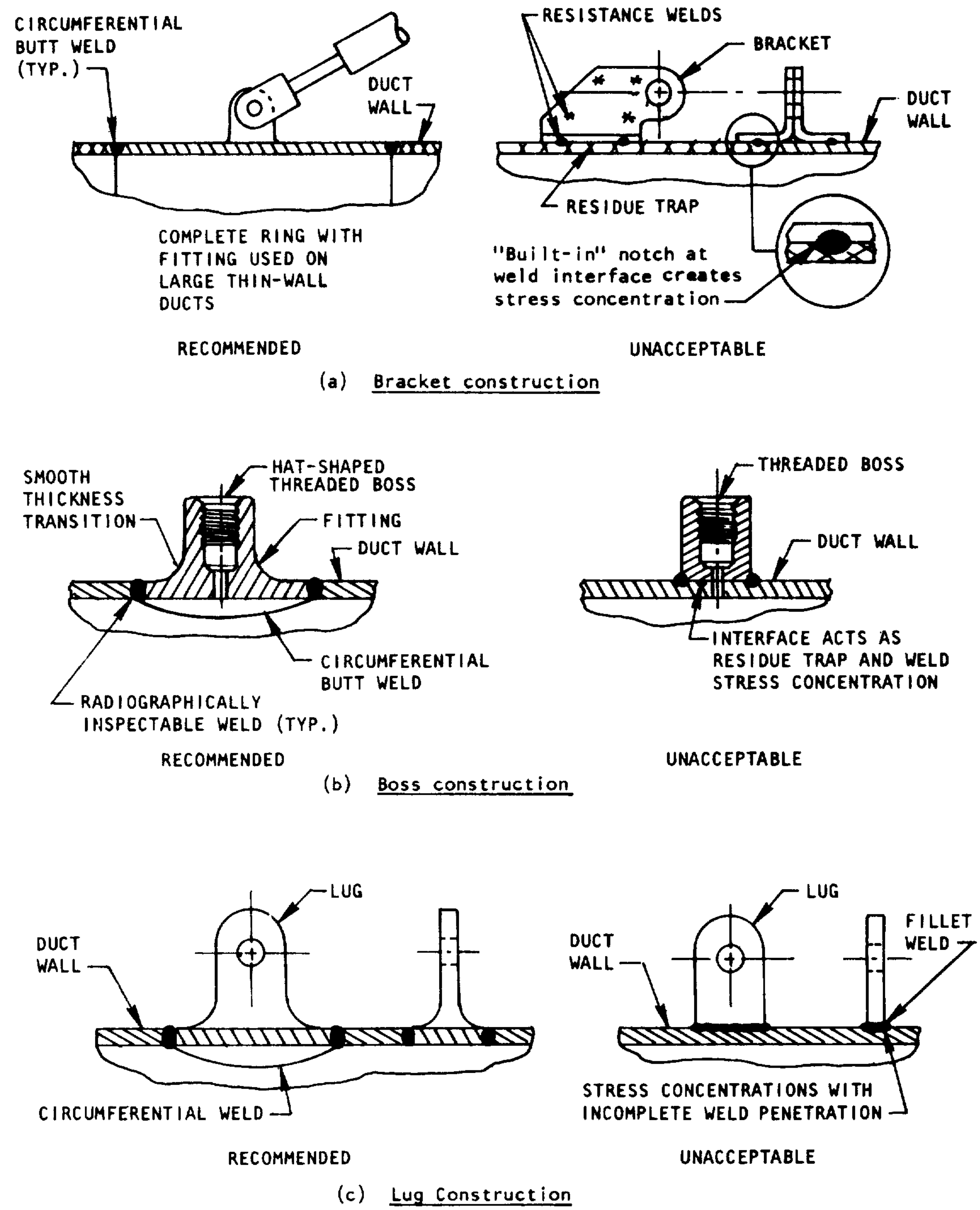


Figure 51. – Recommended and unacceptable designs for bracket, boss, and mounting-lug attachments.

Double-wall vacuum jacketing, either pumped down or cryopumped, is recommended for engine and vehicle use. Good examples of designs for standoffs, evacuation valves, and burst disks in vacuum lines are presented in reference 36.

3.1.5.4.2 Leak Tightness of Vacuum Jackets

Vacuum jackets shall not leak beyond acceptable limits.

Use all-welded construction to provide the most nearly leaktight jacket. Leak test all welds simultaneously through a single port with a mass spectrometer leak detector prior to closing the port with a radiographically inspectable weld (fig. 52).

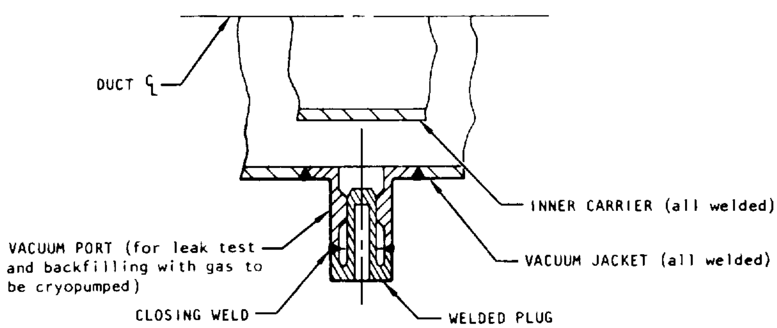


Figure 52. – Recommended construction of vacuum-jacketed closeoff port.

3.1.5.5 HOT-TO-COLD DUCT INTERSECTION

Manifolds carrying cryogenic fluids through walls of ducts carrying high-velocity hot gases shall withstand thermal loads without leaking.

Recommended practice, as shown in figure 53, is to provide a long conduction path to the cold member through a stagnant hot-gas pocket inside the flared portion of the duct wall; this design also gives flexibility between the hot and cold members and the flared duct wall in order to distribute the differential thermal motions of the members. All welds should be inspectable radiographically.

(The particular design shown was used on the oxidizer heat exchanger of the J-2 engine, which vaporizes LOX for use in pressurizing the main oxidizer propellant tank of the vehicle. The heating fluid is turbine exhaust gas.)

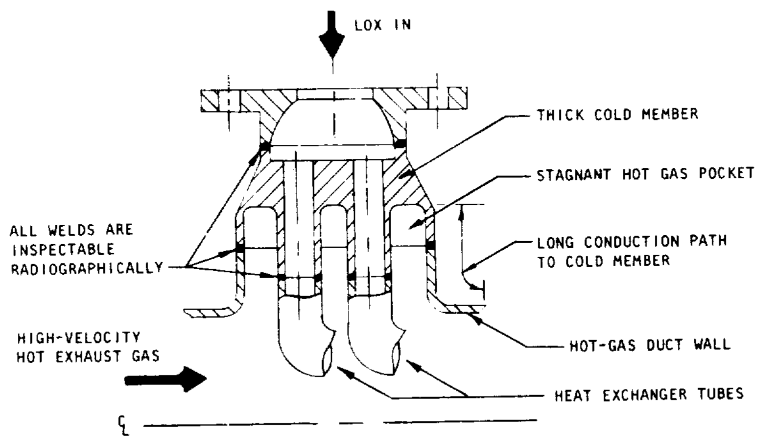


Figure 53. – Recommended design for hot-to-cold intersection of a manifold and duct.

3.1.5.6 ELBOWS

Cyclically stressed elbows shall not be subject to in-service fatigue cracking.

Because of the sometimes unique geometries, thin walls with respect to diameters, and tight bend radii in aerospace ducting elbows, large concentrations of bending stress can develop. Reference 37 is recommended for guidelines useful to designers and stress analysts in giving proper consideration to stress intensifications and thereby precluding fatigue cracking.

3.1.5.7 HANDLING-PROTECTION DEVICES

3.1.5.7.1 Critical Surfaces

A handling-protection device shall protect critical surfaces of the duct assembly (e.g., static seal interfaces and bellows convolutions) against handling damage during all fabrication and shipping operations.

The covers should provide protection against damage and also prevent contamination from entering the duct. The covers themselves should not be a source of contamination. Half-shell bellows covers as depicted in figure 19 are recommended.

3.1.5.7.2 Overdeflection

A handling-protection device shall protect against bellows damage due to overdeflection during all fabrication and handling operations.

Use strongback fixtures to prevent overangulation (and attendant fatigue damage) on low-spring-rate bellows joints during handling and shipment. These fixtures clamp around the flexible joint and prevent movement. A recommended type of strongback fixture is shown in figure 19.

3.1.5.7.3 Removal Safeguards

Safeguards on the handling-protection devices shall ensure removal of the devices before flight.

Design the component or subcomponent protection device so that the part cannot be installed or assembled into the next assembly without removal of the device. In final installations or assemblies, as on a vehicle ready for launch, protective devices still required at that point should have adequate red warning flags and checkout instructions to ensure removal before flight.

3.1.6 Materials

3.1.6.1 CHEMICAL COMPATIBILITY

The materials selected for a line shall be compatible with the system fluid and the external environment under all conditions of storage and service life.

Consider the following in evaluating line material/fluid compatibility:

- Effect of material on fluid decomposition rate
- Susceptibility of material to impact ignition in presence of oxidizers
- Corrosion rate of the material in the fluid
- Consumption of material by chemical reaction with the fluid
- Crack initiation and crack growth in the material as influenced by the fluid.

The effect of all cleaning solvents or processing fluids used in fabrication sequence (e.g., corrosion, surface residuals) should be reviewed and evaluated.

Prepare a list of suitable alloys for the fluid system in which the line is to be used. Use the best available fluid-compatibility data (ref. 156) on the many possible alloys, together with the list of recommended alloys for typical fluid systems presented in table VI. Coating and plating for protection and compatibility are impractical and are not recommended.

Do not use the following alloys in the standard heat-treat conditions cited, because of material susceptibility to stress corrosion: 2014 T4, T6; 2024 T3, T4, T351; PH 15-7Mo RH950, TH1050; 17-7PH Ann, H900; 17-4PH RH950, TH1050; AM350 SCT850; AM355 SCT850.

Corrosion-resistant steels such as 321 CRES are susceptible to pitting corrosion if the metal surface becomes dirty (passivity locally impaired). Ensure that all processing results in a clean part.

Verify that the line material is not sensitive to ignition by impact in the presence of oxidizer propellants. A case in point is titanium, which should not be used in oxygen systems because impact will cause explosive oxidation. The procedure outlined in reference 157 should be used as an acceptance-test criterion for any material under consideration.

The line material should not be susceptible to embrittlement or enhanced crack initiation and growth by environmental fluids. For example, titanium alloys should not be used in hydrogen systems operating above $-100°F$. Consult references 158 and 159 for reduction of tensile properties of nickel-base alloys in hydrogen. See section 3.1.6.1.1 below.

The line material should not catalytically decompose the fluid being transported. For example, do not use molybdenum, iron, copper, or silver with the hydrazine propellants (N_2H_4, A-50, or MMH), because violent decomposition of the propellant can result. Do not use alloys of these elements without verifying safety through compatibility-test data.

The line material should be compatible with propellants involved and their decomposition products. By way of illustration, moisture in a N_2O_4 system can possibly form nitric acid. Alloys for this system therefore must be resistant to nitric acid; stainless steel and aluminum alloys therefore are suitable for use with N_2O_4.

All materials should be of sufficient thickness to resist complete penetration by corrosion. Although thinner materials are in use, thicknesses under 0.010 in. for convoluted details should be avoided. Use nickel-base alloys in those applications where design requirements dictate wall thicknessess under 0.010 in.

3.1.6.1.1 Hydrogen-Environment Embrittlement

Ducting and filter materials, when necessary, shall withstand the harmful effects of a high-pressure gaseous-hydrogen environment.

Table VI. – Recommended Alloys for Lines, Bellows, and Braided Flexible Hose in Use With Typical Service Fluids

Fluid	Temperature range[a], °F	Maximum pressure[a], psi	Recommended alloys[b]	
			Lines only	All components
LH_2	-423	1200	6061	Ti-40A, 321, 347, "K" Monel, In X-750, In 600, 21-6-9, Ti-5Al-2.5Sn, Hastelloy C, In 625, In 718
GH_2	-423 to +160 -423 to +100	1000 1500	6061 –	321, 347, 21-6-9 In 600, Hastelloy C, In 625, In 718
RP-1, RJ-1	-40 to +160	2000	6061	321, 347, Monel, In X-750, A 286, In 600, In 625, In 718
LOX	-297 to +160	2000	6061	321, 347, "K" Monel, In X-750, In 600, 21-6-9, Hastelloy C, In 625, In 718
He, N_2, Hydraulic oil[c]	-50 to +160	3000	6061	Ti-40A, 321, 347, In 600, 21-6-9, Hastelloy C, Ti-6Al-4V, In 718
N_2O_4	+32 to +160	1000	–	321, 347, In 600, 21-6-9, Hastelloy C, In 625, In 718
N_2H_4, A-50, MMH	+32 to +160	1000	6061	Ti-40A, 321, 347, In 600, 21-6-9, Hastelloy C, In 625, Ti-6Al-4V, In 718
Combustion products: RP and O_2, H_2 and O_2	65 to 1200	1000	–	321, 347, In 600, Hastelloy C, In 625, In 718
A-50, MMH, N_2H_4, and N_2O_4	65 to 1600	1000	–	In 600, Hastelloy C, In 625

[a]Represents successful experience, but materials are not necessarily limited to these conditions.

[b]Alloys are listed in order of increasing ultimate tensile strength at 80°F.
See Appendix B for chemical composition. In = Inconel. 21-6-9 = Armco 21-6-9.

[c]Meets specifications in MIL-H-5606C, Amend. 1, Nov. 19, 1973.

Any material considered for ducting or filter use with hydrogen should be evaluated for susceptibility to hydrogen embrittlement by performing specimen tests in the appropriate environment (refs. 39, 40, 41, and 160). Smooth and notched tensile tests, smooth strain-controlled low-cycle fatigue tests, and sustained-load and low-cycle fatigue tests with precracked fracture-mechanics specimens – all should be accomplished if a material is expected to show any significant degree of hydrogen embrittlement (ref. 161).

If it becomes necessary to use a material sensitive to hydrogen embrittlement, the strain level should be kept below an acceptable limit or the surface exposed to hydrogen should be protected with a liner material or plating impervious to hydrogen.

3.1.6.2 PHYSICAL AND MECHANICAL PROPERTIES

The physical and mechanical properties of the line material shall be suitable for the functional requirements of the line and conditions of use.

Select the alloy that will meet the material requirements and will produce the lowest cost line. For the commonly used alloys, use data on physical and mechanical properties, forming limits, and relative line cost as shown in table III.

3.1.6.3 FORMABILITY

The formability of the line material shall permit bending the line to the required path or stretching to the required convolution forms.

Use only high-ductility alloys in convoluted details. Recommended ductile materials are 321 CRES, Hastelloy C, and Inconel 718, which are also excellent materials for tubular elements of line assemblies. Heat-treatable alloys must be formed in an annealed condition.

3.1.6.4 WELDABILITY

Weldability of line material shall be adequate to permit bellows, end flanges, brackets, and bosses to be attached to the tubing by welding.

If a bellows is to be attached to the line by welding, use similar materials in the bellows-to-line joint; i.e., titanium-alloy bellows with a titanium-alloy line, and a nickel-base or steel-alloy bellows with a nickel-base or steel-alloy line.

Materials can be dissimilar but must be sufficiently compatible with flanges, brackets, and bosses to provide weld integrity and required strength. Consult reference 162 for information pertinent to weldability of commonly used dissimilar alloys and recommended filler materials.

3.1.6.5 LUBRICANTS

Lubricants used on line, bellows, or hose assemblies shall not impair the assembly's corrosion resistance.

Use of molybdenum disulfide dry-film lubricant coatings on bearing surfaces of nickel-base-alloy assemblies is recommended; however, care should be exercised in use of this lubricant with corrosion-resistant steels. Corrosion test data such as those obtained in salt-spray testing should be reviewed for the particular metal/lubricant combination under consideration before specifying a lubricant.

3.1.7 Testing

3.1.7.1 TEST REQUIREMENTS

Test requirements for lines, bellows, and hose shall reflect the effect of all operational loads on the functional requirements of the part over its intended service life.

Operational loads considered should include those created by pressure, temperature, flow, vibration, mechanical application by other attaching components, or combinations thereof. Table VII lists test requirements, their applicability, and recommended practice.

When practical and economical, individual tests may be combined and performed simultaneously to simulate more closely actual engine or vehicle operating conditions. For example, tests on mechanical vibration and flow-induced vibration could be done at the same time.

3.1.7.1.1 Qualification Testing

Qualification testing shall verify structural integrity, operability, and functionality of a new line design under all conditions to which the part will be subjected during its service life.

Recommended practices for qualification testing are summarized in table VII. Specific aspects of particular importance are treated in the subsections below.

Table VII. – Recommended Practices for Component Tests

Type of test	Qualification requirement	Sampling plan requirement	Acceptance requirement	Recommended practice
Proof pressure	Yes	Yes	Yes	Apply hydrostatic (water) pressure to the part for five cycles of 2 minutes duration each. Pressure level normally is 1.2 x operating pressure including transients or 1.5 x operating pressure, whichever is greater. No structural failure or permanent distortion allowed.
Leak test	Yes	Yes	Yes	Apply operating pressure to part; use helium for ducts to be used with low-density fluid and gaseous nitrogen for those used with high-density fluid or liquid; pressure held for one cycle of 5 minutes. With pressurized assembly submerged in water, no leakage permitted as evidenced by lack of bubble formation. Use helium mass spectrometer test on vacuum-jacketed lines.
Examination of product	Yes	Yes	Yes	Check part for conformance to drawing requirements including dimensional and handling-damage inspection, verification of material and processing specifications, protective covers and packaging.
Movement verification	Yes	Yes, if considered critical	Yes if, considered critical	With one end of part fixed, traverse opposite end through design deflection range (angular, axial, lateral, torsional, or combinations thereof) to ensure no binding or mismade parts.
Spring rate (force/deflection)	Yes	Yes	Yes, if considered critical	With one end of part fixed, deflect opposite end in incremental steps through design range and check load or moment. Compare results with design requirement. May be done in either an unpressurized or pressurized condition.
Buckling stability	Yes	Yes, if considered critical	Yes, if considered critical	With bellows or duct fixed with rigidity that simulates engine or vehicle installation and with maximum operational deflection, apply proof pressure. No column buckling permitted.
Flexural endurance	Yes	Yes	No	With one end of bellows, duct, or hose fixed, deflect the opposite end through design range. Cycle rate should simulate end usage rate and total cycle life should be four times predicted service life of part without fatigue failure. Loads required to deflect should not exceed allowables during this period.

(continued)

Table VII. – Recommended Practices for Component Tests (concluded)

Type of Test	Qualification requirement	Sampling plan requirement	Acceptance requirement	Recommended practice
Mechanical vibration	Yes	Yes	No	With bellows, duct, or hose mounted in a manner simulating engine or vehicle installation, vibrate the part mechanically through an acceleration and frequency spectrum simulating that of service usage with pressure and temperature controlled to operational conditions. Part must withstand a predetermined period of testing without failure.
Flow calibration	Yes, if considered critical	No	No	With bellows, duct, or hose mounted in a manner simulating engine or vehicle installation, flow end-use fluid through the part at operational pressure, temperature, and flowrates. Measure pressure drop and compare with calculated values.
Flow fatigue	Yes, if considered critical	No	No	The need for this test may be eliminated by analysis per the methods outlined in references 90, 91, and 94. However, if analysis indicates a marginal condition, subject the part to flow-vibration-resonance search over the operational fluid flowrate range. After resonances have been identified, subject the part to endurance testing equal to engine or vehicle specified life evenly distributed over the three highest "g"-force resonances. No failure allowed.
Pressure impulse	Yes	Yes	No	Subject the part to pressure impulses simulating the pressure-vs-time waveform encountered in engine or vehicle operation for a frequency-times-duration life as specified in end-use application. End-use fluid is recommended for test use. No failure permitted.
Sectioning and thickness measurement	Yes	Yes	No	After completion of the test program, section the part and examine it for conformity to drawing, material, heat-treat, and dimensional requirements. Values are recorded for spring-rate and fatigue data correlation.
Burst	Yes	Yes	No	Apply hydrostatic pressure to part at 1.5 to 2.5 x maximum operating pressure (value depends on contractual requirements). Part may deform plastically but not leak at burst pressure value. Good practice is to increase pressure to actual rupture condition in order to determine design margin.

3.1.7.1.1.1 Hydrostatic Proof Test

The hydrostatic proof-pressure test shall verify the pressure-carrying capability of a line, bellows, or hose; the test fluid shall not promote corrosion of material.

Proof-pressure test certification should be performed on every piece in a production lot. Distilled or de-ionized water is recommended as a noncorrosive test fluid, since there is evidence that tap water can promote corrosion. The part should be thoroughly dried after exposure to water and prior to the gaseous leak test, since water can seal up tiny pin holes or crevices by surface-tension effect. After proof-pressure test, inspect all parts for adherence to drawing dimensional requirements. Each bellows should be examined for evidence of deformation that might reduce the critical spacing between convolutions. For hardware that operates at elevated temperature, the acceptance proof-pressure test can be performed at room temperature with the test pressure level increased to compensate for the change in material properties at the elevated temperature. This practice is simpler and less expensive than performing the test at elevated temperature.

In the case of alloys such as 321 CRES, whose strength and modulus vary differently with temperature, care should be taken not to specify a room-temperature test at a pressure that could be in excess of the bellows critical buckling pressure. Since, for this particular type of alloy, elastic modulus decreases at a rate greater than the rate of decrease in yield or ultimate strength, the part could buckle, because buckling pressure is a direct function of modulus.

3.1.7.1.1.2 Leak Tests

Leak tests shall accurately measure leakage rates and shall not affect line, bellows, or hose material.

The gas-pressurized, submerged-in-water test is recommended as the most economical, practical, and accurate leak-test method. For safety, the test should be conducted in a location remote from personnel (ref. 68).

A helium mass-spectrometer leak detector test is recommended for verifying leak tightness in vacuum-jacketed lines.

The use of soap or detergent solution as an alternate to either the immersion or MSLD tests is not recommended because these materials may cause stress corrosion in many aerospace alloys. A solution described in reference 67 has been found acceptable and is recommended.

3.1.7.1.1.3 End-Use Simulation

The conditions for qualification testing of lines, bellows, or flexible hoses shall simulate end-use conditions as closely as possible.

Within budget and practical limitations, the qualification test setup should include all of the end-use influences. All next-assembly or installation attaching hardware that could affect the test component should be included.

When the component must operate in a vacuum environment, the tests should be performed under space-simulated vacuum conditions. In particular, the flow-fatigue testing of cryogenic lines should be performed under vacuum conditions to preclude repetition of the Saturn AS 502 flex-line failure (ref. 163).

3.1.7.1.1.3.1 Flow-Vibration Testing

Flow testing of lines, bellows, or flex hoses shall determine their response to flow-induced vibration and ability to withstand the attendant loads under worst-case operational conditions.

Because of the complexity of the phenomenon of flow-induced vibration, conduct all flow-resonance search-and-dwell tests (refs. 90 and 99) with the actual end-use fluid at operational flowrate, temperature, and pressure conditions and in the operational external environment. In cases where the line or hose may be exposed to more than one fluid in its operational lifetime, test with each of these fluids.

3.1.7.1.2 Sample Testing

The sampling test program shall verify consistency of quality of production units.

The frequency of sampling should be based on a quality-assurance statistical analysis, although the relatively small number of parts in a typical aerospace production contract may require that the requirements of statistical-analysis theory be tempered with engineering judgement.

3.1.7.1.2.1 Fatigue Test at Temperature

Bellows or flexible-hose flexural endurance tests shall provide an accurate indication of actual fatigue life at operating temperatures.

As an economy measure, comparative flexural endurance tests can be performed on pre-production bellows or hose at room temperature and operating temperature to form a basis for expected fatigue life. Production-lot sample tests then can be performed under the less complicated room-temperature conditions. Keep in mind, of course, that fatigue life is reduced at elevated temperature and increased at low temperature.

3.1.7.1.3 Acceptance Testing

Acceptance testing of every production part shall verify conformance to the basic requirements of the engineering procurement document.

The implementation of acceptance-test requirements should ensure that the program is carried out in the most efficient manner. For example, hardware made by a subcontractor should be acceptance tested at the source; tests should not be repeated after receipt of the hardware unless obvious handling damage has occurred in shipping. Records should be kept of all tests performed as proof of their completion and for possible analysis later in the event of a production or service problem.

3.1.7.1.3.1 Flow-Passage Clearance

For line assemblies in which pressure drop is critical, tests shall verify the line free-flow area on each assembly.

A recommended excellent, low-cost test for verifying line flow area is the ball test, in which a metal ball with an OD slightly smaller than the ID of the line at maximum permissible weld dropthrough is required to pass completely through a line assembly.

3.1.7.2 TEST INSTRUMENTATION

3.1.7.2.1 Standard Practices

Pressure, temperature, and flowrate measurements shall be of acceptable accuracy.

The standard practices outlined by the Interagency Chemical Rocket Propulsion Group in reference 69 are recommended.

3.1.7.2.2 Special Vibration Instrumentation

Test instrumentation shall accurately measure response of a line, bellows, or hose assembly to flow-induced vibration in flow tests.

A recommended practice for instrumenting braided flexible hoses is to cement miniature accelerometers to the braid ring collar of each flexible section, as shown in figure 54. One part Epibond 121 cement combined with two parts Lithafrax filler is capable of withstanding thermal effects of liquid hydrogen (−423°F) and the effects of high-frequency, high-g accelerations. Mount one accelerometer to measure lateral motion and the other, perpendicular to it, to measure axial motion. Vibration measurements made simultaneously on the same flexible section with both strain gages on convolutions and collar-mounted accelerometers indicate good correlation.

For free (not covered with braid) bellows, strain gages measuring bending strains in the convolution crowns are recommended as vibration detectors.

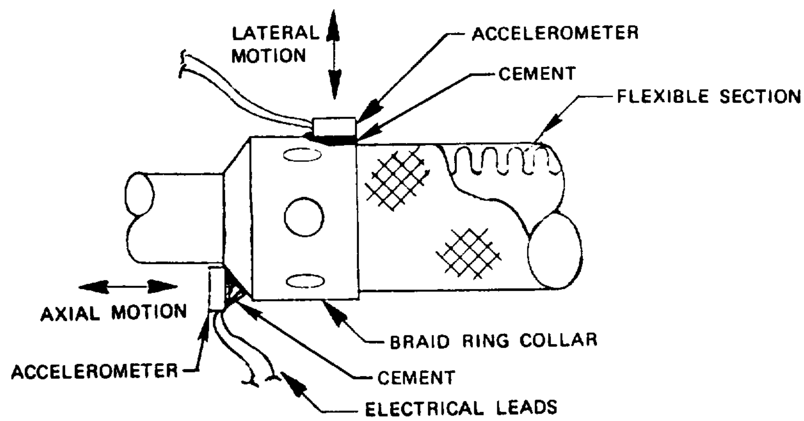


Figure 54. – Recommended technique for mounting vibration-measurement instrumentation on braided flexible hose.

3.2 BELLOWS JOINT

3.2.1 Bellows

3.2.1.1 PRESSURE CAPABILITY

The bellows shall withstand the design proof pressure without permanent distortion that could affect its ability to meet other performance requirements and shall withstand burst pressure without leakage or rupture.

Use stress equations based on elastic properties, and correlate test data on that basis. The bellows should not yield under proof pressure to the point where changes in the convolution shape could affect spring rate and fatigue life. Design bellows in the plastic range when low spring rates and low weights are required within limited fatigue-life allowances. See reference 16 for equations defining bellows designs.

3.2.1.2 FATIGUE LIFE

The bellows fatigue life shall be adequate for its anticipated duty cycle plus an acceptable margin.

Estimate the deflection duty-cycle requirement of a new design, and limit working stresses to values that will produce the desired fatigue life as indicated from the S-N curve. Bellows fatigue data are available (in the form of S-N curves) for the most frequently used bellows materials (refs. 46 and 82). Miner's rule (ref. 164) can be used to sum the fatigue damage imposed by various deflections (end-point motions) expected in the duty cycle. If the design is still considered marginal after this analysis, prototype flexural-endurance testing should be performed. Materials with fatigue strengths higher than that of 321 CRES (e.g., the nickel-base alloys) offer some additional fatigue-life margin.

The above practices are applicable to the large deflections induced by thermal differential effects, thrust vector control gimballing, and installation misalignment, i.e., deflections producing what is generally known as low-cycle fatigue. The bellows deflections caused by mechanical or flow vibration are low-amplitude, high-frequency in nature, and the fatigue damage they create falls into the high-cycle fatigue category. The reduction in fatigue life caused by this high-frequency motion must be analyzed separately from that treated in the low-cycle analysis. It is desirable to keep all high-frequency deflection stresses below the endurance limit of the material. High-cycle-fatigue S-N curves for the materials most often used in bellows are available in reference 165.

3.2.1.2.1 Vibration Susceptibility

The bellows shall not be susceptible to harmful vibration.

Review the bellows design carefully to identify all possible modes and natural frequencies in which the bellows can vibrate. Analyze the operational environment to identify all possible sources of vibratory forces and their magnitudes and frequencies. To ensure that the bellows will not be overstressed, determine the response (deflections) of the bellows to these forces and compare them with those allowable.

References 20 and 83 through 91 contain information on techniques for calculating bellows natural frequency. Techniques for calculating flow-induced vibration excitation and bellows response are outlined step by step in reference 90.

3.2.1.2.2 Flow-Induced Vibration

Bellows stresses caused by flow-induced vibration forces shall be below the fatigue limit of the bellows material.

Size the bellows (bellows ID has the strongest influence) to ensure that flow-induced stresses do not exceed the minimum fatigue-limit stress for the material. The number of applied stress cycles in service cannot be predicted, because the time spent at each of the high frequencies involved is not predictable. Several hundred thousand cycles can be accumulated in only a few seconds of operation; therefore, flow-induced stresses should be kept below the endurance limit of the material. (Note: This condition is in contrast to end-point motion stresses, which are deliberately high to take advantage of predictable cyclic life.)

The effect of acoustic loading should also be taken into account; use the methods outlined in references 85 and 89.

3.2.1.2.3 Mechanically Induced Vibration

A bellows shall be capable of withstanding all mechanical vibration forces encountered in operation.

Design the bellows initially to avoid matching the bellows natural frequencies with known frequency inputs. The mechanical inputs from attaching structure are difficult to predict; nevertheless, on the basis of previous experience with similar hardware configurations and with use of the best-known analytical techniques, an estimate of these inputs should be made in the design phase. Accelerometer measurements then should be made early in the engine development program to fully map the vibration environment of the bellows for correlation with the design analysis. Some redesign and retesting may be indicated by the vibration measurements.

3.2.1.3 BUCKLING STABILITY

The bellows or flexible line assembly shall be stable in buckling.

Recommended practice is to apply proof pressure to the part while it is restrained in the maximum operational deflection and length position.

When space limitations require the use of the double-bellows spoolpiece configuration and bellows with live lengths that exceed their diameters, use external sleeves or scissors-like linkages to control buckling. External sleeves as shown in figure 55 can be used as an antibuckling stabilizer on a double-bellows spoolpiece arrangement; at least three tie bars are required. On a compression-type bellows, scissors-like linkages (figs. 6 and 20(b)) tying across the bellows can be used to stabilize the center ring. This design in effect reduces the effective buckling live length of the overall bellows length by half, since the center ring cannot move transverse to the duct centerline when end flanges are fixed. The duct can absorb axial, angular, offset, and torsional motions.

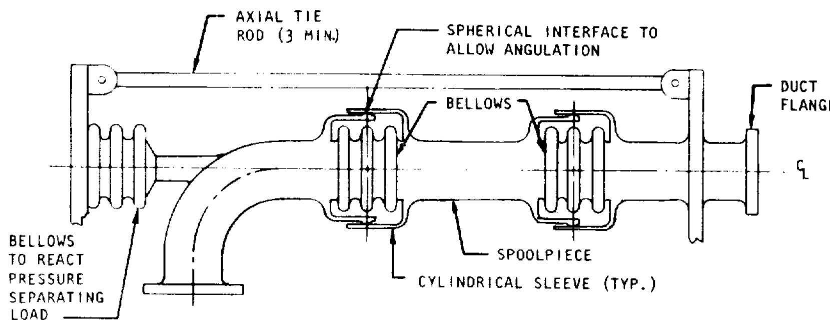


Figure 55. – Recommended vibration damper and antibuckling device for double-bellows spoolpiece arrangement.

3.2.1.4 CORROSION RESISTANCE

The bellows shall be resistant to all internal and external corrosive environments.

Recommended materials for bellows are nickel-base alloys such as Inconel 718, Inconel 625, and Hastelloy C. No corrosion problems have yet been encountered in bellows applications involving these materials. They are inherently much more corrosion resistant than 321 CRES and perform well in cryogenic and high-temperature service. Entrapment of cleaning fluid contaminants can cause corrosion; however, these fluids can be detected by utilizing a halogen sniff tester after completion of the cleaning cycle.

3.2.1.5 MANUFACTURING

The bellows fabrication shall involve only proved manufacturing techniques and forming limits.

Before forming the bellows, planish longitudinal welds in the tube to reduce the notch effect and improve the structure at the crowns by eliminating stress concentrations in the weld area. Tubes may be annealed after planishing to reduce the residual stresses induced by the cold working of the material.

Uniform circumferential spacing of longitudinal welds should be specified for individual tubes of multi-ply bellows in order to distribute stresses in welds at crowns during forming. Spiraling of these welds is recommended to prevent the full weld length from taking all the bending stresses at one time.

Bellows designs are limited by how severely the material can be worked during forming. Geometric limits for general use in bellows design have been determined (ref. 2). The following geometric limits are recommended for bellows up to three plies:

$$\text{Maximum: } \frac{OD}{ID} = 1.35$$

$$\text{Maximum: } \frac{R_{conv}}{t} = 2.0 \text{ (for straight-side-wall convolutions)}$$

where

R_{conv} = radius of convolutions, in.

t = wall thickness, in.

3.2.1.5.1 Interply Lubricants

Lubricants used to facilitate the assembly of multi-ply bellows shall not adversely affect the service life of the bellows.

Ensure that any lubricant used on a part is chemically compatible with the propellant to be contacted. Compatibility is particularly important if the propellant is LOX. The lubricant must be nonvolatile at operating temperatures.

In general, avoid the use of lubricants. If one must be used, solid dry-film lubricant (e.g., molybdenum disulfide) is recommended. If other types are used, provide vent holes and bake the bellows assembly to remove the lubricant. After baking, make a second circumferential weld inboard of the vent holes; then trim the bellows neck to eliminate the hole area.

3.2.1.5.2 Weld Necks

Doublers on bellows necks shall not create stress raisers in bellows convolutions.

A straight bellows doubler should end at least one convolution radius from the end convolution (fig. 56(a)). A doubler contoured to fit snugly against the end convolution (fig. 56(b)) also is an acceptable design; however, the end convolution is partially inactivated, and increased strain thus is applied to the remaining convolutions.

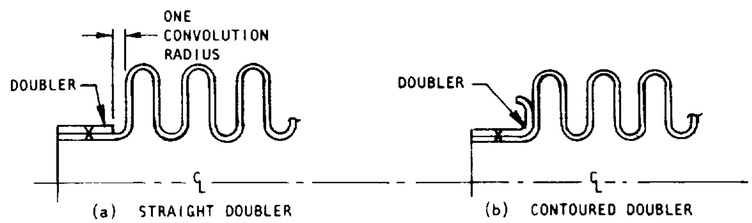


Figure 56. – Recommended configurations for bellows weld-neck doubler.

In instances where the bellows and line are made of different materials and one is heat treated, a short collar made of the non-heat-treated material should be welded to the heat-treated part prior to heat treatment. The bellows-to-line joint can then be made with common materials, the optimum weld-rod material being used.

Brazing is not recommended for pressure-carrying joints, because a brazed joint has relatively low strength and it is difficult to ensure good braze quality. There have been exceptions, however, dictated by manufacturing capability.

3.2.2 Bellows Restraint

3.2.2.1 MECHANICAL LINKAGES

3.2.2.1.1 Restraint Capability

The linkage shall be capable of restraining the pressure separating load of the bellows and preventing unacceptable axial elongation.

For most applications, use a gimbal ring, internal or external (fig. 24) as the needs dictate. Weight is a factor in most aerospace applications; therefore, high-strength materials should be used. Inconel 718, a nickel-base alloy, is recommended for both cryogenic and high-temperature applications. For an external gimbal ring, the diameter of the ring should be kept as close to the bellows OD as possible to minimize bending moments in the ring.

3.2.2.1.2 Load-Deflection Capability

The linkage shall be capable of permitting the design deflection of the bellows while restraining against pressure separating load and other external loads applied to the duct.

Ensure in the design layout phase that no linkage binding can occur under the maximum accumulation of manufacturing tolerances within the maximum specified deflection of the joint or duct. In complex linkages, a wooden or aluminum mockup is recommended as a physical check before committing expensive hardware in the shop.

3.2.2.1.3 Excursion-Limiting Stops

Excursion-limiting stops on the linkage shall prevent overdeflection and possible damage to the bellows.

Use limiting stops machined integrally with the bellows linkage. Adjustable stops are not recommended because of the possibility of error in adjustment. Verify by test that the stops prevent overdeflection.

3.2.2.1.4 Articulating Friction

The linkage bearing friction shall be a minimum consistent with the system loads limitations.

Use journal-type bearings in the pin joints of hinge and gimbal linkages. Metal-on-metal bearings may be used or, if a lower friction coefficient is desired, solid dry-film lubricant (molybdenum disulfide) may be applied to both journal and pin surfaces. Bearing pressures of 20 000 to 30 000 psi have been successfully accommodated with dry lubricant. When the bearings are in the flow path, care should be exercised to use only lubricants chemically compatible with the environmental fluids, particularly in oxidizer service.

In some applications it may be necessary to use antifriction bearings, as in the case of the ball joint (fig. 26), when the turning radius of the bearing is large and would require an excessive rotating moment if plain bearing surfaces were used.

3.2.2.1.5 Bending Moment

The spring rate of a flexible-joint assembly shall be consistent with the structural capabilities of the flex-joint and duct next-assembly mounting points and, in the case of gimballing ducts, with the actuator forces available.

The moment required to deflect a flex joint assembly angularly must overcome the inherent spring rate of the metal bellows, the friction between plies (if the bellows is multi-ply), the friction in the bearings of the linkage, and a bellows-straightening-moment geometric effect. Reference 166 presents a method for calculating the bending moment of a flex joint taking all of these factors into account. A computer program for making the calculations is included in the reference.

3.2.2.1.6 Centering of Pivot Point

The pivot point of the restraining mechanism shall ensure full deflection utilization of the bellows and thereby lowest bending stress.,

The live length of the bellows should be carefully centered on the pivot point of the restraining mechanism. This centering is especially important in the case of the internal chain-link restraint, where the pivot point can be in one of two different locations, one link diameter apart, the location depending on the plane in which angulation is taking place. Figure 57 shows the recommended design, which was used in the turbine-exhaust system of the H-1, J-2, and Phoebus systems. References 25 and 26 present data concerning pressure losses in this particular type of bellows restraint.

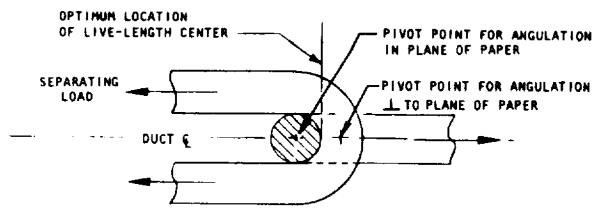


Figure 57. – Recommended design for centering bellows live length in chain-link restraint.

If the fatigue life is sensitive to variations in pivot point (as in the example in section 2.2.2.1), a ball bearing should be installed between the links to offset the inherent effect of change in pivot-point location with links alone. A recommended design, used on the Titan II and III booster and sustainer engines, is shown in figure 58.

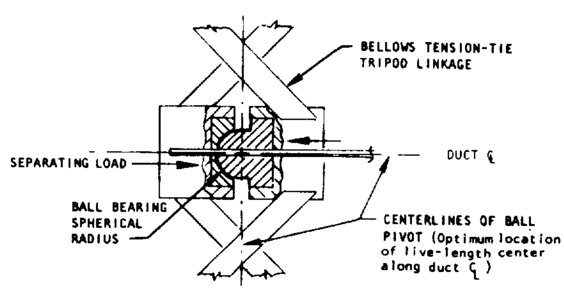


Figure 58. – Recommended ball-bearing installation to offset effect of change in pivot-point location.

3.2.2.1.7 Load Distribution

The bellows restraint shall distribute the pressure separating load evenly to adjacent flanged joints or primary components (e.g., pump inlets and valves) when those components are load limited.

Normally, when a length of hard line $\geqslant 2\ D_o$ is adjacent to a flexible joint, the concentrated loads are evenly distributed over the circumference of the hard line and thus do not represent a problem.

In closely coupled ducts where the flexible joints are located very near to the end-connecting bolted flanges of the duct, the external-ball-joint type of restraint is recommended for distributing the load within a short length of duct. This design, shown in figure 26, may use plain spherical bearings or ball bearings if low bending moment is a requirement.

3.2.2.2 THRUST-COMPENSATING LINKAGES

Straight sections of ducting that require large axial travel in a tension-type system shall not produce pressure separating loads on the attached structure.

Use a thrust-compensating joint to compensate for volume changes without creating a pressure separating load on the attached structure. There are many different concepts for the thrust-compensating design. Figures 31, 32, and 33 show three acceptable designs.

3.2.2.3 COMPRESSION SYSTEM

The installation of unrestrained (compression-type) bellows shall preclude axial or bending loads that may overload connecting hardware or buckle the bellows.

Locate the bellows so the pressure-separating-force vector passes through the center of the duct mounting points; if necessary, bellows can be installed with a biased shear deflection to offset the effect of thermal expansion during operation (fig. 35). This principle can be applied to axial deflection in the same installation. If the bellows is extended when installed, thermal expansion of the duct compresses the bellows toward neutral (i.e., to its free length).

3.2.3 Bellows-to-Duct Attachment

3.2.3.1 JOINT QUALITY

The method for joining the bellows to the duct shall provide a joint of the highest strength and highest quality consistent with other requirements.

Use a butt weld for joining the bellows neck to the duct (fig. 36). This technique provides a Class I weld (radiographically inspectable) and eliminates any stress-raising crevices that also could entrap contamination. In the case of thin-gage materials, a burndown weld, which also in the end result is a butt weld, should be used.

When butt welds are impractical, a lap joint can be used, but it should be welded on both ID and OD. Lap joints have little advantage other than being self-aligning for assembly ease.

To attach multi-ply bellows to a mating duct section, the multi-ply bellows end should be resistance welded and trimmed through the center of the nugget, or joined by another welding method. The resulting edge should then be butt welded to the mating duct.

3.2.3.1.1 Joint for High-Heat-Flux Region

Bellows-to-duct junctions subject to high heat fluxes and attendant thermal gradients shall be resistant to thermal fatigue failure.

A recommended design for this application is the forged Y-ring (fig. 59). The bellow-to-duct attachments should be Class I welds. (Note that this design is free of contaminant traps usually present in other designs ordinarily used.)

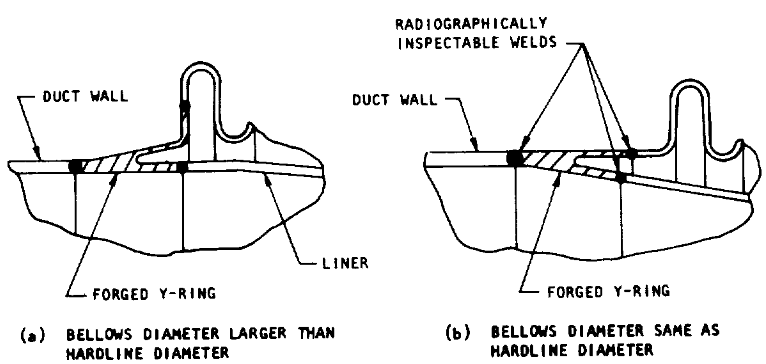


Figure 59. – Recommended bellows-attachment design for high-heat-flux region.

3.2.4 Flow Liners

Flow liners shall protect the bellows from flow-induced vibration and reduce pressure loss.

Recommended designs are full cantilevered liners, short conical liners, and half liners (fig. 40). A liner cantilevered from the upstream end and having a scalloped leading edge for flexibility to ease installation and fitup into the duct wall plus additional weld length for structural purposes can be used for bellows designed for axial deflection. Short conical liners can also be used; they can provide vibration protection and reduction in pressure loss equal to those of a full-length liner. For an angulating bellows, a half-liner from each end will minimize interference of the free end with the flow stream when the bellows is in the maximum deflected position.

Although liners for flexible hose have been investigated, none have been successful enough to warrant recommendation.

3.2.4.1 LINER BINDING

Flow liners for bellows shall perform the required flow function without interfering with the bellows deflection function.

Design the liner with sufficient clearance so that it will not bottom out against ducting structure under extreme bellows deflection conditions and thereby produce high reactive loads at the duct-attach points. Careful design layout of the bellows position with respect to the liner at maximum fabrication and deflection-excursion tolerances will ensure successful operation in all deflected positions.

3.2.4.2 LINER DRAINAGE

Flow liner design shall provide for proper cleaning and pressure relief.

Provide drain holes in flow liners for drainage of entrapped cleaning fluids or other contaminants. The holes also permit some relief of the pressure differential across the liner wall caused by sudden changes in flow through the bellows.

3.2.4.3 LINER COLLAPSE STRENGTH

Flow liners shall be capable of withstanding operational flow and pressure cycling without collapse.

The liner wall thickness should be sufficient to prevent collapse under the worst possible pressure differential. Reference 111 presents an analytical technique for a specific case that could be adapted for more general use.

An analysis should be made, in the preliminary design stage, of the possibility of flow conditions creating an external pressure on the liner. References 95 and 96 attest to inadequate design analysis of this condition and subsequent costly failures in the field.

3.3 FLEXIBLE HOSE

3.3.1 Routing

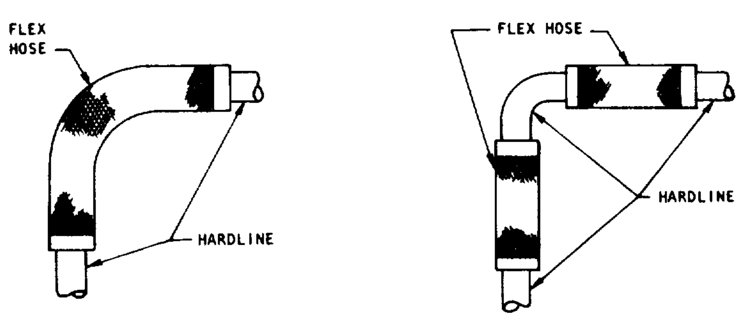
The design criteria and recommended practices for line-assembly routing (section 3.1.1) are also applicable to flexible-hose assemblies.

3.3.1.1 BEND CONFIGURATION

A bend in a flexible-hose assembly shall not cause excessive deflections or pressure losses in the flexible-hose elements.

Use a hard-tube elbow connected by straight lengths of flexible hose in the bend of a line. Undesirable and preferred configurations are shown in figure 60.

Although the configuration on the left is undesirable from the standpoint of steady-state stress and high pressure loss, it may be less susceptible to vibration modes perpendicular to the plane of the elbow than the version on the right, which, particularly in large-diameter high-pressure applications, would be quite stiff. If vibration is a problem, a tradeoff of the attributes of each design should be made before making a selection.



(a) UNDESIRABLE FLEX-HOSE ELBOW DESIGN

(b) PREFERRED FLEX-HOSE ELBOW DESIGN

Figure 60. – Undesirable and preferred configurations for a flexible elbow.

3.3.2 Sizing

3.3.2.1 FLOW AREA

Innercore geometry and flow area shall be consistent with the pressure-drop requirements of the line.

Use test data and empirical prediction methods (refs. 120 through 125) in the design layout phase to predict flow losses. Design the convolution geometry and line size to minimize friction loss.

3.3.2.2 INNERCORE/HARDLINE RELATION

Where pressure drop is critical, the inner diameters of the hard line and innercore shall be optimized for minimum total pressure loss.

For minimum pressure loss, calculate the total loss coefficient (expansion + innercore + contraction) for a range of ratios of innercore diameter to hardline diameter. Select the diameter ratio with the minimum total loss coefficient.

3.3.3 Innercore

The design criteria and recommended practices for bellows on pressure capability (sec. 3.2.1.1), fatigue life (sec. 3.2.1.2), corrosion resistance (sec. 3.2.1.4), and manufacturing (sec. 3.2.1.5) apply equally to flexible hoses. The criteria and practices on vibration (secs. 3.2.1.2.1 through 3.2.1.2.3) apply also, except that the innercore ordinarily is braid-restrained to vibrate only in the individual-convolution mode.

3.3.3.1 OPERATIONAL CAPABILITY

The innercore shall withstand dynamic deflections and temperature extremes.

In general, use metal hoses in all vehicle applications unless pressure drop is a critical factor and the environment is compatible with the nonmetals.

Hoses with a nonmetal innercore (e.g., Teflon, Kel-F, rubber) are not recommended for use in cryogenic applications, especially when pressure surges or dynamic deflections are involved, because they are subject to brittle fracture. They also are not recommended for temperatures above 500°F because of loss of strength and chemical decomposition.

3.3.3.2 BENDING MOMENT

Bending moments of flexible hoses that cross the gimballed engine-to-airframe interface shall be as low as possible.

The objective is to minimize loads on the gimbal actuator, particularly when many hoses are flexed as a bundle. All design variables such as wall thickness, number of plies, live length, and routing should be optimized to yield the lowest bending moment. Further reduction is possible through the application of a bonded dry-film lubricant to the outer surfaces of the innercore and to all surfaces of the restraining braid.

3.3.3.3 BUCKLING STABILITY

Flexible-hose braid restraint shall provide buckling stability when the hose is pressurized.

Do not constrain the flexible-hose elements in the line by clamps or brackets. They must be free to the extent that they can expand freely in the axial direction and permit the braid to tighten sufficiently to restrain the innercore from buckling. On this account, choose the length of braid so that it will tighten as required under load (sec. 3.3.4.2).

3.3.4 Braid

3.3.4.1 FREEDOM OF MOVEMENT

The braid configuration shall permit the hose assembly to angulate freely.

Braid in which the wires bottom out when angulated and thus prevent incomplete or unequal innercore deflection should be avoided.

The hose flexible section should be of sufficient length to permit each wire to have at least a 360° helical wrap with a 45° braid angle, thus making optimum use of the braid strength.

3.3.4.2 LENGTH STABILITY

The length of the flexible section shall be within dimensional tolerances after the application of proof pressure.

Provide sufficient braid-wire cross-sectional area to restrain the proof-pressure separating force of the innercore without yielding. Use equation (2) to determine the end strength of a braid. Determine axial growth due to tautening of the braid after application of proof pressure. Utilize manufacturing experience or equation (4) to predetermine the amount of stretch so that the flexible section will be within length tolerances after proof pressure is applied.

3.3.4.3 PRESSURE CAPABILITY

The flexible section shall retain its pressure-carrying capability during application of burst pressure.

Provide sufficient braid-wire cross-sectional area to restrain the burst-pressure separating force of the innercore. Use adequate braid to meet the burst-pressure requirement. Utilize equations (2) and (3) for determining braid-limited burst pressure (pressure above which the hose will leak or lose pressure). Some yielding is permitted, but the hose must retain its pressure-carrying ability. Also ensure that the braid will meet the nonyielding proof-pressure requirement.

3.3.5 End Construction

End construction for flexible metal hose shall provide structural integrity for braid attachment and innercore-to-tube end attachment under all operational conditions.

For small-diameter (< 3 in.) braided metal hoses to be used in the cryogenic to +400°F range, the end construction shown in figure 44(a) is recommended. Use a burndown-type fusion weld to attach the innercore to the tube adapter and attach the braid to the tube adapter with braze material. Use a thin-shell collar fitted snugly over the braid, with holes in the collar to permit observation of adequate fill during the brazing operation.

Use the end construction shown in figure 44(b) for braided metal hoses to be exposed to temperatures above 400°F either from environmental heat or hot internal fluid. In this all-welded construction, the braid and collar are fusion welded to the tube adapter. Make allowance in the original design for loss of braid strength due to welding.

The end construction shown in figure 44(c) is recommended for large-diameter (> 3 in.) braided metal hoses to be used in the cryogenic to +400°F temperature range.

3.3.6 Bending Moment

The moments required to deflect braided metal hoses shall be within the allowable limits for the hose intended usage.

Obtain from the intended supplier the predicted bending moments for flexible metal hoses designed for a specific application; otherwise, rely on data developed in previous engine and vehicle programs. Verify by structural analysis and later by test that the deflection-actuation system and reaction-load-carrying ability of attaching structure are adequate for the predicted and actual bending moments.

3.4 FILTERS

3.4.1 Filter Element

A filter element shall reduce fluid-system contamination to a tolerable level.

Use a surface filter element such as twilled-double-Dutch wire cloth for control of contaminant particle size in applications where maximum surface area with minimum weight is required. Design the element to provide a minimum of residual contamination and filter-material migration.

Use a depth filter such as stacked etched-metal disks when less control of contaminant size is required and contaminant tolerance is not a consideration.

3.4.1.1 FILTER RATING

Physical data from the glass-bead qualification tests shall provide an accurate determination of (1) the largest spherical particle that will pass through the filter and (2) the size distribution of particles passed.

Conduct filtration tests using artificial contaminant (glass beads) under specified conditions. The detailed test procedure is presented in MIL-F-8815 (ref. 133). To prevent glass beads from coagulating and obscuring the count, add a l-percent solution of wetting agent. An effective wetting agent commercially available is Triton X-100.

When control of maximum particle size is required, multiply the glass bead rating by a factor of 2.5 for twilled-double-Dutch wire cloth.

3.4.1.2 SYSTEM CONTAMINATION/FILTER AREA

3.4.1.2.1 System Contamination

System contamination shall be limited by prefiltering the operating fluids prior to final filtering in the system.

Use a large-capacity system prefilter having a finer rating than that required by the system, so that fluid contamination is reduced to a level that will preclude overloading of the filter installed to protect contamination-sensitive components such as spool valves and metal-to-metal poppets and seats. Prefiltering is not always required when adequate external propellant loading filters are utilized.

3.4.1.2.2 Filter Area

The amount of filter area shall be a maximum based on careful analysis of pressure-drop and contaminant-tolerance requirements.

Use pleated twilled-double-Dutch woven-wire cloth to obtain the maximum amount of surface area for maximum service life. A cylindrical or conical shape provides the best design. Stacked etched-metal disk filters should be used for finer filtration when pressure drop and weight are not critical considerations.

3.4.1.3 RESIDUAL FILTER CONTAMINATION

The initial filter cleanliness shall be at an acceptable level.

Control the environment during fabrication, assembly, and testing. Use the SAE ARP599 (ref. 139) test method with modification. When flow can occur in either direction, conduct the test for both the upstream and downstream sides separately by flowing through the filter in both directions. When flow is in one direction, the downstream side only need be checked. The exception to ARP 599 is that each sample should be 500 ml in 5 equal increments of 100 ml. It is recommended that for normal installations the particle count level be consistent with system requirements (ref. 131). When extremely contamination-sensitive components are involved, the particle-count level specified in table VIII is recommended.

Table VIII. – Recommended Allowable Contaminant-Particle Count for a Critical-Installation Filter

Size range, μ	Number of Particles (500 ml sample)
5 to 10	*
10 to 25	100
25 to 50	20
50 to 100	5
100 to 300	1
300 and over	0
Fibers	0

*No slurry of fine particles that covers more than an estimated 5 percent of the total effective membrane filter area shall be allowed.

3.4.1.4 FILTER MATERIAL AS CONTAMINANT

The filter material and supporting structure shall not be a source of contamination in the system.

Woven-wire cloth is recommended for use in applications requiring filtration ratings down to 15μ absolute (GBR). The woven-wire filter cloth having the best resistance to media migration is twilled double-Dutch. When a filtration rating finer than 15 μ absolute (GBR) is required, use stacked etched-metal disks, taking into consideration weight and differential-pressure penalties. If weight and differential pressure are critical, less desirable techniques such as sintered matted metal fibers or the addition of a sintered powder to the upstream side of the wire cloth or calendering of the cloth may be employed to obtain the finer filtration rating. Design the supporting structure so that clearances are held at a maximum to prevent contact under flow-induced or system vibration. Attach wire-cloth elements to the supporting structure by welding; use tungsten-inert-gas, heliarc, or electron-beam process.

Do not confuse media-migration and residual-contamination requirements.

3.4.1.5 PRESSURE DROP

Design calculations shall predict the pressure drop across the filter for the required flowrate.

Assume that essentially all the pressure drop across the filter will occur at the smallest cross-sectional flow area, which is usually determined by the supporting structure. Calculate the pressure drop for the smallest cross-sectional area by means of the conventional equations for flow through an orifice.

Attention also must be paid to the pressure drop created by the attachment of the element to the case and to the case itself. Use the empirical equations of reference 132 to determine the pressure drop for the filter.

3.4.1.6 TESTING

Test fluids used for evaluation and performance-demonstration tests shall provide consistent and reliable data.

The following fluids are recommended as most suitable for the given filter tests, providing that correlation by test with the working fluid is maintained:

Test	Fluid
Flow/pressure drop	Freon*
Largest particle passed	De-ionized water
Dirt capacity and collapse pressure	De-ionized water
Bubble point	Isopropanol (reagent grade)

*Meets specifications of MIL-C-81302B, Amend. 1, May 16, 1973.

Before a test, the test fluids should be prefiltered with an 0.8μ-maximum membrane-type filter. After test, the filter should be oven-vacuum dried (135°F and 1.0 psia) for a minimum period of 1 hour.

3.4.2 Filter Case

A filter case shall prevent movement of the filter element during engine operation, and the case design shall preclude inadvertent omission of the filter element during assembly.

Design the filter element such that it can be permanently attached to the case by welding. Case designs that permit removal and replacement of the element should provide minimal clearance at each end to prevent lateral movement. An anti-rattle Belleville spring can be used to prevent longitudinal movement when tolerance-stackup gaps exist between the end of the element and case.

Design the filter case such that the filter element is permanently attached by welding to one of the case ends. Case designs that permit removal and replacement of the element should incorporate a test to verify that the element is installed.

APPENDIX A

Conversion of U.S. Customary Units to SI Units

Physical quantity	U.S. customary unit	SI unit	Conversion factor[a]
Angle	degree	radian	1.745×10^{-2}
Density	lbm/in.^{3}	kg/m^3	2.768×10^{4}
Elasticity	psi (lbf/in.^{2})	N/cm^2	6.895×10^{-1}
Energy	Btu	J	1.054×10^{3}
Force	lbf	N	4.448
Length	ft	m	3.048×10^{-1}
	in.	cm	2.54
	micron	μm	1.00
Mass	lbm	kg	4.536×10^{-1}
Pressure	psi	N/cm^2	6.895×10^{-1}
Strength	psi	N/cm^2	6.895×10^{-1}
	ksi (1000 psi)	N/cm^2	6.895×10^{2}
Stress	psi	N/cm^2	6.895×10^{-1}
Temperature	°F	K	$K = \frac{5}{9}(°F + 459.67)$
Thermal conductivity	Btu/(hr-ft-°F)	J/(sec-m-K)	1.730
Thermal expansion coefficient	in./(in.-°F)	m/(m-K)	1.8
Volume	gal	m^3	3.785×10^{-3}
	ml	cm^3	1.00

[a]Multiply value given in U.S. customary unit by conversion factor to obtain equivalent value in SI unit. For a complete listing of conversion factors for basic physical quantities, see Mechtly, E. A.: The International System of Units. Physical Constants and Conversion Factors. Second revision, NASA SP-7012, 1973.

APPENDIX B

GLOSSARY

Term or Symbol	Definition
A	effective area of bellows
A_w	cross-sectional area of each wire in a braid
angulation	angular deflection imposed on a flexible joint
axial deflection	elongation or compression of a flexible joint along its longitudinal axis
B_e	braid efficiency
bellows	corrugated thin-wall pressure vessel, usually cylindrical in shape, that when integrated into a duct or line assembly can accommodate duct movements through deflection of the corrugations
bubble point	gas pressure at which a gas bubble forms at the surface of a filter immersed in a test fluid (per ref. 135)
burndown weld	fusion butt weld with no material added; usually applies to thin-gage duct materials with burndown lips bent up on ends to be welded
compression system	duct system wherein the fluid-column loads due to internal pressure are reacted by the support structure
contaminant tolerance	the amount (by weight) of a standard contaminant (added at the inlet of a filter under specified fluid flowrate, temperature, and pressure) that causes the pressure loss in the filter to exceed a maximum allowable value
contamination tolerance level	the value of contaminant particle size, or level of contamination, in a fluid system at which the specified performance, reliability, or life expectancy of the components of the system is adversely affected
convolution (corrugation)	longitudinal wave plastically formed in a thin-wall (usually metal) tube
coupon	a piece of material, representative of the material used in a part, that accompanies the part during processing and subsequently is used as a test specimen to evaluate properties of the part

Term or Symbol	Definition
cryogenic	fluids or conditions at or below -238°F
cryopump	to condense and freeze water vapor and gases on extremely cold surfaces (e.g., those at liquid-hydrogen temperature) with the result that, in a confined cavity, the pressure is lowered
D_m	mean diameter of bellows, $D_m = ½ (OD + ID)$
D_o	outside diameter of bellows ($\equiv$ OD)
E	Young's modulus of elasticity
effective area	area of a bellows joint at the main diameter of the convolutions; internal pressure exerted over this area creates axial or end thrust (pressure separating force) tending to elongate the bellows
ELI	extra low interstitial (metallurgical term)
end strength	axial-load-carrying capability of braided wire sheath of a flexible hose section; load is created by pressure separating force in the hose
endurance limit	maximum stress at which a material presumably can endure an infinite number of stress reversals
exhaust blowback	backflow of the exhaust from the rocket engines into the upper portion of the engines
exhaust plume	hot gases ejected from the thrust chambers of rocket engines; the plume expands as the vehicle ascends, thus exposing the engine and vehicle to greater radiative area
F_B	braid end strength
F_w	strength of a single wire in a braid
fatigue life	number of cycles of stress, under a stated test condition, that can be sustained by a material prior to failure
filter	device in a fluid system that limits size and amount of particulate contamination in the system downstream of itself
flexible hose	pliant conduit consisting of an inner core of convoluted metal or plastic or plastic tubing and an outer braided wire sleeve that is attached to fixed ends to prevent buckling and separation when the core is pressurized

Term or Symbol	Definition
flexible joint (flexible section)	metal bellows, flexible hose, or ball joint assembly that joins two duct sections and permits relative motion between the ducts in one or more planes; includes both the flexible member and the restraint linkage
GBR	glass bead rating
Gimbar	gimbal ring with crossed bars for structural strength
gland	the cavity in a flange or fitting that retains a static seal
gpm	gallons per minute
H900	tempering process for martensitic precipitation-hardening steels
hard line	line or duct that incorporates no flexible joints but is provided with deflection capability through the use of loops and elbows and low-modulus or thin-wall material
high-cycle fatigue	fatigue life, characterized by elastic strains, greater than 10^4 cycles
homokinetic plane	in universal joints (or gimbal joints), the plane that is normal to the plane containing the shafts and bisecting the angle between them (ref. 106)
ID	inner diameter
innercore	pressure-carrying tubular member of a flexible hose; made of convoluted metal or plastic material or plain tubular plastic material such as Teflon or rubber
intergranular corrosion	phenomenon generally associated with stainless steels; caused by formation of a complex chromium carbide that depletes the chromium content along the grain boundaries
kinematics	the study of motion exclusive of the influences of mass and force
ℓ	total length of one braid wire between end connections
LEM	lunar excursion module
line or duct	terms used interchangeably for an enclosed passageway (made of sheet metal or other suitable material) that conveys fluids under pressure
live length	overall length of the convolutions in a bellows that are capable of accommodating imposed deflections

Term or Symbol	Definition
low-cycle fatigue	fatigue life, characterized by plastic strains, in the 10^3 to 10^4 cycle range
manifold	fluid-flow enclosure that distributes the flow in a desired manner from an inlet or inlets to an outlet or outlets
MPR	maximum particle size rating
MSLD	mass spectrometer leak detector
n	total number of wires in braid
NASTRAN	acronym for NASA structural analysis
OD	outer diameter ($\equiv D_o$)
orange peel	surface roughening that occurs when a metal of coarse grain is stressed beyond its elastic limit; the grain pattern formed has the appearance of an orange peel
outgassing	gradual release of sizable quantities of gas from enclosed surfaces when an enclosure is vacuum pumped
P_b	hose burst pressure (maximum pressure a flexible hose can retain without losing pressure or fluid)
P_i	hose internal pressure
plain Dutch single weave (PDSW)	woven wire cloth in which each shute wire passes successively over one and under one warp wire, each successive shute wire alternating the order; the shute wires are smaller than the warp wires and are closely spaced, the result being a dense weave
planish	to produce a smooth surface finish on a wrought metal, usually by rolling with highly polished rolls
Pogo	term for feed-system-coupled longitudinal oscillations of a rocket vehicle; named after motion of a pogo stick
pressure separating force (or load)	force (or load) generated by internal pressure tending to separate two parts of a line assembly along the line of the force; for a convoluted section such as a bellows or metal innercore, this force is equal to the product of the internal pressure and the cross-sectional duct area at the mean diameter of the convolutions

Term or Symbol	Definition
σ_b	bending stress (motion stress)

Material*	Identification
A-286	heat-treatable, high-strength austenitic steel
Aerozine 50, A-50	50/50 mixture of UDMH and hydrazine per MIL-P-27402
AM 350, AM 355	semi-austenitic precipitation-and transformation-hardening steels
Armco 21-6-9	iron-base 21Cr-6Mn-9Ni alloy manufactured by Armco Steel Corp.
body-centered cubic	common metal crystal structure in which the cubic unit cell has one atom located at each corner and one atom in the center of the cube; this structure is not suitable for cryogenic use because it has fewer degrees of freedom for movement than face-centered-cubic structure
CRES	corrosion-resistant steel
Electrofilm 77S	solid dry-film lubricant manufactured by Electrofilm Co. (N. Hollywood, CA)
electroless nickel	nickel plate achieved by a chemical reduction process as distinguished from the electrolytic deposition process
Epibond 121	an unfilled epoxy adhesive manufactured by Furane Plastics Co. (Glendale, CA)
face-centered cubic	common metal crystal structure in which the cubic unit cell has one atom located at each corner and one atom in the center of each face of the cube; this structure is suitable for cryogenic use because it has many degrees of freedom for movement
Freon	trade name of E. I. du Pont de Nemours & Co. for a family of fluorinated hydrocarbons
GH_2	gaseous hydrogen
GN_2	gaseous nitrogen
GOX	gaseous oxygen

*Additional information on metallic materials herein can be found in the 1972 Handbook, SAE, Two Pennsylvania Plaza, New York, NY; in MIL-HDBK-5B, Metallic Materials and Elements for Aerospace Vehicle Structures, Dept. of Defense (Washington, DC), September 1971; and in Metals Handbook (8th ed.), Vol. 1: Properties and Selection of Metals, Am. Society for Metals (Metals Park, OH), 1961.

Material	Identification
Hastelloy C	heat-resistant nickel-base alloy manufactured by Stellite Div., Cabot Corp.
Haynes Stellite 6B, Star J	cobalt-base Cr-Ni-W-Fe alloys manufactured by Stellite Div., Cabot Corp.
helium (He)	pressurant helium per MIL-P-27407
Inconel 600, 625, 718, X-750	nickel-base alloys manufactured by International Nickel Co.
Kel-F	trade name of 3M Corp. for a polymer of chlorotrifluoroethylene
"K" Monel	trade name of International Nickel Co. for a wrought age-hardenable alloy containing Ni, Cu, and Al
LH_2	liquid hydrogen (H_2), propellant grade per MIL-P-27201
Lithafrax filler	lithium aluminum silicate, a filler material with a low or negative coefficient of thermal expansion; manufactured by Carborundum Corp. (Niagara Falls, NY)
LN_2	liquid nitrogen
LOX	liquid oxygen (O_2), propellant grade per MIL-P-25508
Micarta	trade name of Westinghouse Electric Corp. for a group of laminated plastics involving paper or cloth and phenolic or melamine resins
MMH	monomethylhydrazine, propellant grade per MIL-P-27404
Monel	Ni-Cu alloy manufactured by International Nickel Co.
N-155	iron-base Co-Cr-Ni-Mo-W alloy
N_2H_4	hydrazine, propellant grade per MIL-P-26536
N_2O_4	nitrogen tetroxide (oxidizer), propellant grade per MIL-P-26539
PH15-7Mo	semi-austenitic precipitation- and transformation-hardening steel
Refrasil	trade name of HITCO (Los Angeles, CA) for fibrous silica of high purity
Rene 41	trade name of General Electric Co. for an austenitic Ni-Cr-Co-Mo alloy

Term or Symbol	Definition
pressure-volume compensator (PVC)	flexible duct system that, by means of mechanical linkage to a series of secondary bellows having an effective area equal to that of the primary bellows, creates an opposing force that counteracts the end thrust
psig	pounds force per square inch, gauge
$R = -1$	designation showing that imposed strain in fatigue testing is equal in both directions from neutral
R_{conv}	radius of convolution
R/D	ratio of the centerline bend radius (R) to the inside diameter (D) in an elbow
Re	Reynolds number
retort	vessel used in an oven or furnace to enclose the work being heat treated in a controlled atmosphere
RH950	tempering process for austenitic precipitation-hardening steels
safety factor	an arbitrary multiplier greater than 1 applied in design to account for design uncertainties, e.g., slight variations in material properties fabrication quality, and load distributions within the structure
SCT850	tempering process for austenitic precipitation-hardening steels
scc/sec	standard cubic centimeters per second
S-N	stress vs number of cycles to failure; plots of such data are used in fatigue testing
SSME	Space Shuttle main engine
storable propellant	propellant with a vapor pressure such that the propellant can be stored in a specified environment (earth or space) at moderate ullage pressures without significant loss over a specified period of time
stubout	tubular nipple protruding from a component to which the connecting line is welded or brazed
super-insulation	high-efficiency laminated-foil insulator used in low temperature applications; thermal conductivity is 1/10 to 1/150 that of common insulating materials

Term or Symbol	Definition
surfactant	surface active agent; e.g., material that reduces surface tension when dissolved in water
T3, T4, T6, T73, T351	designations for heat-treating and tempering processes for aluminum alloys
t	thickness
tension system	duct system wherein the fluid-column or longitudinal forces due to internal pressure are not transmitted to the supporting structure; the fluid-column loads of such a duct system are reacted by axial tension in the duct walls
TH1050	tempering process for austenitic precipitation-hardening steels
thrust vector control	steering or guidance of the vehicle by angular deflection of the rocket engine thrust vector; e.g., by gimballing the engine about a pivot point
torsional deflection	deflection imposed on a flexible joint by applying a torque about its longitudinal axis
twilled double Dutch weave (TDDW)	woven wire cloth in which each shute wire passes successively over two and under two warp wires and each warp wire passes successively over two and under two shute wires; the shute wires are smaller than the warp wires and overlap, the result being a dense weave
weld, Class I	weld that must be inspected by visual, dimensional, magnetic particle or penetrant, and radiographic or ultrasonic means and meet specifications on defects and porosity
weld, Class II	weld that must be inspected by visual, dimensional, and magnetic particle or penetrant means and meet specifications on defects and porosity
wraparound	term for a flexible duct or hose assembly that "wraps around" the thrust-vector-control gimbal of a rocket engine in the plane of the gimbal
δ	braid or helix angle
θ	required gimbal angle
ξ	braid elongation
σ_B	bulging stress

Material	Identification
RJ-1	ram-jet fuel, propellant grade per MIL-F-25558
RP-1	kerosene-base hydrocarbon fuel, propellant grade per MIL-P-25576
Teflon	trade name of E.I. du Pont de Nemours & Co. for a variety of polymers of tetrafluoroethylene
Ti-5Al-2.5Sn	titanium-base Al-Sn alloy used in the annealed condition only
Ti-6Al-4V	titanium-base Al-V alloy used in either annealed or solution-treated and aged conditions
Ti-40A	one of five grades of commercially pure titanium, yield strength of approximately 40 ksi
trichloroethylene	$CHCl{=}CCl_2$, a chlorinated hydrocarbon solvent per MIL-T-27602
Triton X-100	wetting agent manufactured by Rohm & Haas (Philadelphia, PA)
UDMH	unsymmetrical dimethylhydrazine, propellant grade per MIL-P-25604
Viton	trade name of E.I. du Pont de Nemours & Co. for linear copolymer of vinylidene fluoride and hexafluoropropylene
Waspaloy	precipitation-hardening nickel-base superalloy manufactured by Pratt & Whitney Aircraft Div., United Technologies Corp.
zeolite	designation for a group of sodium aluminum silicates
17- 4PH, 17-7PH	precipitation-hardening wrought stainless steels
18%-Ni maraging steel	iron-base Ni-Co-Mo, age-hardenable alloy
18-8 CRES	family of austenitic corrosion-resistant steels containing approximately 18% chromium and 8% nickel
19-9DL	trade name of Cyclops Corp. for an austenitic iron-base 19Cr-9Ni alloy
304, 304L, 310, 316, 321, 347	chromium-nickel austenitic stainless steels (18-8 family)
2014, 2024, 2219	wrought aluminum alloys with Cu as the principal alloying element
6061	wrought Al alloy with Mg and Si as the principal alloying elements

Material	Identification
7075	wrought aluminum alloy with Zn as the principal alloying element

ABBREVIATIONS

Organization	Identification
AFBSD	Air Force Ballistic Systems Division
AFML	Air Force Materials Laboratory
AFRPL	Air Force Rocket Propulsion Laboratory
AFSC	Air Force Systems Command
AIAA	American Institute of Aeronautics and Astronautics
ASCE	American Society of Civil Engineers
ASLE	American Society of Lubrication Engineers
ASME	American Society of Mechanical Engineers
BSD	Ballistic Systems Division
CPIA	Chemical Propulsion Information Agency
DMIC	Defense Metals Information Center
ICRPG	Interagency Chemical Rocket Propulsion Group
JSME	Japanese Society of Mechanical Engineers
MSC	Manned Spacecraft Center (now Lyndon B. Johnson Space Center)
MSFC	Marshall Space Flight Center
SAE	Society of Automotive Engineers
SAMPE	Society for Advancement of Material and Processing Engineering
SRI	Southwest Research Institute
UCLA	University of California at Los Angeles

Organization	Identification
WADC	Wright Air Development Center
WPAFB	Wright-Patterson Air Force Base

Document	
AIR	Aerospace Information Report
ARP	Aerospace Recommended Practice

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