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8000 psi Hydraulic Systems: Experience and Test Results

RATIONALE

This document has been determined to contain basic and stable technology which is not dynamic in nature.

STABILIZED NOTICE

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1. SCOPE:

1.1 Shortly after World War II, as aircraft became more sophisticated and power-assist, flight-control functions became a requirement, hydraulic system operating pressures rose from the 1000 psi level to the 3000 psi level found on most aircraft today. Since then, 4000 psi systems have been developed for the U.S. Air Force XB-70 and B-1 bombers and a number of European aircraft including the tornado multirole combat aircraft and the Concorde supersonic transport. The V-22 Osprey incorporates a 5000 psi hydraulic system. The power levels of military aircraft hydraulic systems have continued to rise. This is primarily due to higher aerodynamic loading, combined with the increased hydraulic functions and operations of each new aircraft. At the same time, aircraft structures and wings have been getting smaller and thinner as mission requirements expand. Thus, internal physical space available for plumbing and components continues to decrease.

1.2 In the 1960s, the U.S. Navy began a methodical process of developing lightweight hydraulic systems (LHS) for aircraft. The Navy was keenly aware that aircraft designers would be faced with requirements for higher horsepower hydraulic systems in future high-performance, high-density aircraft. The Air Force has also pursued the use of LHS to complement their Fire Resistant Hydraulic Systems program. One logical way to achieve smaller and lighter weight hydraulic components is to raise the system operating pressure levels. Studies conducted under both Navy and Air Force programs have indicated that 8000 psi hydraulic systems are feasible and can achieve significant weight reduction and space savings for certain aircraft. Many of the hardware elements necessary for 8000 psi hydraulic systems have been designed, fabricated, and tested in the programs discussed herein. The purpose of this document is to outline experience and test results to date when operating hydraulic systems at 8000 psi.

2. TEST PROGRAMS:

2.1 Exploratory Development Programs:

2.1.1 In the Navy's development programs, conducted by Rockwell, many laboratory tests and analyses have been performed to determine the performance of fluids, pumps, transmission lines, actuators, and seals; and to investigate component sizing and system response at elevated pressures. Based on the promising results obtained, it was concluded that practical systems could be designed for operation at 8000 psi; and that pressure was selected for further exploration. All the Navy studies and programs use MIL-H-83282 fluid.

2.1.2 In 1971, Grumman Aircraft prepared an analysis of an F-14 aircraft wherein a LHS designed for 8000 psi was compared to the existing 3000 psi system. The basic guidelines of the study were as follows:

a. The hydraulic system, as originally proposed for the Lot I F-14 aircraft, (3000 psi) was used as the basis for comparison.

2.1.2 (Continued):

b. The F-14 actuator stroke lengths, attaching points, loads, and excursion rates were maintained.

c. No changes in the lengths and routing of the F-14 hydraulic transmission lines were made.

d. The F-14 reservoir fluid temperature level for the pre-dash condition served as a criterion for the LHS system.

e. The LHS system maximum pressure drop values was to be within a ratio of 8:3 when compared to an equivalent F-14 tube size.

f. The present F-14 relationship between system pressure and operating pressure was used to determine LHS actuator bores

3000 psi system pressure versus 2400 psi operating pressure (F-14)

8000 psi system pressure versus 6400 psi operating pressure (LHS)

g. Tubing wall thicknesses for the LHS system was determined using the ultimate stress value at 450 °F of the tube material (Titanium 3Al-2.5V cold worked per GM3107) together with a burst pressure value of 24 000 psi (3 x 8000 psi) for pressure lines. The wall thickness for return lines was determined by using a burst value of 12 000 psi (3 x 4000 psi). Suction line burst pressure values were equal to 3 x 600 psi or 1800 psi.

The results of the Grumman study indicated that reductions obtainable due to the higher pressure would yield a system weight reduction of 30.4% and a volume reduction of 40.4%.

2.1.3 Laboratory tests were conducted at Rockwell to demonstrate that properly designed hardware will operate for extended periods at 8000 psi. The systems contained most of the components normally used in aircraft hydraulic systems. In addition, special test actuators were used to evaluate many seal configurations.

Following the laboratory tests, a Navy/Rockwell T-2C trainer was used to flight test 8000 psi hydraulic system hardware. An 8000 psi lateral control system was installed in the T-2C that was instrumented and then flown for over 11 h. An 8000 psi fly-by-wire rudder actuator installation was subsequently evaluated: over 40 h of flight time were accumulated with no problems.

2.2 Navy Program:

2.2.1 After the flight test, Rockwell and the Navy began a program in 1979 with the following goals:

a. Redesign of high performance aircraft hydraulic system from 3000 to 8000 psi.

b. Verify weight and volume improvement

c. 600-900 h of full-scale simulator testing (1200 h completed)

d. 100-300 h flight test evaluation (not performed)

e. Establish total specification requirements for 8000 psi aircraft hydraulic system

The Phases were designated as follows: Phase I - System Design, Hardware Fabrication and Test and Phase II - Full-Scale Simulator Testing.

2.2.2 A Vought A-7E aircraft was selected for performing a retrofit conversion from 3000 to 8000 psi. Primary flight control (FC) actuators included ailerons, spoilers, roll feel, rudder, and unit horizontal tail. Secondary FCs included the speed brake and wing leading edge flaps. The automatic flight control system (AFCS) included roll, pitch, and yaw actuators. The aircraft hydraulic system was reconfigured from three independent power control systems operating at 3000 psi to two independent 8000 psi FC systems and one 3000 psi utility system.

2.2.3 As part of Phase I, an analysis was made of the weight and space savings that would result of retrofitting the A-7E hydraulic system from 3000 psi to 8000 psi. Ground rules were basically the same as those used for the F-14 comparison discussed in 2.1. Results were a weight reduction of 30.2% and a space reduction of 36.3%. The results were very similar to those attained by Grumman.

2.2.4 Phase II effort consisted of the following:

a. Fabrication and assembly of a full-scale A-7E 8000 psi simulator

b. Performance of the following tests: Proof Pressure, Dynamic Performance, and 1200 h Endurance.

2.2.5 The Vought Corporation played a major role in the development of 8000 psi technology. For the Navy/Rockwell program, Vought designed and fabricated a majority of the actuators used in the simulator. In addition, Vought Corporation performed an extensive seal development and evaluation program to provide the necessary static, rod, and piston seals. Supplementing a prior effort done under a Navy contract, Vought continued to develop coiled tubing for 8000 psi systems.

2.2.6 Vought also conducted a low-temperature 8000 psi evaluation. This program compared the warm up performance of an 8000 psi lateral control system using 3/16 in x 0.020 wall thickness tubing with an equivalent 3000 psi control system using larger diameter tubing. Results of all tests and analysis indicated there was no essential difference in warm up time for systems operation at 3000 or 8000 psi pressure levels.

2.2.7 In addition, Vought completed a survivability assessment of an 8000 psi hydraulic system. It indicated that the conversion from 3000 psi to 8000 psi would result in an average of 39.7% reduction in probability of kill (P_k) for FCs.

2.3 Fire Resistant Hydraulic Systems Program - Air Force:

2.3.1 The McDonnell Aircraft Company conducted a program for the Air Force to determine the flight worthiness of fire-resistant hydraulic systems. The objective was to reduce the system weight penalty associated with the use of nonflammable chlorotrifluoroethylene (CTFE) fluid while: (1) improving maintainability, (2) limiting cost increases, and (3) developing technical innovations.

2.3.2 8000 psi was selected as the system operating pressure. Concepts and approaches employed included:

a. Force motor actuated direct-drive valves

b. Nonlinear control valves

c. Distribution system modifications which include “odd-even” line sizes, asymmetric line loss, local velocity reduction, and restrictor elimination in utility functions.

2.3.3 Weight analysis of an F-15 showed that a 3000 psi hydraulic system using CTFE fluid would weigh 1551 lb. A retrofit CTFE fluid hydraulic system operating at 8000 psi would weigh 1120 lb. This is a weight savings of 27.7%.

2.3.4 Weight analysis of a KC-10A aircraft showed that a 3000 psi hydraulic system using CTFE fluid would weigh 7256 lb. A retrofit CTFE hydraulic system operating at 8000 psi would weigh 5136 lb. This is a weight savings of 29.2%.

2.3.5 A stabilator actuator with a direct-drive valve, a trailing-edge flap actuator, and a distribution/fluid system representing an F-15 aircraft using CTFE fluid were endurance tested for 750 h (–65 to +275 °F). The results of this program are reported in 4.2.

3. HYDRAULIC FLUID DEVELOPMENT:

3.1 Physical Properties at 8000 psi:

3.1.1 System Fluid: Petroleum based hydraulic fluid per specification MIL-H-5606 has been used successfully in military aircraft for over 35 years. This fluid has many excellent properties, but it also has two significant shortcomings: (1) a maximum operating temperature limit of +275 °F and (2) high flammability. Considerable effort has been expended in recent years to develop a fluid that has improved thermal stability and is fire resistant.

3.1.1.1 Tests reveal that MIL-H-5606 exhibits poor shear stability when used in a system operating at pressure levels from 6000 to 9000 psi. This is attributed to polymeric additives, which are used to improve the viscosity index of the base stock. The severe fluid shearing action that occurs in very high pressure systems tends to rupture the long polymeric molecules causing permanent degradation of viscosity. This loss in viscosity adversely affects component wear and internal leakage rates.

3.1.1.2 A new synthetic hydrocarbon hydraulic fluid has been developed that is compatible with MIL-H-5606 fluid but is superior in several areas. The fluid is defined in specification MIL-H-83282. The Naval Air Development Center (NADC) in Warminster, PA and the Air Force Systems Command, WPAFB in Dayton, OH have both evaluated MIL-H-83282 fluid and the results have been published. MIL-H-83282 is more viscous than MIL-H-5606 at low temperature. For example, MIL-H-83282 has a viscosity at –40 °F equivalent to MIL-H-5606 at –65 °F. This may cause undesirable system warm up time prior to achieving full operational capability dependent upon aircraft system design. Specification requirements of MIL-H-83282 fluid compared to MIL-H-5606 fluid are as follows in Table 1:

TABLE 1 – MIL-H-83282 and MIL-H-5606 Comparison

	MIL-H-5606B	MIL-H-83282
Maximum Operating Temperature	+275 °F	+400 °F
Flash Point	+200 °F	+410 °F
Auto Ignition Temperature	+475 °F	+685 °F
Kinematic Viscosity of 2200 cs	-65 °F	-40 °F

3.1.1.3 There are no polymeric additives in MIL-H-83282 fluid; shear stability is therefore improved over MIL-H-5606 fluid. MIL-H-83282 fluid was evaluated at 8000 psi in the Navy A7-E Lightweight Hydraulic Simulator at Rockwell in Columbus. Results of testing showed no fluid property degradation after extensive use at 8000 psi.

3.1.2 Physical Properties:

3.1.2.1 Viscosity: Values of fluid viscosity versus temperature at atmospheric pressure are shown on Figure 1. Data for viscosity at 8000 psi were generated by measuring pressure loss in straight tubing, and calculating the “effective viscosity” encountered under low-shear-rate, laminar-flow conditions.

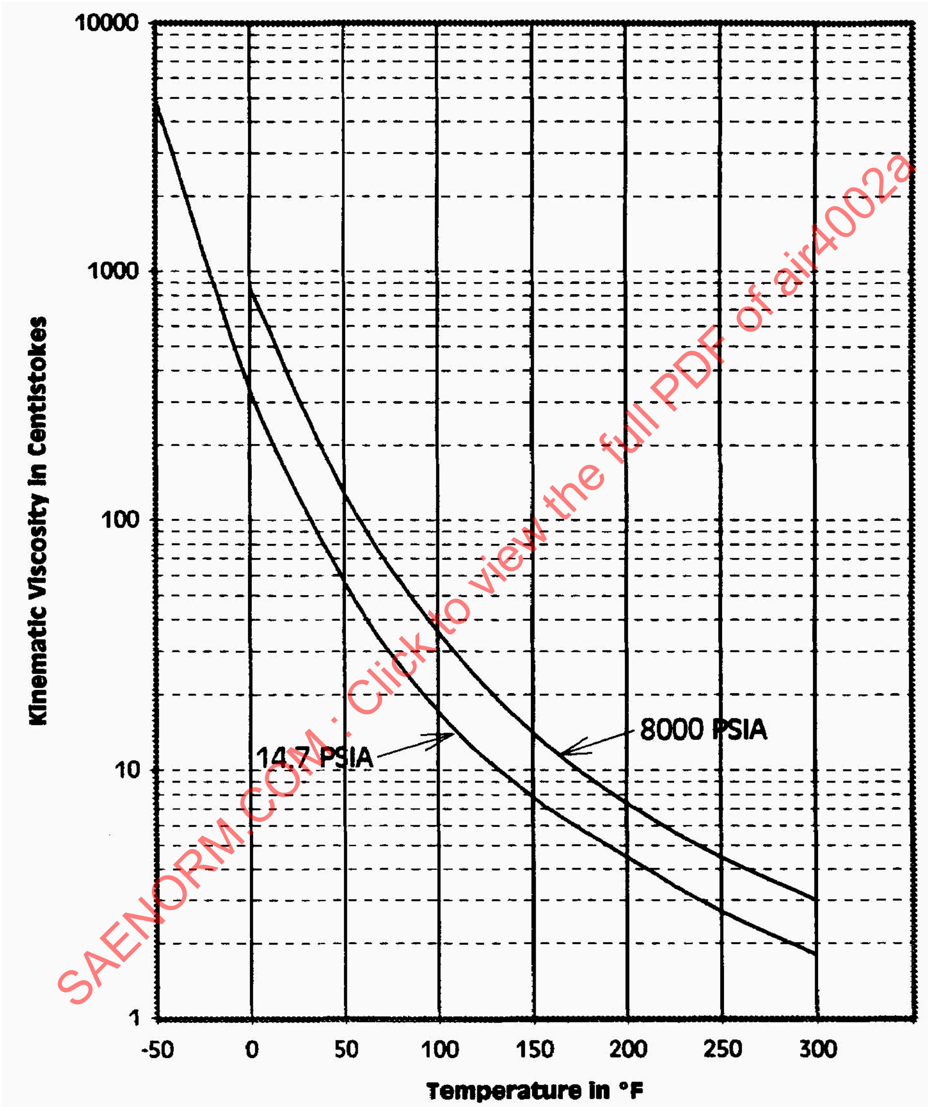


FIGURE 1 – Viscosity Versus Temperature of MIL-H-83282 Fluid

3.1.2.2 Bulk Modulus: Fluid bulk modulus can be expressed in six forms. The application determines which type should be used. The most commonly used moduli are

3.1.2.2 (Continued):

a. Adiabatic (Isentropic) Tangent (Dynamic Bulk Modulus): Used in applications involving high-rate, small-amplitude pressure fluctuations such as occur in servoactuators

b. Adiabatic (Isentropic) Secant (Dynamic Bulk Modulus): Used in applications involving high-rate, large-amplitude pressure excursions such as occur in hydraulic pumps

c. Isothermal Secant (Static Bulk Moduls): The simplest and most accurate method of measuring fluid compliance produces isothermal secant moduli. This modulus has limited use in the analysis of fast acting hydraulic systems.

The adiabatic tangent curves presented on Figure 2 were derived from pressure wave velocity tests conducted by Rockwell-Columbus for the Navy.

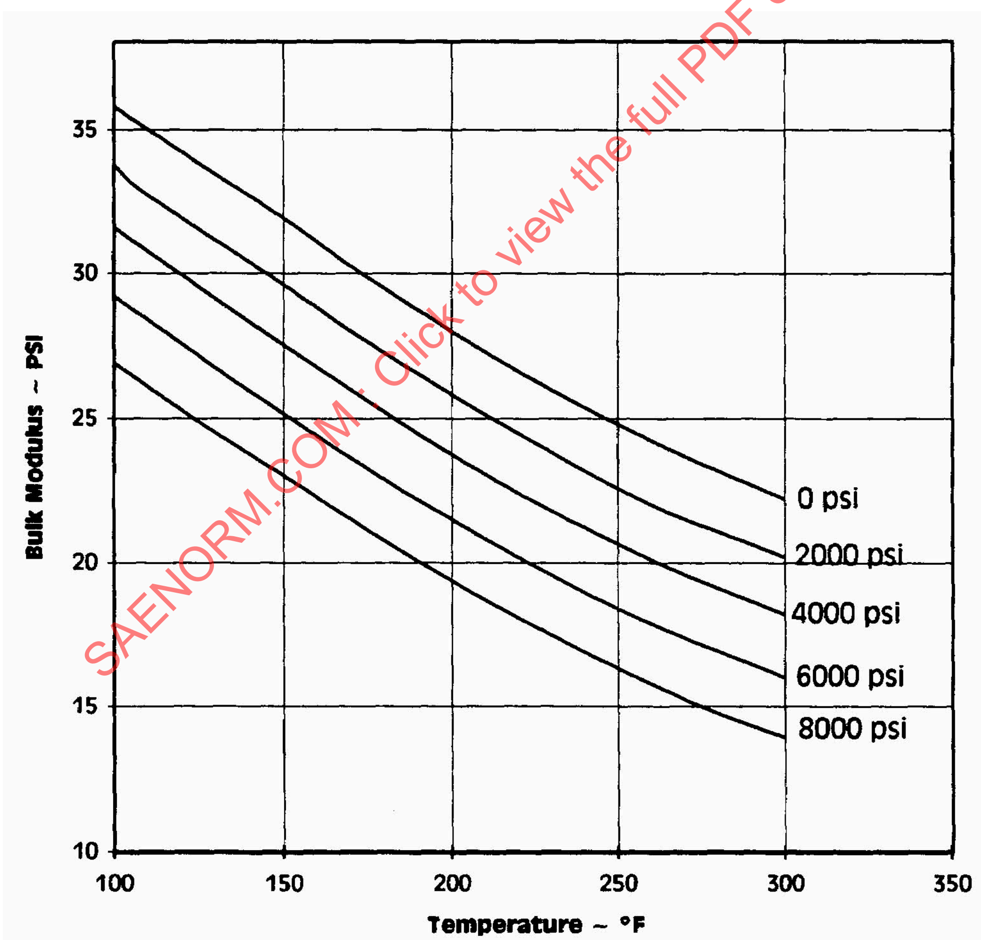


FIGURE 2 – Adiabatic Tangent Bulk Modulus of MIL-H-83282 Fluid

3.1.2.3 Mass Density: The mass density curves shown on Figure 3 were developed from published density versus temperature data and isothermal secant bulk modulus data.

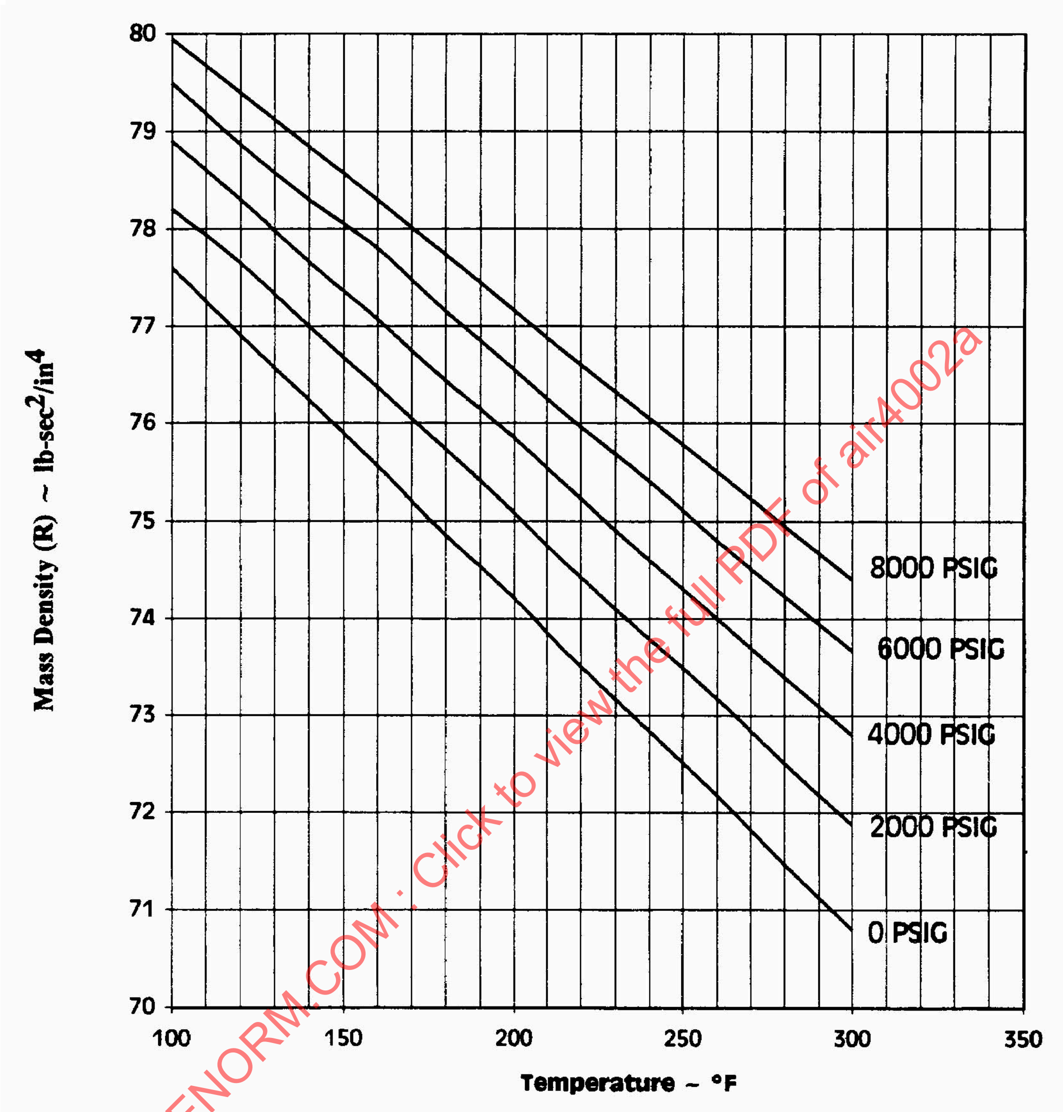


FIGURE 3 – Mass Density of MIL-H-83282 Fluid

3.1.3 CTFE Fluid: CTFE is a nonflammable hydraulic fluid currently being developed by the Air Force Wright Aeronautical Laboratories (AFWAL) Materials Laboratory. There are no flash and fire points for this fluid. Although the fluid does have an autoignition temperature (+1170 °F), it is considered essentially nonflammable. This feature is particularly important for aircraft safety and survivability. Other advantages include inertness and excellent shear stability. Major concerns are high density and compatibility with seal materials. CTFE fluid is still under development by AFWAL.

3.2 The fully formulated CTFE fluid most commonly used at this time (A0-2) is the Halocarbon Products Corporation CTFE base stock SAFETOL 3.1 plus a 3M lubricity additive and a barium sulfonate rust inhibitor additive. The AFWAL Materials Laboratory is responsible for formulating this CTFE fluid. Fluid viscosity, density, and bulk modulus data are shown on Figures 4, 5, and 6, respectively. This data was generated by McDonnell Aircraft Company. Additional CTFE fluid property data is found in Reference 32 prepared under AFWAL contract by the Boeing Military Airplane Company.

4. SEAL DEVELOPMENT:

4.1 8000 psi MIL-H-83282 Fluid System Seals - Navy:

4.1.1 Rod Seals: The proper selection of actuator rod seals is recognized as being extremely critical to the successful demonstration of component and system reliability in 8000 psi, MIL-H-83282 fluid systems. The rod seals in contemporary aircraft are the most likely source of external leakage. External leakage is the most frequent cited cause of actuator removals.

4.1.1.1 Vought conducted a rod seal test program that was 410.2 h in duration. The following conclusions were reached:

a. Reliable long-life rod seals can be attained for 8000 psi systems.

b. Rubber against the rod wears well if protected properly. The backup ring and the extrusion gap are key elements in determining elastomer wear. The extrusion gap is one of the most powerful factors influencing seal life in a 3000 psi system and is even more important in an 8000 psi system. The range of 0.001 to 0.003 in for diametral clearances may be too large. The results seem to indicate that much longer life can be achieved if the extrusion gap is held to 0.002 in or less.

c. The cap strip must be relatively thick to provide acceptable wear at 8000 psi.

d. TFE based seals leak more than rubber sealing elements. TFE seals may meet the 25-cycles/drop requirement, but all the really dry seals had rubber in contact with the rod.

e. Bronze-filled TFE cap strips did not wear as well as some other materials. This may have been due to insufficient thickness of the cap seal, but appears to be also attributable to the material.

f. A two-stage, unvented seal can reverse pressurize the first-stage seal; therefore, this should be considered in selecting the first-stage configuration. Select a first-stage seal with extrusion resistance in the reverse direction. A unidirectional seal for the first-stage seal seems to add life to the second-stage seal.

g. The no-backup width cap strip is more stable than a one-backup width cap strip.

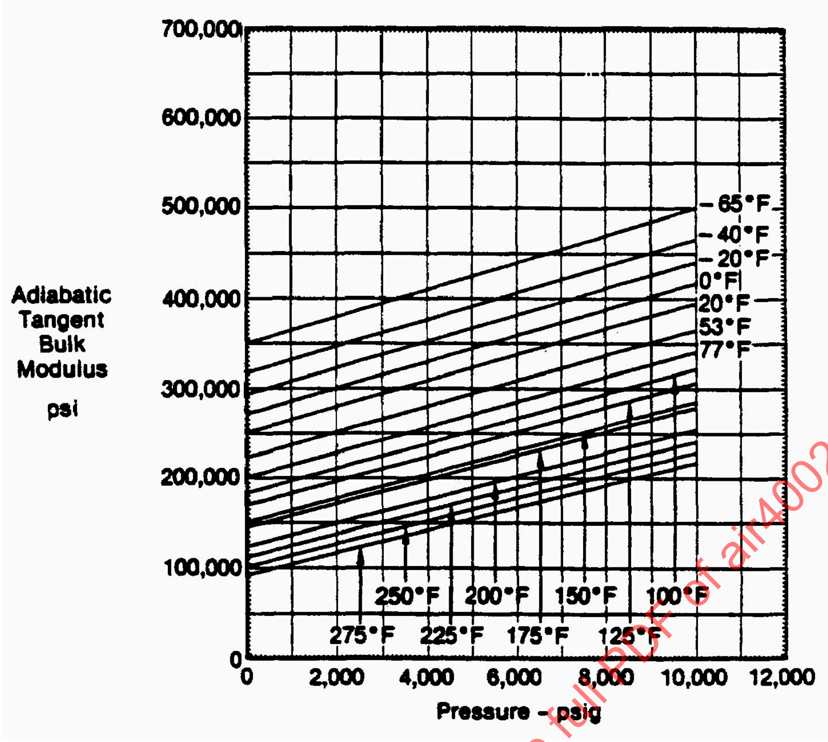


FIGURE 4 – A0-2 CTFE Base Fluid Bulk Modulus Versus Pressure

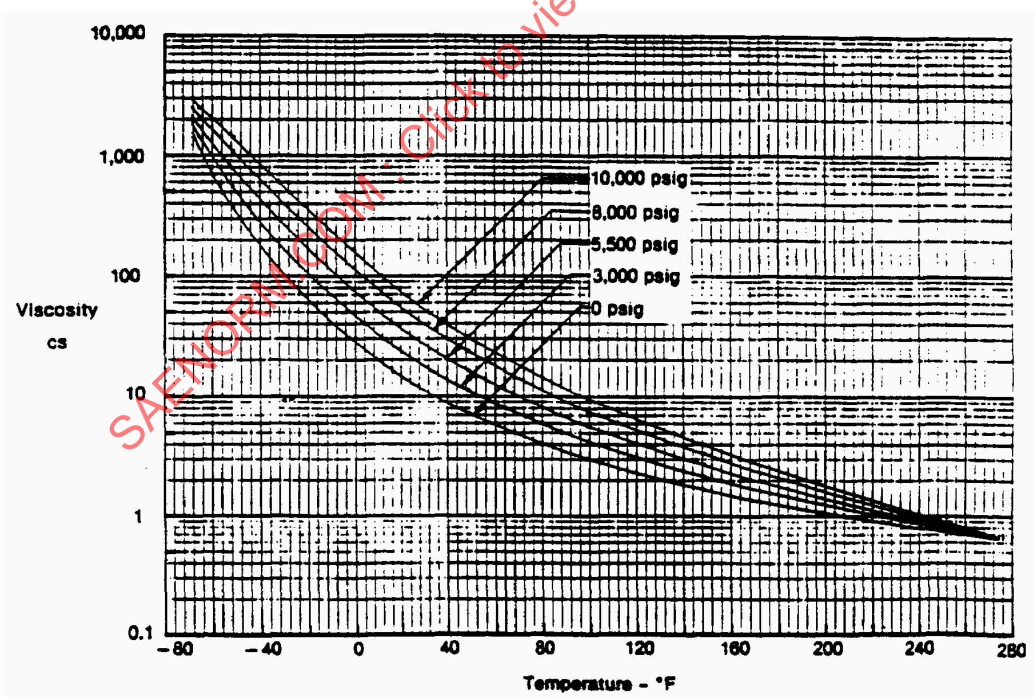


FIGURE 5 – A0-2 CTFE Base Viscosity Versus Temperature

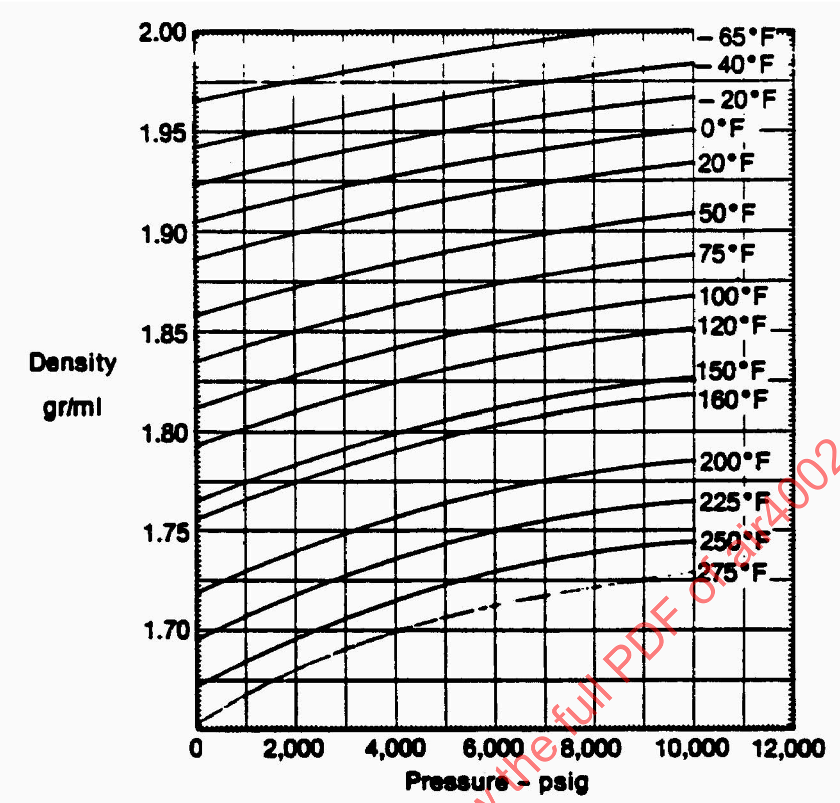


FIGURE 6 – A0-2 CTFE Base Density Versus Pressure

4.1.1.1 (Continued):

h. Glass/moly-filled backups and cap strips did not cause rod scoring. Most of the test was with long-stroke cycling which may have been a factor (not recommended).

i. On excluder-type rod scrapers, the curling up of the scraping edge is a problem that limits effectiveness.

j. Uncut, filled backup rings can wear on the ID to larger size and no longer be effective in eliminating nibbling if the backup material is not somewhat compliant. A hard backup ring such as nylon against the rubber is not compliant enough to keep the extrusion gap closed. In the "T" configuration, rapid nibbling occurred.

k. Short-duration testing of spiral backups indicated the thin member tends to extrude.

l. With one exception, all elastomeric seals were made from MIL-P-83461 compound. No problem occurred with this compound.

m. No problem occurred with the MIL-H-83282 test fluid.

4.1.1.2 Rod seal configurations recommended as candidates for 8000 psi actuators were as follows in Table 2 and Figure 7:

TABLE 2 – Recommended Rod Seal Configuration

Application	Seal Configuration	Figure	Comments
Primary Flight Control Actuators	Two-stage seal: 1st stage cap seal with backup ring on each side 2nd stage O-ring with two backups on low pressure side	7	Can be installed in standard unsplit groove Provides dual seal redundancy Rubber outer seal assures dryness Subject to erroneous installation.[1]
Automatic Flight Control System Actuators	Single-stage seal cap seal with two backups on low-pressure side	8	Need to keep friction low
Utility Actuators	Single-stage seal: or standard O-ring with trapezoidal backup	9	Utility application has limited life requirement Rubber outer seal assures dryness

[1]The second-stage seal must be carefully inspected to ensure that the backup rings are installed on the low pressure side.

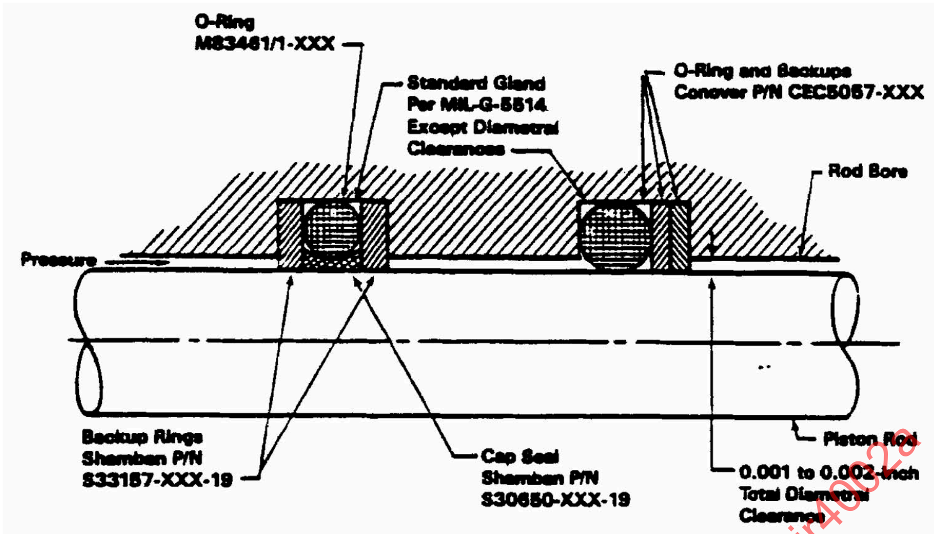


FIGURE 7 – Rod Seals in Flight Control Actuators

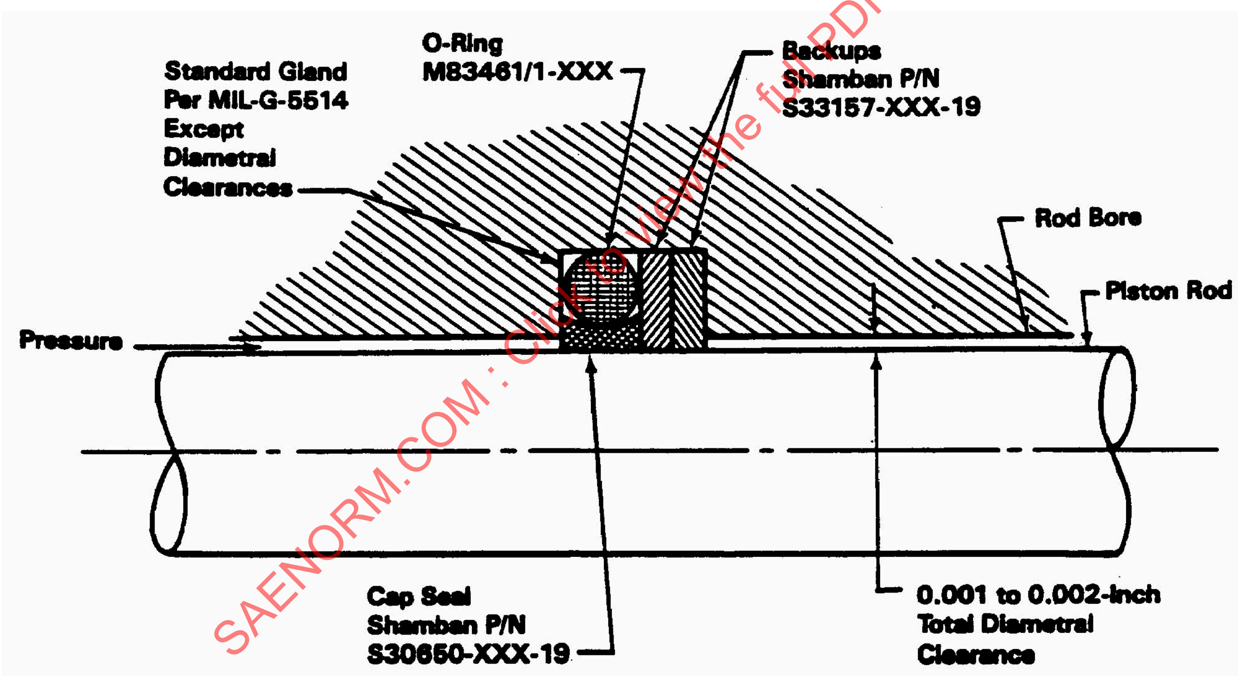


FIGURE 8 – Rod Seals in AFCS Actuators

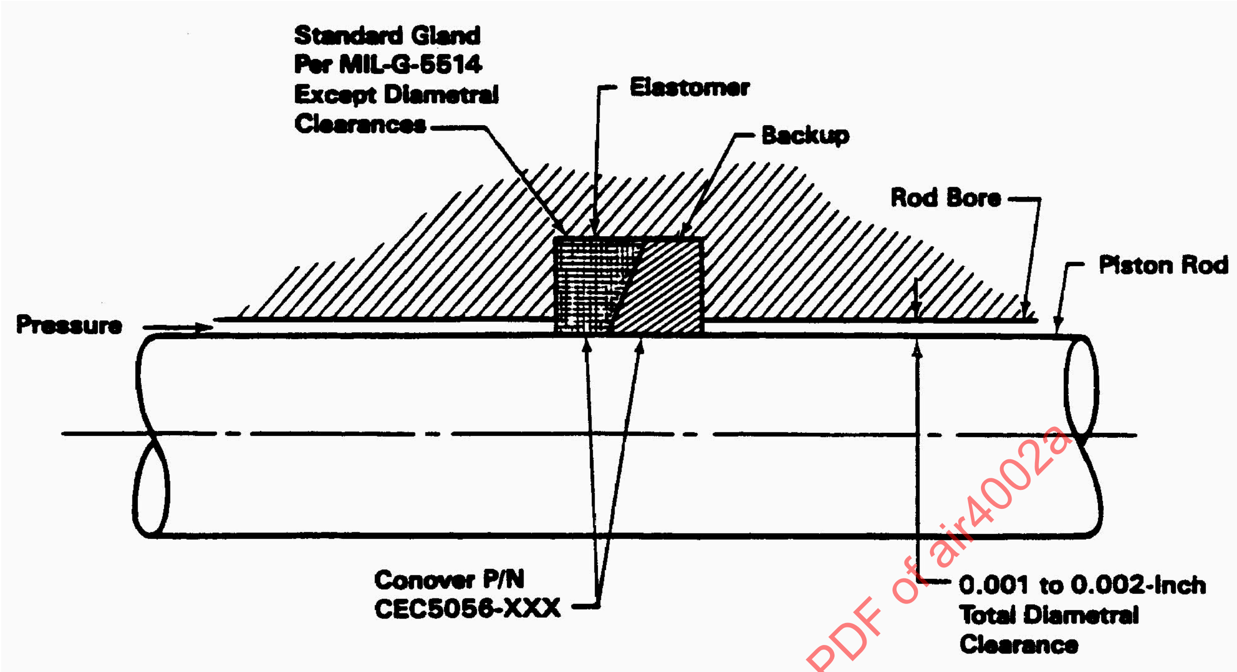


FIGURE 9 – Rod Seals in Utility Actuators

4.1.2 Piston Seals: The initial selection of piston seals was based upon extended endurance tests conducted by Rockwell. A Greene, Tweed "T" seal configuration initially performed very well. However, subsequent testing showed that the metal backup rings would wear out and cause scoring of the cylinder barrel. The metal backup rings were replaced with nylatron backup rings, Figure 10. This seal is used in the LH-UHT (FC-2), AFCS, RFI, spoiler, and rudder (FC-2) actuators on the Navy Lightweight Hydraulic Simulator.

4.1.2.1 Another seal candidate made by Shamban, performed very well, Figure 11. It is used on the RH-UHT actuator. The cap seal for this installation has heavy-duty width noncut backup rings.

4.1.2.2 Piston seals used in the rudder (FC-1), aileron, LH UHT (FC-1) actuators are the Shamban "double delta" configuration.

4.1.2.3 A summary of the total number of cycles accumulated during the Phase I and Phase II LHS program on the Navy LHS simulator is shown in Table 3.

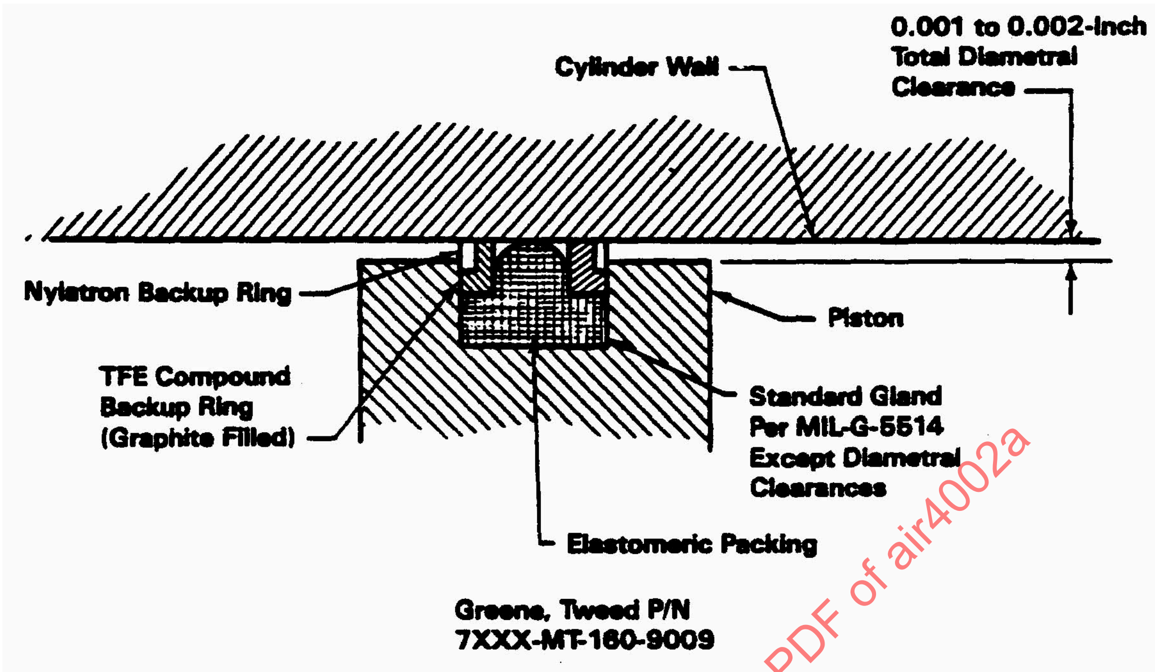


FIGURE 10 – Piston Seal - Greene, Tweed "T" Seal

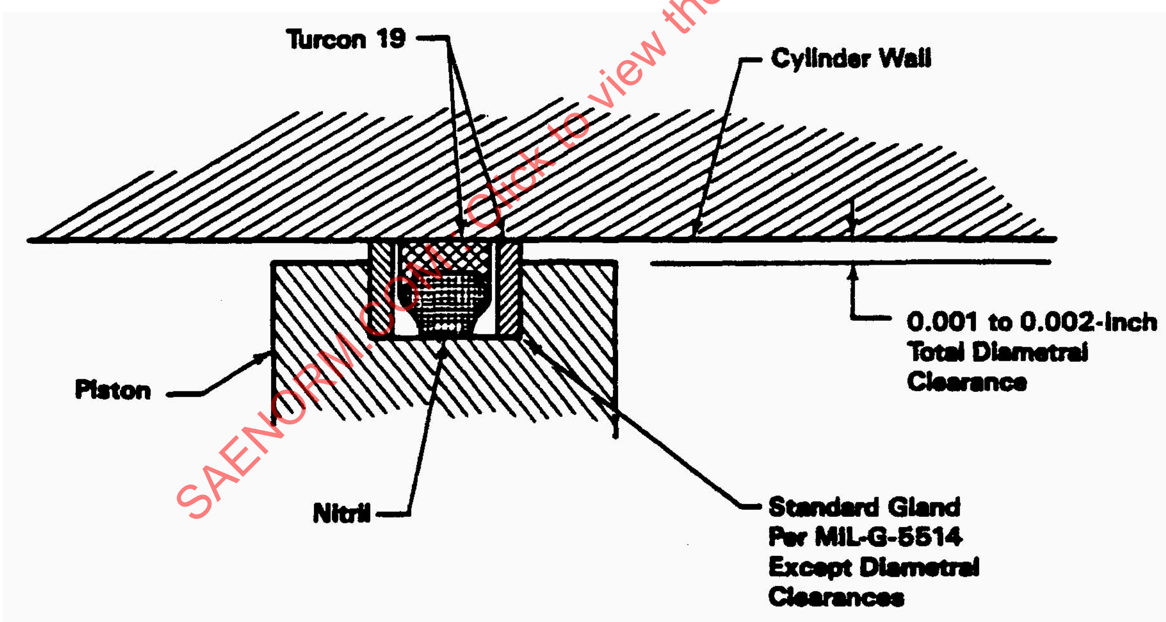


FIGURE 11 – Piston Seal - Shamban Plus Seal®II

TABLE 3 – Seal Test Summary - At 900 h of Operation LHS Simulator - Navy

	System	System	System	System	Cycles
Left Hand Unit	/1/	/2/			3 993 053
Horizontal Tail			/4/	/4/	
			/6/	/6/	1 163 042
Right Hand Unit	/3/	/3/	/5/	/5/	
Horizontal Tail			/6/	/6/	
Rudder	/1/	/6/			4 050 450
			/4/	/4/	
			/7/	/7/	
Aileron		/2/			3 979 620
				/4/	
				/6/	2 471 460
Spoiler	/2/	/2/			632 723
	Single Piston 2	Single Piston 2	Single Rod 4	Single Rod 4	1 349 350
					2 568 000

NOTES:

/1/ Piston Seal: Shamban Double Delta®II with solid backup ring, and standard O-ring

/2/ Piston Seal: Greene, Tweed "T" Ring, staged assembly, Nytril, P4, Nylatron backup ring

/3/ Piston Seal: Shamban Plus Seal®II, Turcon 19

/4/ Rod Seal: Shamban Double Delta®II, Channel Seal, glass-filled solid backup ring, standard O-ring

/5/ Rod Seal: Shamban Turcon®19, solid backup ring - Turcon 19

/6/ Rod Seal: Conover Revonoc, solid backup, standard O-ring

/7/ Rod Seal: Conover CEC 4862 - XXXNC, standard O-ring

4.1.2.4 The testing results to date indicate that long, seal life is achievable when tolerances for diametral clearances are held to 0.002 or less. Conventional rip-stop design is difficult to implement because tolerances will not permit the extrusion gap to be held to 0.002 in maximum. The conventional rip-stop configurations shown in Figure 12 are not recommended because of difficulty in maintaining the required seal clearances in the barrel.

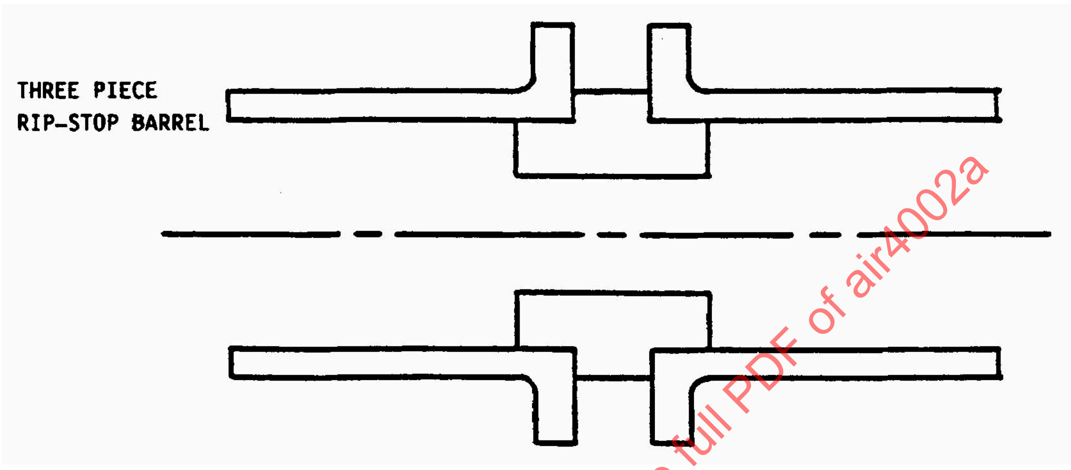


FIGURE 12 – Conventional Rip-Stop Configuration

4.1.2.5 The barrel should be machined from one piece as shown in Figure 13 to maintain the desired dimensions for rod seal clearance. An alternative rip-stop configuration is the three-piece brazed-assembly shown on Figure 14. The barrel is machined after assembly to maintain tolerances.

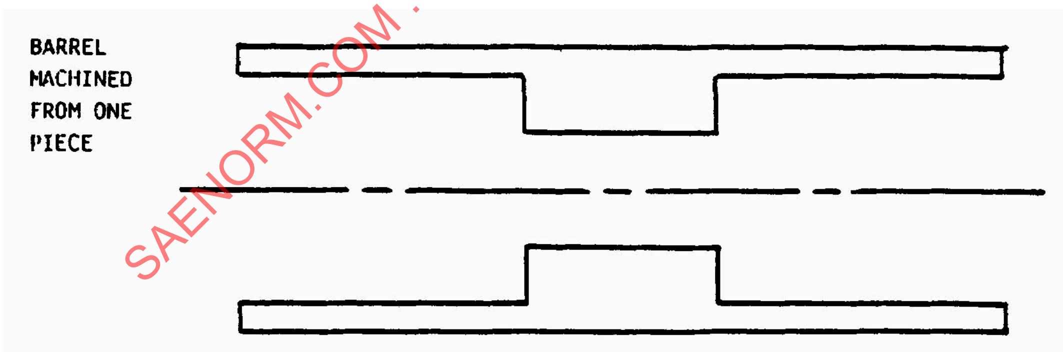


FIGURE 13 – One Piece Machined Rip-Stop Configuration

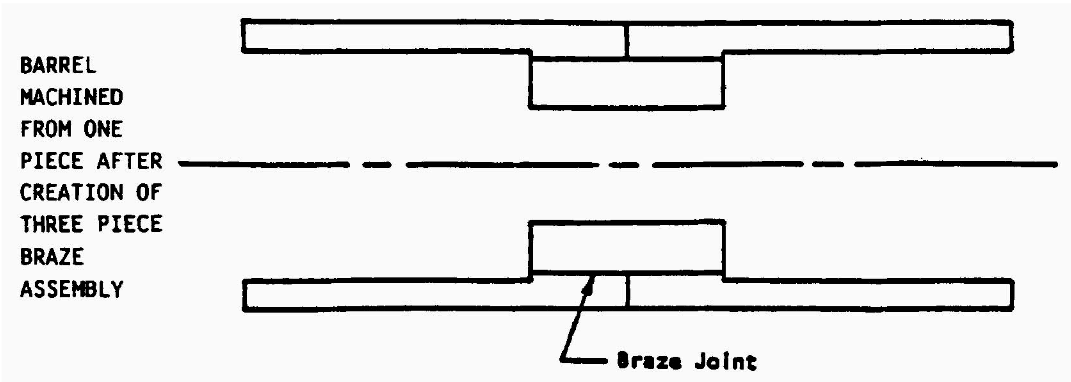


FIGURE 14 – Three Piece Assembly Rip-Stop Configuration

4.2 8000 psi CTFE Fluid System Seals - Air Force:

4.2.1 The Air Force and McDonnell Aircraft Company have been engaged in an effort to eliminate aircraft hydraulic fluid fires by using nonflammable CTFE fluid. The use of CTFE fluid has created new areas of concern in the area of seals. This is mainly due to the incompatibility of the CTFE fluid with common seal system materials such as nitrile rubber. The following initial guidelines were established for seal and gland designs:

a. Seal Materials

(1) All seals will be of phosphonitrilic fluoroelastomer (PNF) compound and backup rings will be manufacturers' recommended material unless otherwise noted. All backup rings will be uncut except where not feasible.

(2) Static seals will be MS28775/MS28774 configuration.

(3) Boss seals will be MS28776.

b. Tolerances for seal extrusion gaps

(1) Tolerances on pistons, rods, and piston bores were established to provide a diametral clearance of 0.001 to 0.002 in.

(2) Cylinder breathing was established not to exceed 0.001 diameter at midstroke with 8000 psi applied.

c. Seal gland dimensions were per MIL-G-5514 with the following exceptions:

(1) Gland inside corners not to exceed 0.035 in.

(2) Outside corner edges were broken not to exceed 0.007 in.

4.2.1 (Continued):

(3) Vertical angles were 0° ± 1/2.

(4) The gland depth was controlled to give a minimum of 5% squeeze on the seal.

(5) All rod seal groove widths were the same (two backup widths).

NOTE: Switched to green PNF (80 durometer) from original black PNF (70 to 75 durometer) and added Viton® and ethylene propylene diene monomer (EPDM) rubber.

The seal test completed 750 h of endurance testing at –65 to 275 °F. The testing included 3 000 000 stabilator cycles and 30 000 trailing edge flap cycles. A seal test results summary and the number of cycles at teardown is shown in Table 4.

4.2.2 A summary of the elastomer/CTFE experience/compatibility is shown below:

a. “Green” PNF (80 Durometer) - Unacceptable, cannot tolerate dither cycles, independent of fluids.

b. Modified PNF (70-75 Durometer) - Generally acceptable, may have low temperature problems, gets soft.

c. EPDM - Acceptable.

d. Viton® - Acceptable, has expected leakage below –40 °F.

e. Buna N - Unacceptable, poor compatibility.

f. EPDM/PNF - Insufficient experience to evaluate.

The above seal tests should be considered as preliminary. Further tests are being conducted on an ongoing basis by many companies using improved designs and materials. The results of these tests will be added to this document when they become available.

TABLE 4 – Seal Test Summary - Air Force

	Cycles	Comment Number
Stabilator Actuator		
Bertea Torlon Piston Seal	2 000 000	1
Shamban EPDM/PNF Piston Plus Seal®II	416 000	2
Shamban Plus Seal®II EPDM Center Gland		
- Primary	2 116 000	3
- Secondary	2 116 000	3
Shamban Double Delta®II With Bertea Nylon Backup On Tail Rod (Primary)	2 116 000	3
Conover Conohex EPDM Rod End Seals		
- Primary	416 000	4
- Secondary	2 000 000	4
Conover Conohex Center Gland Seal (Secondary)	2 116 000	4
Conover Sandwich EPDM On Tail Rod (Secondary)	2 116 000	4
BAL Center Gland Seal (Primary)	3 000 000	5
Trailing Edge Flap Actuator		
Bertea Torlon Piston Seal	30 000	6
Conover Trapezoid Rod Seal	30 000	6
Transfer Tube Seals	2 080	7
Static Seals		8
NOTE: Fluid is CTFE		

NOTES:

1. Torlon material provided excellent wear life.

   Had problems with keeping the torlon rings oriented correctly on the metal energizer ring. Rings would rotate over the spacers welded to the metal ring resulting in broken torlon rings.

   This seal worked the best of the piston seals and, with design improvements, shows high potential for future applications.

2. After 270 h of endurance testing, the actuator was disassembled because of high leakage in the actuator. When disassembled, the seal had disintegrated. It was replaced with an identical configuration for the remainder of the test.

3. The primary and secondary seals were replaced once. Backup rings were also replaced once.

   Hatseal®IIs were used at the rod end. Replacements were made twice after a short number of hours because the seals had deteriorated.

   Double Delta®II seal on tail rod performed well. Bertea nylon backup ring was used.

4. Conohex seal performed well.

   Primary rod end seal was replaced two times during the 750 h test because it showed severe wear. However, this seal was in a position subjected to side loading and no external leakage occurred.

5. Seal (BAL) went entire test with no failure in its position as a primary high pressure dynamic seal.

6. Conover trapezoid rod seal and Bertea torlon piston ring went through entire test with no apparent problems.

7. Biggest problem was the transfer tube seals. The elastomers were nibbled away, eventually creating leakage and were replaced.

8. Materials

   a. Viton® - Performed well in the pump with no problem.

      Conover -50 °F Viton® performed well. Used in the stabilator center dam area for performance testing. Leakage occurred at -65 °F.

   b. Conover Modified PNF - Performed well in the stabilator valve.

   c. EPDM - Overall performed very well. Did not show the tendency to deteriorate as did green PNF.

   d. General Comments - Standard teflon backup rings showed a tendency to feather rapidly, and several replacements were made. Hard teflon backup rings (Conover) worked much better.

5. 8000 psi HYDRAULIC SYSTEMS - GENERAL:

5.1 Work on the concept of using very high hydraulic system operating pressures in aircraft was begun with a theoretical analysis of factors associated with the development of very high pressure fluid power systems. Subsequent studies showed that significant weight savings could be gained by operating at 8000 psi instead of the conventional 3000 psi level. Work progressed in logical phases from theoretical considerations to evaluation of experimental hardware to endurance testing and limited flight testing. A brief review covering the state-of-the-art of 8000 psi system development is presented in the following sections. Areas to be discussed include: (1) analysis and design and (2) hardware development.

5.2 Analysis and Design:

Several system material properties and operating characteristics are affected by operating pressure level; in particular, fluid viscosity, actuator stiffness, pressure surges, and heat generation. The degree to which 8000 psi (versus 3000 psi) affects these parameters and, thus system performance, is a primary concern. Theoretical analyses were made in 1966 (Reference 1 in Appendix A). Since then, many hours of laboratory testing and investigation have provided new insights into the effect of high pressure on these parameters.

5.2.1 Fluid Viscosity: Tubing pressure losses and component internal leakage rates are directly related to fluid viscosity. Temperature has a marked influence on fluid viscosity; pressure has a less pronounced effect (the kinematic viscosity of MIL-H-83282 fluid is about 1-1/2 times greater at 8000 psi than at 3000 psi). Classic fluid flow theory for pressure drop, orifice flow, capillary flow, etc., is applicable at 8000 psi. Thus, procedures used at 3000 psi to determine line losses, orifice size, and valve dimensions are also valid at 8000 psi.

5.2.2 Actuator Stiffness: Weight savings produced by operating at 8000 psi instead of 3000 psi begins with smaller net areas on actuator pistons. Lower flow demand because of less displaced fluid results in a general decrease in the size of supply lines, pumps, reservoirs, etc. However, use of a smaller piston area reduces actuator physical stiffness, which in turn lowers system resonant frequency. The following paragraphs will attempt to show that this reduction in actuator stiffness will not significantly degrade system performance.

5.2.2.1 Mechanical elements that contribute to physical stiffness include bearings, piston/rod, cylinder, and actuator end pieces; hydraulic elements are the fluid and seals. Based on practical experience, it has been found that actuator physical stiffness is approximately equal to the stiffness due to fluid bulk modulus as shown in Figure 15.

$$Kf = \frac{4\beta_f A\eta}{S}$$

Where, Kf = Stiffness due to fluid compressibility (actuator piston at mid-stroke)

ßf = Bulk modulus of fluid (taken at one-half system pressure, approximately 15% higher at 4000 psi than at 1500 psi)

A = Piston net area

S = Piston stroke (total)

η = Ratio of fluid volume swept by the piston to the total fluid volume contrained between the piston and control valve, typically between 0.85 and 0.95 for an integrally mounted valve.

FIGURE 15 – Actuator Physical Stiffness

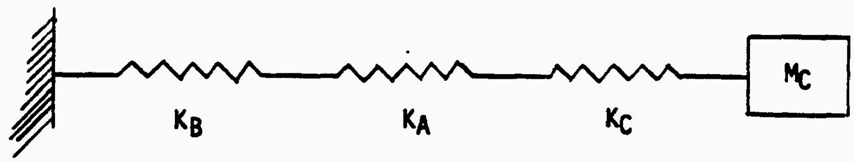
5.2.2.2 In a practical hardware installation the actuator backup structure, the actuator, and the control surface are three significant springs (Figure 16). Control surface inertia is the only significant mass. The three springs are in series, anchored to the aircraft at one end and supporting the mass at the other. If the spring rates of each are assumed equal, then the actuator is twice as stiff as the combined structure/surface spring. Normally, however, the actuator is the stiffest spring in the system. This is shown in Figure 16.

5.2.2.3 System natural frequency is established by net total stiffness and varies, not directly, but with the square root of the stiffness as shown in Figure 17.

5.2.2.4 Physical stiffness is the composite effect of mechanical and hydraulic compliant elements between the actuator mounting points. Actuator functional stiffness is due to closed loop servoaction and is related to loop gains and control valve performance characteristics. In a conventional position feedback system, functional static stiffness is higher than physical stiffness.

5.2.2.5 Frequency response tests of similar sized “muscle” actuators operating at different pressure levels were conducted in a lumped mass load fixture.

5.2.2.6 Performance characteristics of the 6000 and 9000 psi actuators were very similar to the 3000 psi actuator for large-amplitude, manual type inputs. For small-amplitude inputs, such as those encountered with automatic control, response capability of the 6000 and 9000 psi actuators were satisfactory to 10 Hz. The reduction in resonant frequency noted previously is, therefore, not considered critical.



Where, K_B = Backup structure spring rate
K_A = Actuator spring rate
K_C = Control surface spring rate
M_C = Control surface mass

Now,
$$\frac{1}{K_T} = \frac{1}{K_B} + \frac{1}{K_A} + \frac{1}{K_C}$$

Where, K_T = Net total spring rate

Now let
$$\frac{1}{K_{BC}} = \frac{1}{K_B} + \frac{1}{K_C}$$

If, $K_A = K_B = K_C$

Then, $K_A = 2\ K_{BC}$

FIGURE 16 – Spring Rates

$$W_n = \sqrt{\frac{K_T}{M_e}}$$

Where, W_n = System undamped natural frequency

K_T = System net spring rate

M_e = Effective mass

FIGURE 17 – System Natural Frequency

5.2.2.7 The majority of conventional 3000 psi aircraft actuators are not stiffness critical. However, design approaches for the actuators that are stiffness critical are:

a. Size actuator piston based on stiffness and increase the size of the distribution system to provide associated higher flow requirements.

b. Apply enhanced dynamic stiffness techniques as demonstrated by the McDonnell Douglas Corporation (Reference 37 in Appendix A).

5.2.3 Pressure Surges: Pressure surges are normal in aircraft hydraulic systems and are an important design parameter because of their effect on the fatigue and functional characteristics of system components. Surges result primarily from:

a. Sudden stopping of high velocity fluid

b. Sudden porting of high pressure fluid into a chamber filled with a low pressure fluid

c. Bottoming of an actuator piston

d. External energy derived from load inertia

When the flow of a mass of fluid is suddenly decelerated by a rapidly closing valve, water hammer results. Assuming instantaneous valve closure, this surge may be calculated by Equation 1:

$$\Delta P = v\sqrt{\rho\beta e} \qquad \text{(Eq. 1)}$$

where:

ΔP = Maximum pressure rise above system pressure

V = Fluid velocity

ρ = Fluid mass density

βe = Effective bulk modulus (fluid compressibility + tube elasticity)

5.2.3.1 Pump response time is also a factor causing surges. Operation of the delivery control mechanism normally occurs in 0.050 s or less. Thus, when a valve closes, the pump momentarily continues to discharge fluid until the control mechanism adjusts to the new flow demand; this can result in a pressure overshoot.

5.2.3.2 Surges in 8000 psi systems are less, percentage wise, than in 3000 psi systems because of:

a. Better damping at 8000 psi due to increased fluid viscosity

b. The minor effect of operating pressure level on water hammer magnitude

c. Faster pump response at 8000 psi

5.2.3.3 Typical peak surges observed in 3000 and 8000 psi systems compare as follows:

a. 3000 psi: 3900 psi (130%)

b. 8000 psi: 9100 psi (114%)

The maximum allowable surge in 3000 psi systems is 135%, reference MIL-H-5440E. It is proposed that the maximum allowable surge in 8000 psi systems be limited to 120%. The validity of the 120% design value has been confirmed many times by laboratory testing.

5.2.4 Heat Generation: Hydraulic systems generate heat because it is impossible to convert all input power into useful work. Thus, all hydraulic systems normally operate at temperatures above ambient. Temperature stabilization is reached when the heat loss rate equals the generation rate. Hydraulic fluid temperatures must be maintained below stated maximums to prevent thermal breakdown of the fluid and seals. For Type II systems this temperature is 275 °F; for Type III systems it is 390 °F per MIL-H-8891A. If heat dissipation through conduction, radiation, and convection is not sufficient to maintain reasonable fluid temperatures, then a heat exchanger is required. A hydraulic system must be designed so that a heat balance is achieved at a satisfactory operating temperature.

5.2.4.1 The principal sources of heat generation in hydraulic systems are as follows:

a. Pump and valve internal leakage

b. Orifices and valves used to throttle and control flow (these devices are inherent heat generators)

c. Resistive pressure drops in lines, fittings, and porting passageways

5.2.4.2 The principal means of heat dissipation are as follows:

a. Conduction from hydraulic system components through attachments into aircraft structure

b. Convection aided by airflow around system components

c. Radiation from system components

d. Heat exchangers (i.e., fuel-oil, air-oil)

5.2.4.3 In aircraft where variable delivery pumps drive servoactuator systems, nearly all input power is eventually dissipated as heat (under steady state operating conditions). When a system has no work output, then equilibrium is attained when system temperature is high enough above ambient to cause a heat transfer rate equal to the energy input rate at the pump. Thus, see Equation 2 as follows:

$$W_{in} = Q_{loss} = UA\left(T_{sys} - T_{air}\right) \quad \text{(Eq. 2)}$$

where:

W_{in} = Work input

Q_{loss} = Heat loss

U = Overall coefficient of heat transfer

A = Surface area of system

T_{sys} = System temperature (average)

T_{air} = Ambient air temperature

5.2.4.4 Since operating temperatures are related directly to the surface area of a hydraulic system, cooling requirements will be somewhat greater at 8000 psi due to the inherent compactness of the system (assuming 3000 psi and 8000 psi pump efficiencies are the same). Therefore, 8000 psi systems must be designed to operate at slightly higher ΔT allowables or provide supplemental cooling via heat exchangers. Ambient air temperatures in the stratosphere is –70 °F. Aerodynamic heating increases the air temperature to 70 °F at Mach 1.4 and to 390 °F at Mach 2.5. Hydraulic lines will not be cooled, but rather heated by air at higher Mach numbers. Reducing heat generation and increasing cooling by fuel-oil heat exchangers becomes necessary.

6. PRESSURE LEVEL SELECTION CRITERIA:

6.1 This section examines criteria used by Rockwell to select the 8000 psi operating pressure level used for the Navy LHS Program.

6.2 Pump Performance:

6.2.1 Pressure compensated variable volume pumps have three operating modes: (1) full discharge flow, (2) full cutoff (zero discharge flow), and (3) operating along the cutoff slope which produces a median flow. Pump efficiency normally varies from zero at full cutoff to some value near 90% at full flow. Full flow is developed when the discharge pressure drops approximately 200 psi below full cutoff pressure. Data analyzing pump performance at full flow and full cutoff will be examined.

6.2.2 Pump input power at full cutoff is shown on Figure 18. The curves indicate that, based on input power requirements at full cutoff, the optimum discharge pressure is 7350 psi.

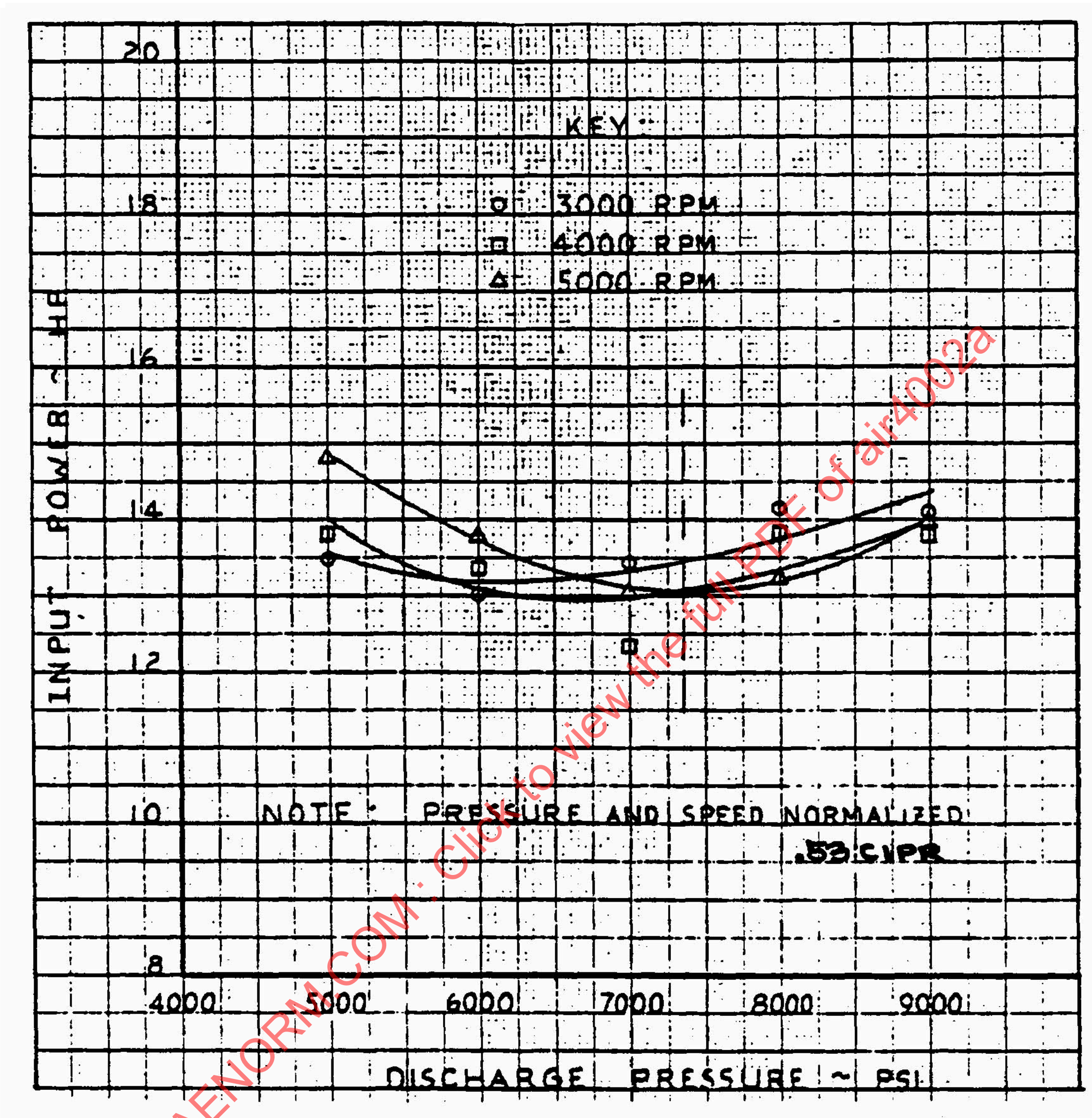


FIGURE 18 – Pump Input Power at Full Cutoff

6.2.3 Figure 19 shows pump power loss at full flow (200 psi below full cutoff pressure). Power loss is given as a percent of the generated output power. These curves indicate that the best operating pressure level is approximately 6200 psi. Since the pump was designed for optimum performance at 6000 psi, this data confirms the design. Optimum performance could be achieved at any pressure level up to 9000 psi by retiming the pump. This procedure would involve pump disassembly and minor rework.

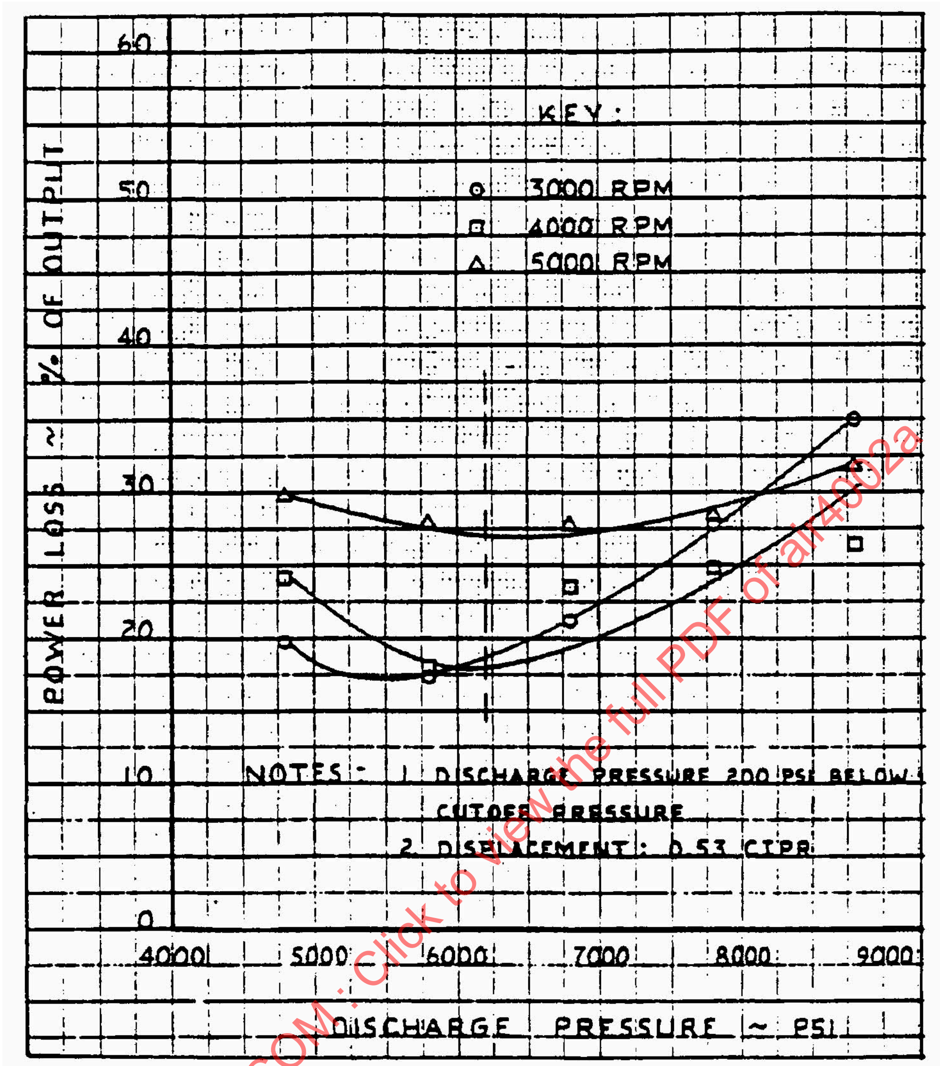


FIGURE 19 – Pump Power Loss at Full Power

6.2.4 Aircraft pumps operate approximately 90% of the time producing only sufficient flow to overcome system internal leakage (generally less than 10% of pump capacity). In view of this, and the information discussed previously, the most favorable discharge pressure level for the pump was judged to be approximately 7500 psi.

6.3 Power Transmission Systems:

6.3.1 Fluid velocity limitations for aircraft with 3000 psi hydraulic systems are established in MIL-H-5440. Prior to release of the 'E' version in 1968, fluid velocity was limited to a maximum of 15 ft/s. This was based on accepted levels of fluid friction loss and pressure surge generation. Maximum fluid velocity permitted by MIL-H-5440E is not specifically stated; instead, the requirements are based on obtaining satisfactory system performance. Results of the surge tests conducted by Rockwell indicate that fluid velocities of 25 ft/s can be used if the operating pressure level is over 6000 psi. This fact, plus the previously accepted frictional losses at 15 ft/s in a 3000 psi system, were used to establish the criteria for selecting the operating pressure level of very high pressure transmission lines.

6.3.2 The percent of transmitted tubing power lost versus fluid velocity for five line sizes is shown in Figures 20 through 24. In each of these figures, line sizes and wall thickness are based on using random available tubing (i.e., steel, aluminum) in 1972 time frame. A line of constant power loss was drawn horizontally through the intersection of the 15 ft/s and 3000 psi lines. The intersection of the constant power loss line with the 25 ft/s line established the operating pressure level.

6.3.3 The average very high pressure operating pressure level for tubing is 8000 psi. At 8000 psi: (1) power losses with 25 ft/s fluid velocity are no greater, percentage-wise, than losses with 15 ft/s at 3000 psi; and (2) pressure surges developed at a fast closing solenoid valve will be less, percentage wise, than the surge generated with 15 ft/s at 3000 psi. Therefore, 8000 psi was selected as the operating pressure level for transmission lines.

6.4 Pressure Level Selection:

6.4.1 Results of the foregoing analysis are summarized in Table 5. The relative importance percentage is based on Rockwell's judgement of the importance of each factor relative to the overall problem. Using this number to weigh the influence of each parameter, the most advantageous pressure level was determined to be 8075 psi. The operating pressure level for aircraft hydraulic systems was, therefore, chosen to be 8000 psi.

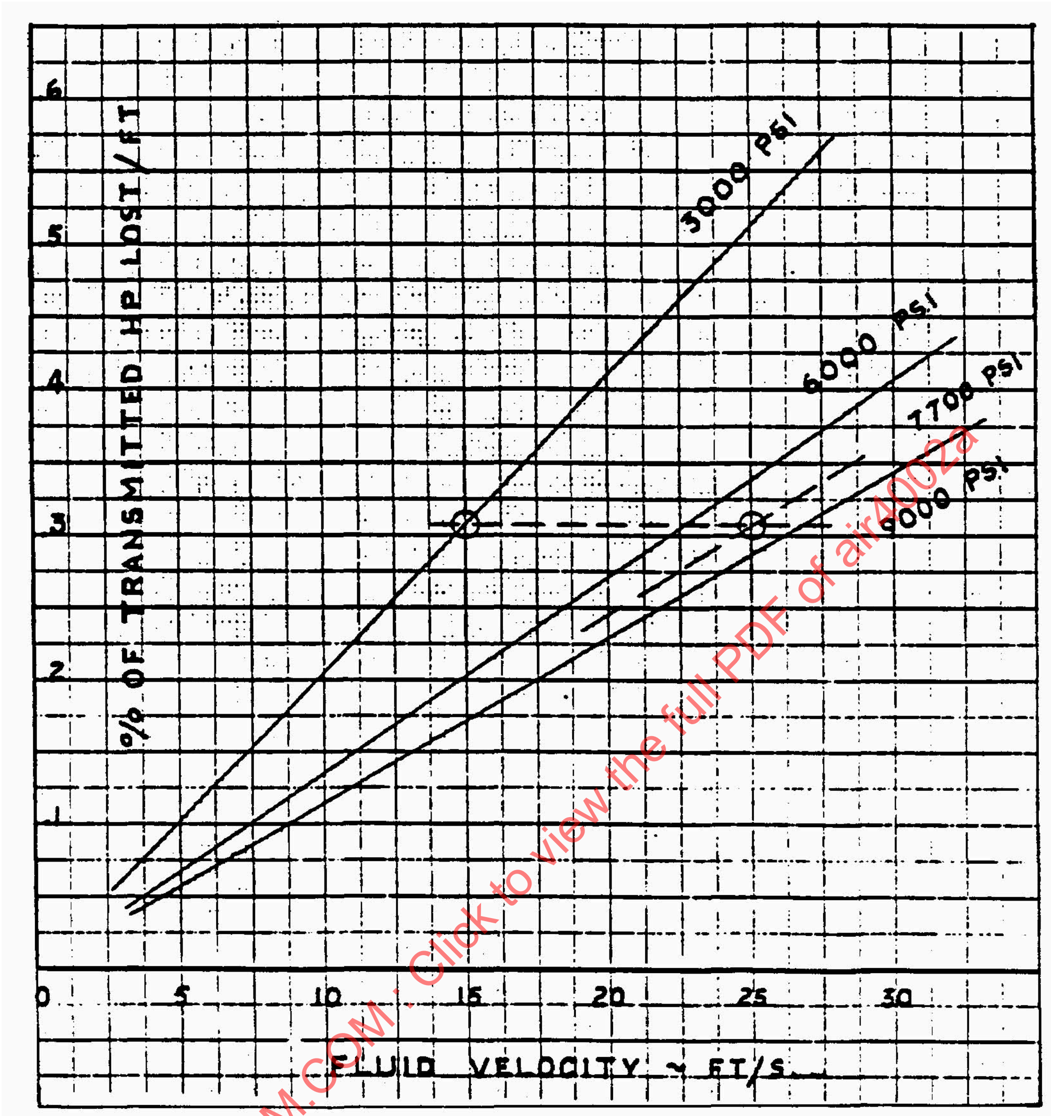


FIGURE 20 – Transmitted Power Loss Versus Fluid Velocity - 3/16 x 0.028 Tubing

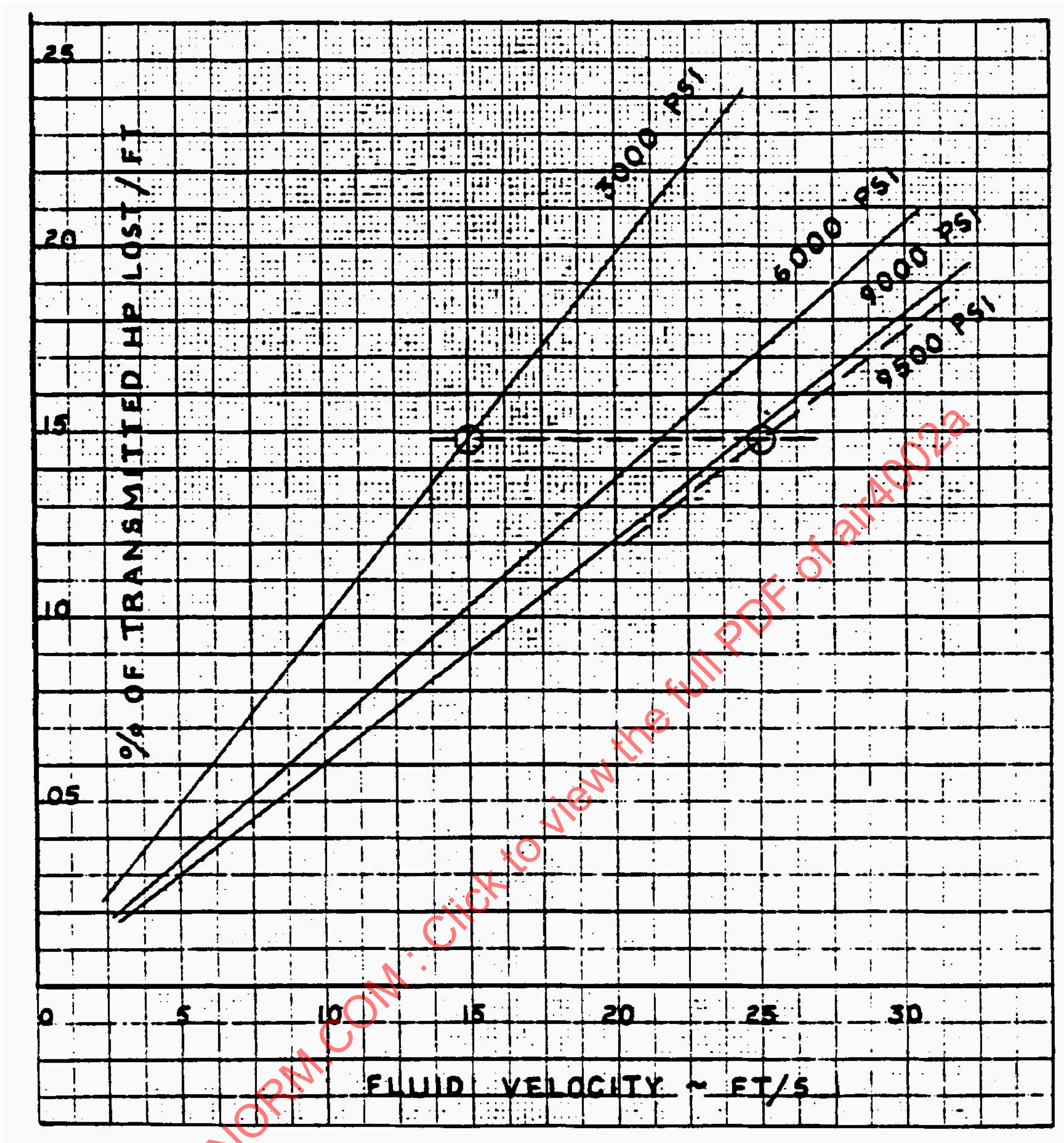


FIGURE 21 – Transmitted Power Loss Versus Fluid Velocity - 1/4 x 0.035 Tubing

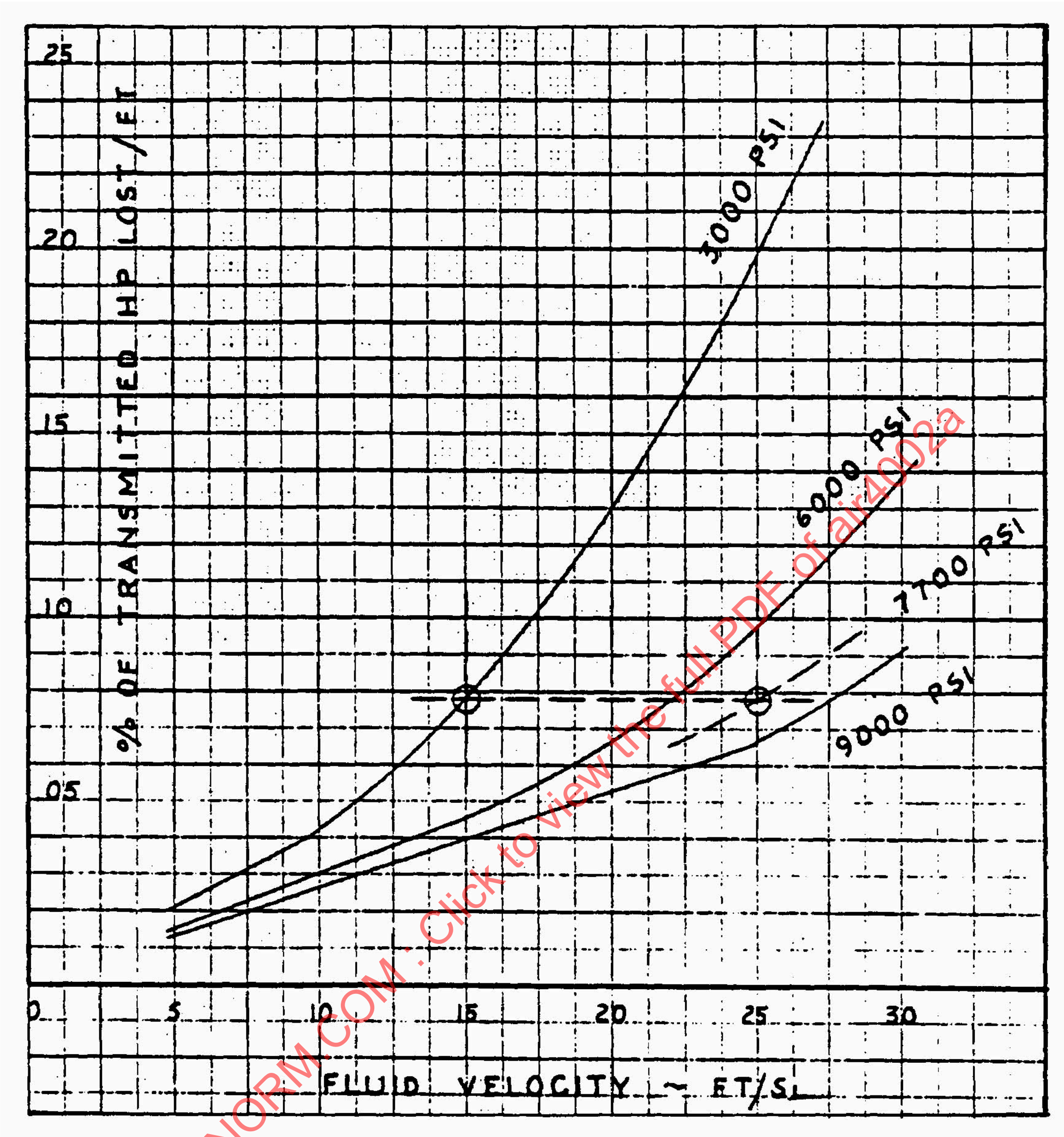


FIGURE 22 – Transmitted Power Loss Versus Fluid Velocity - 3/8 x 0.049 Tubing

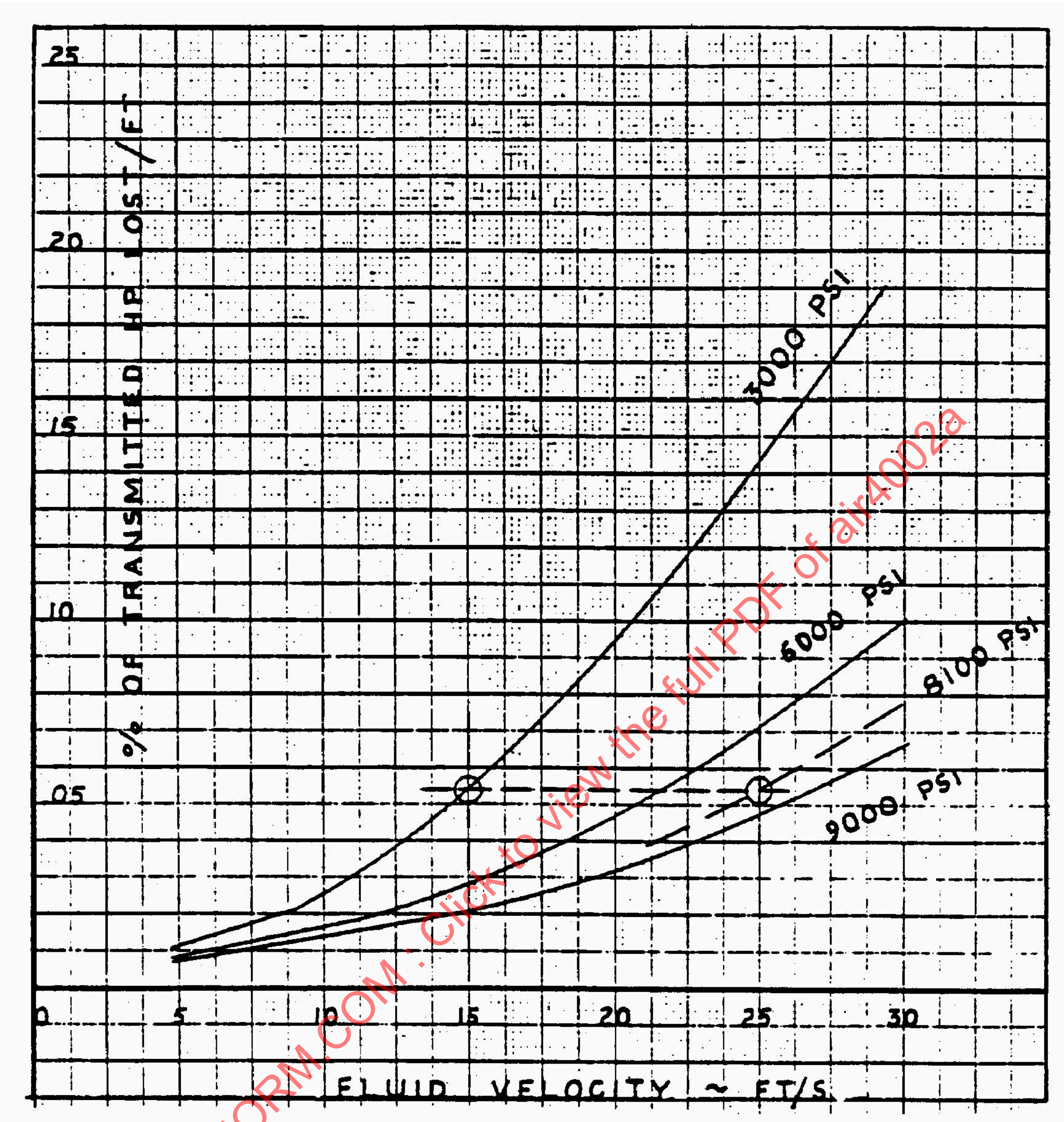


FIGURE 23 – Transmitted Power Loss Versus Fluid Velocity - 1/2 x 0.065 Tubing

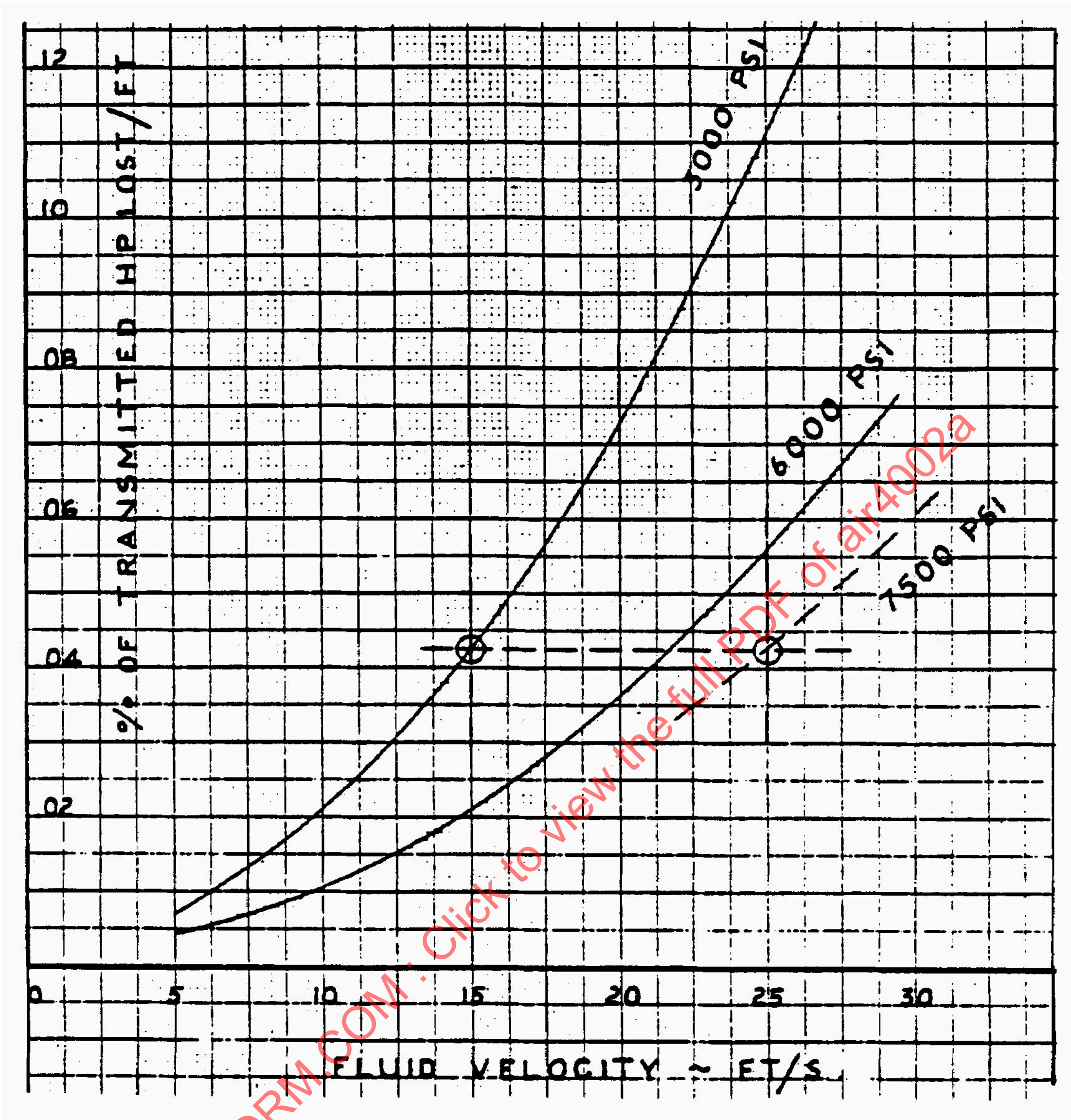


FIGURE 24 – Transmitted Power Loss Versus Fluid Velocity - 5/8 x 0.058 Tubing

TABLE 5 – Pressure Level Selection

	Selected Operating Pressure Level, psi	Relative Importance Factor, %
Power Generation		
Pump Performance	7500	30
Power Transmission		
Tubing Losses	8000	15
Pressure Surges	9000	25
Actuation[1]		
Servovalve Losses	7500	5
Actuator Sizing	8000	25
	Total:	100%

[1]See Section 10

7. WATER HAMMER:

7.1 Water Hammer - 3000 psi Systems:

7.1.1 Classic "water hammer" transient pressures occur when fluids moving at a given velocity are suddenly stopped by closing a valve. For conventional fluids (MIL-H-83282, MIL-H-5606), pressure increases of 40 to 50 psi for each foot-per-second sudden-velocity reductions are typical. The actual transient varies with line size and material. (A 5000 psi pressure rise can occur due to sudden stoppage of a fluid moving at 100 ft/s.) There are "fast-closing" valves associated with the transients being discussed and "slow-closing" valves. It is alleged that any valve closing faster than or equal to the time for a pressure wave to travel to the basic reflection point and back, is a "fast-closing" valve and maximum transients can occur. The valve is also assumed to have a constant closure rate.

7.1.2 In aerospace vehicles, the classic water hammer pressure peaks associated with fast-closing valves with constant rates occur in:

a. FC actuators - during control reversals

b. Utility functions - controlled by solenoid valves with linear openings/slots

7.1.3 For all other control modes, nonlinearities result in a reduction in pressure peaks that may be generated. For example, a control surface may be commanded from one position to another at maximum rate. During most of the motion the valve is wide open. As the commanded new position is approached, the error signal resulting from the feedback portion of the control loop (mechanical or electrical) begins closing the valve. As the null position is approached, closing rate is significantly reduced. The result is a nonlinear closing of the valve which brings the actuator (and fluid) nicely and smoothly to rest. Typically, there is no pressure overshoot at all. So, for FC actuators, maximum transient peaks are almost exclusively associated with maximum rate control reversals.

7.1.4 The magnitude of the peak (for a given fluid) is associated primarily with the valve slew rate, fluid velocity upstream of the valve, and pressure base (pressure immediately upstream of the control valve). Figures 25 through 27 present a look at the important parameters. Figure 25 presents the pressure-loss distribution at maximum no-load rate for a typical 1/3, 1/3, 1/3 design pressure drop in the pressure lines, valve and manifold, and return lines, respectively. The base pressure from which the transient propagates is then the 2000 psi immediately upstream of the valve and manifold.

7.1.5 Figure 26 presents typical reduction in line diameters as the flow is directed to the actuator inlet. Transient peaks versus fluid velocity for a family of lines is also presented. A different velocity is associated with each line size. The smallest flow diameter is usually associated with the coiled tubing or hose at the actuator. So, the highest fluid velocity (and potential for water hammer) occurs at the valve.

7.1.6 Figure 27 presents a typical total pressure versus valve operating time for a given system similar to that identified in Figure 26. One half of the valve operating time is associated with valve closing. The other half is associated with valve opening after reaching null. For valve operating times (2 to 6 ms) that tend to limit wave propagation to the coiled tubing, the peak is highest. As valve operation is slowed, the pressure wave has time to move further toward the pump (and into larger lines) so the average velocity and the peak is reduced. Current valve “stop-to-stop” operating times are in the 15 to 40 ms range.

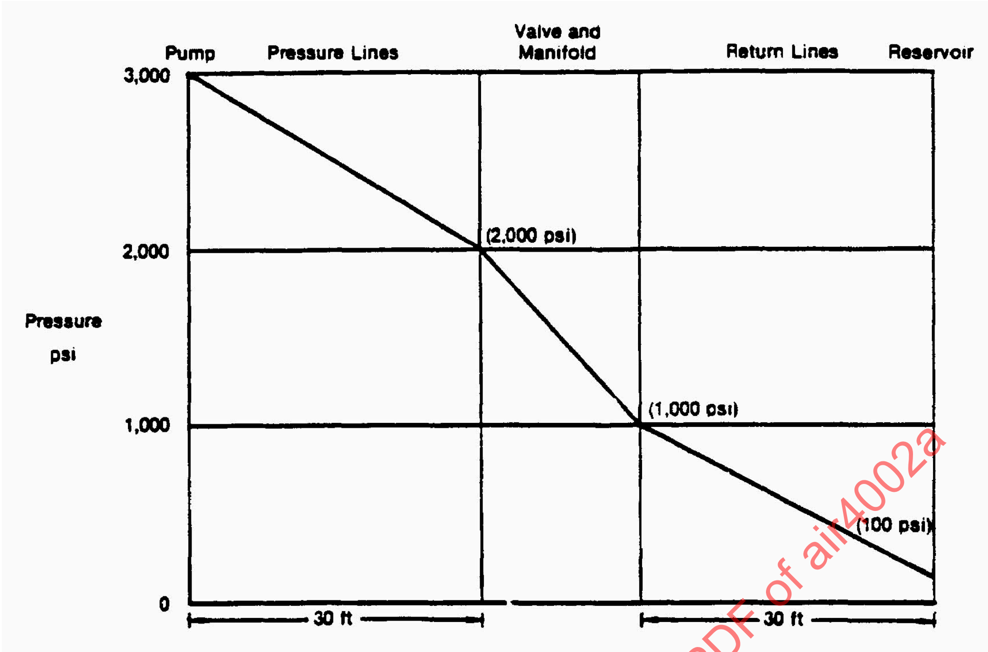


FIGURE 25 – Typical Maximum No-Load Rate System Pressure Drop Distribution

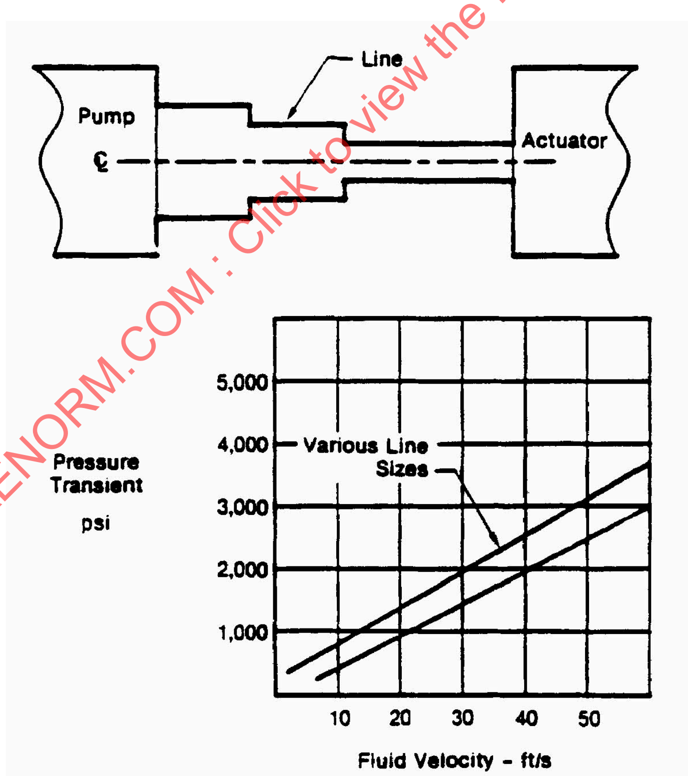


FIGURE 26 – Line Pressure Peaking Characteristics Versus Fluid Velocity

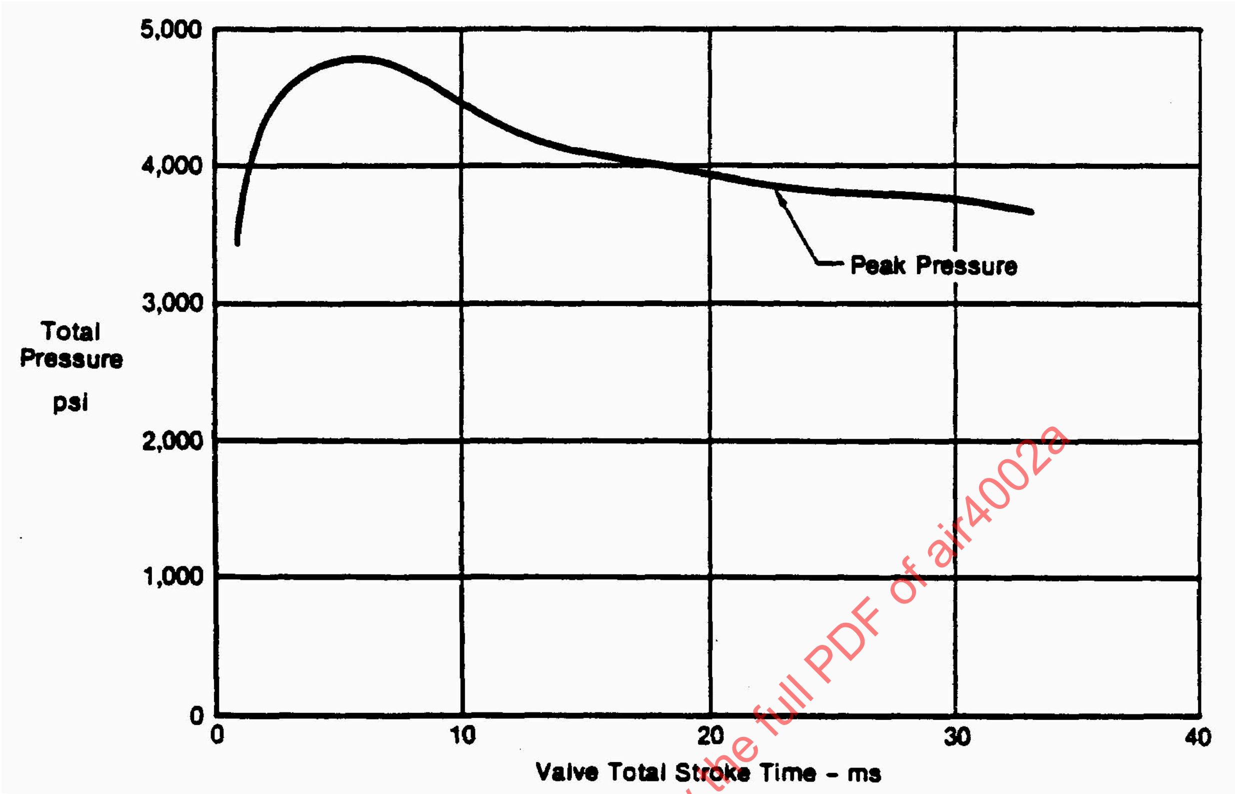


FIGURE 27 – Peak Transient Pressure Versus Valve Operating Time

7.1.7 Figure 28 presents test data taken on the F-15 Iron Bird. The stabilator surface test data shows a 4230 psi peak at the stabilator pressure port for a maximum no-load rate actuator control reversal. The F-15 system is similar to the previously mentioned 1/3, 1/3, 1/3 pressure distribution, continually reducing cross-section tubing approaches were used as was previously discussed. Maximum fluid velocities in the F-15 are 60 to 80 ft/s locally.

7.1.8 With the advent of the CTFE nonflammable fluid, a new dimension in water hammer transient peak potential was experienced. The CTFE fluid is 2.2 times as heavy as MIL-H-83282 or MIL-H-5606 fluids.

7.1.9 Analysis indicated that the CTFE should cause approximately a 40% increase in the magnitude of the transient peak, 60 to 70 psi for each foot per second of velocity for CTFE, for a given velocity and valve operating time.

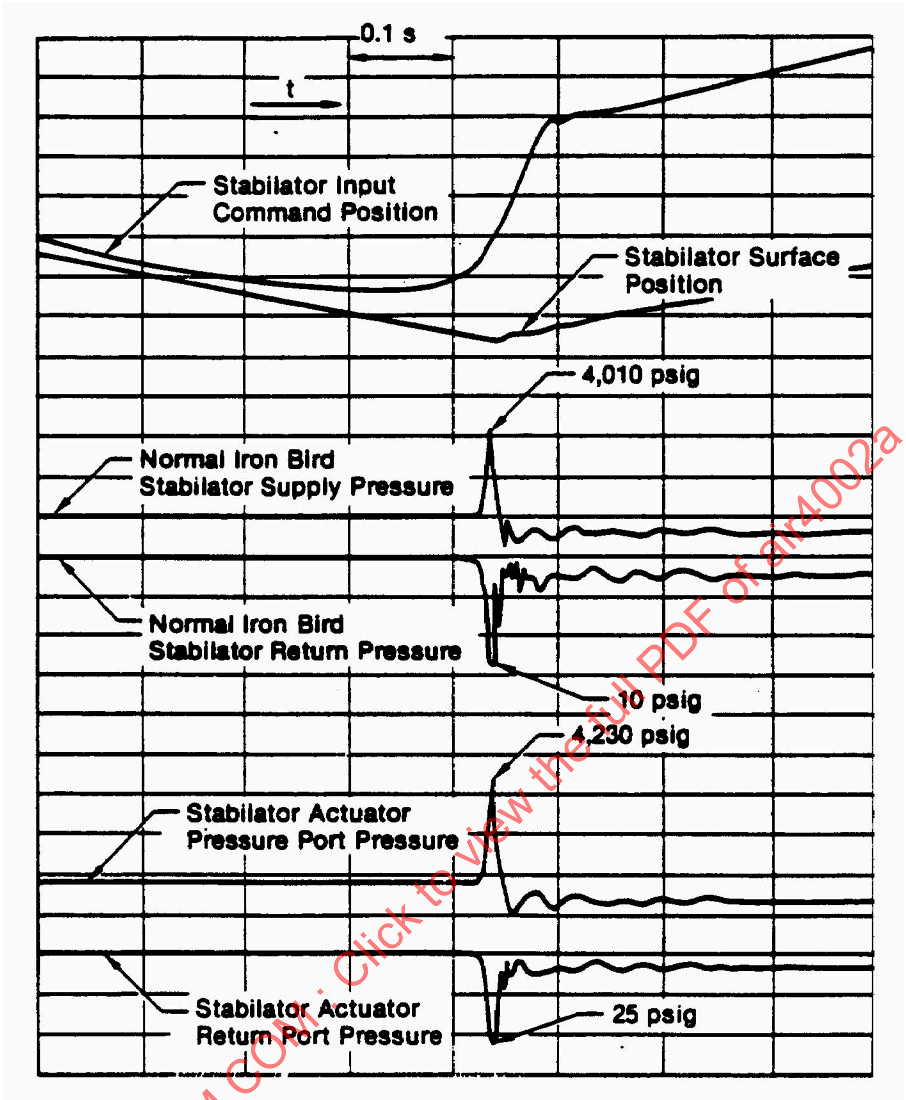


FIGURE 28 – Transient Caused by Maximum Input Rate Reversal While Retracting

7.2 Reduction of Water Hammer Transients - 8000 psi FC Systems:

7.2.1 Various approaches to reduce transient peak pressures when using CTFE fluid were evaluated. The best solutions were the use of asymmetric line loss, nonlinear low-loss valves, and local velocity reduction to achieve acceptable water hammer transient control in FC surface circuits. Figure 29 presents the benefits of asymmetric line loss and nonlinear low-loss valves. The base from whence the transient propagates is reduced from 5053 to 1960 psi, a 3093 psi reduction.

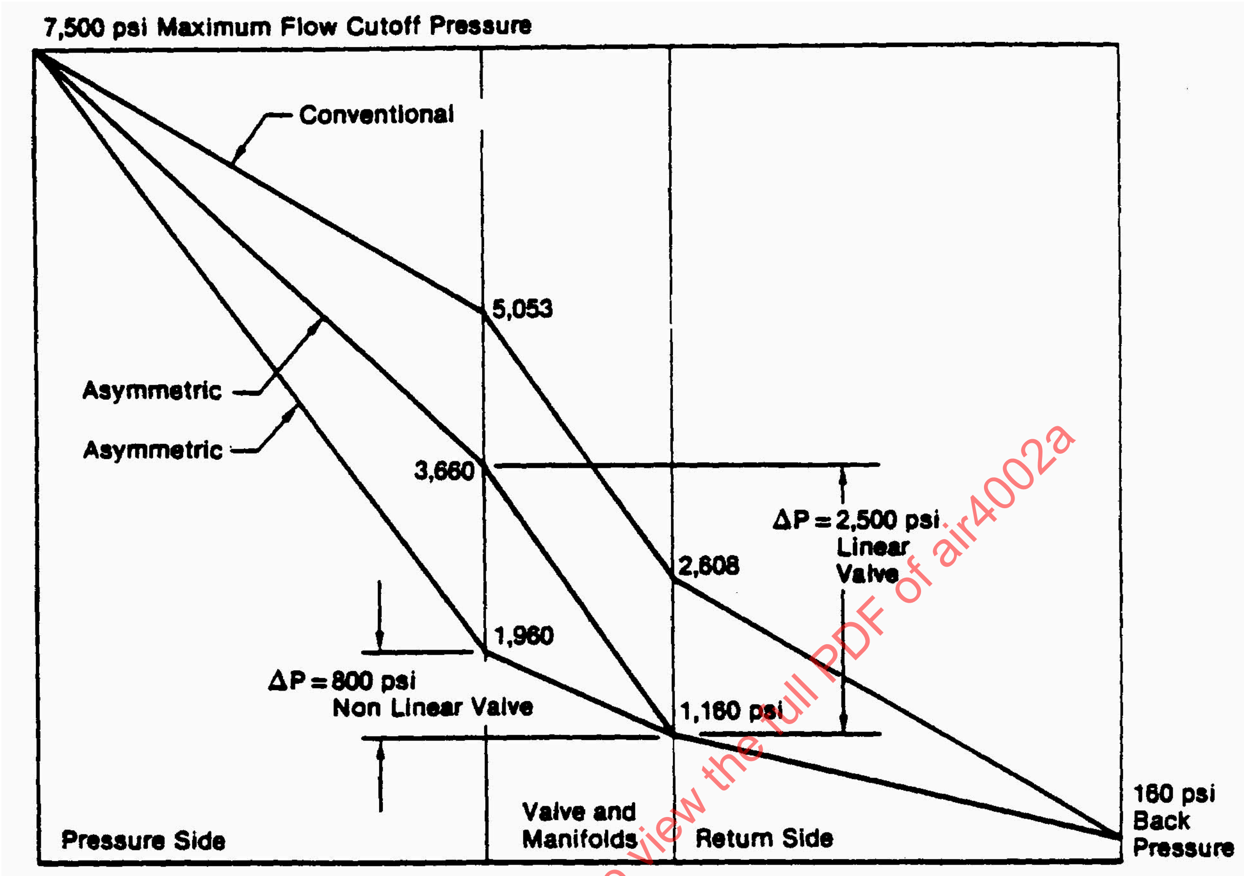


FIGURE 29 – Pressure Loss Distribution

7.2.2 The local velocity reduction technique is shown in Figure 30. This technique uses a venturi-like approach for accelerating and decelerating the fluid. The result is quite low fluid velocities at the actuator and valve. This can significantly reduce the water hammer peak for “fast” valve reversals. Utilizing 8000 psi pressure and resulting smaller lines makes it practical to increase tube size at the actuator, from 3/16 to 3/8 in, for example.

7.2.3 Figure 31 presents analytical results showing benefits of the use of asymmetry, low-loss valves, and local velocity reduction for a specific system. The standard approach, which increases fluid velocity enroute to the valve, shows the potential for achieving transients in excess of 13 000 psi (160 to 170% of rated pressure). In contrast, asymmetry, etc., can maintain transients below system rated pressure. Further, the results show that the final section of tubing immediately upstream of the valve has a powerful influence. Note the difference between the 5/16 and 7/16-in diameter coiled tubing.

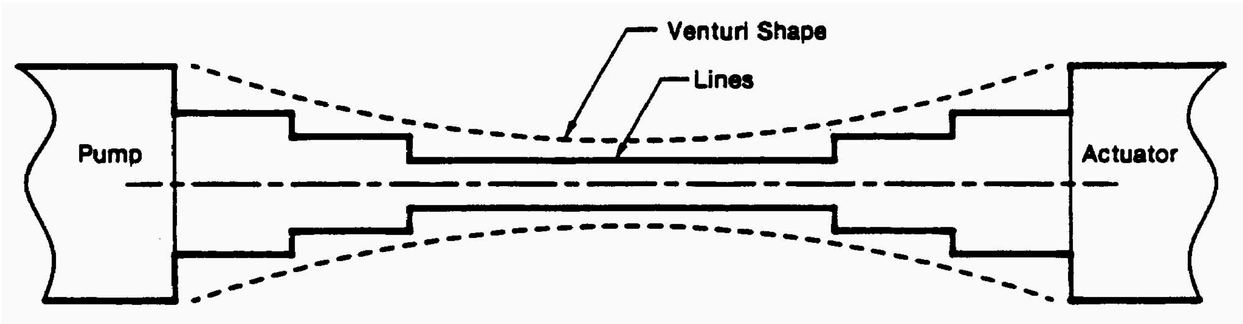


FIGURE 30 – Local Velocity Reduction Concept

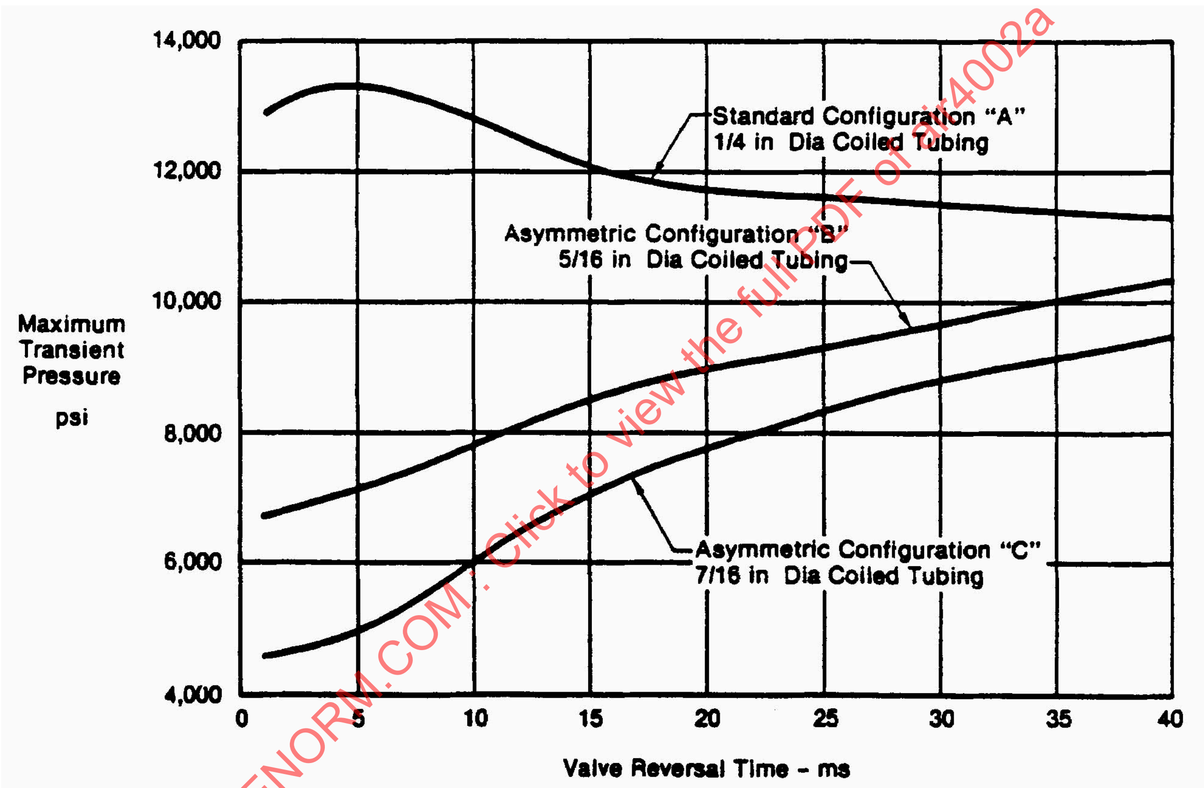


FIGURE 31 – F-15 PC-1 HYTRAN Analyses Summary

7.2.4 The test results for an 8000 psi CTFE fluid system meeting F-15 stabilator requirements are presented in Figures 32 through 37. The pressure side of the test system starts with a 3/8-in diameter line, reduces to 1/4-in diameter, then 3/16-in diameter. As the system approaches the actuator, the line size increases to 1/4-in diameter and the coiled tubing at the actuator is 3/8-in diameter.

7.2.5 The maximum fluid velocities associated with the test are shown in Table 6. The velocity ranges from 141 ft/s (3/16-in diameter) to 32 ft/s (3/8-in diameter).

TABLE 6 – Stabilator Actuator Fluid Velocity

Fluid Temp = 253 °F
Main Ram Velocity = 7.2 in/s
Flow = 5.975 gpm = 23.0 in^3/s

Tube O.D. (in)	Wall Thickness	Fluid Velocity (ft/s)
3/16	0.028	141.0
1/4	0.035	75.3
3/8	0.049	31.8

7.2.6 Figure 32 presents the valve position versus time. At valve reversal, the time from stop to stop is approximately 40 ms (20 ms from full open to closed).

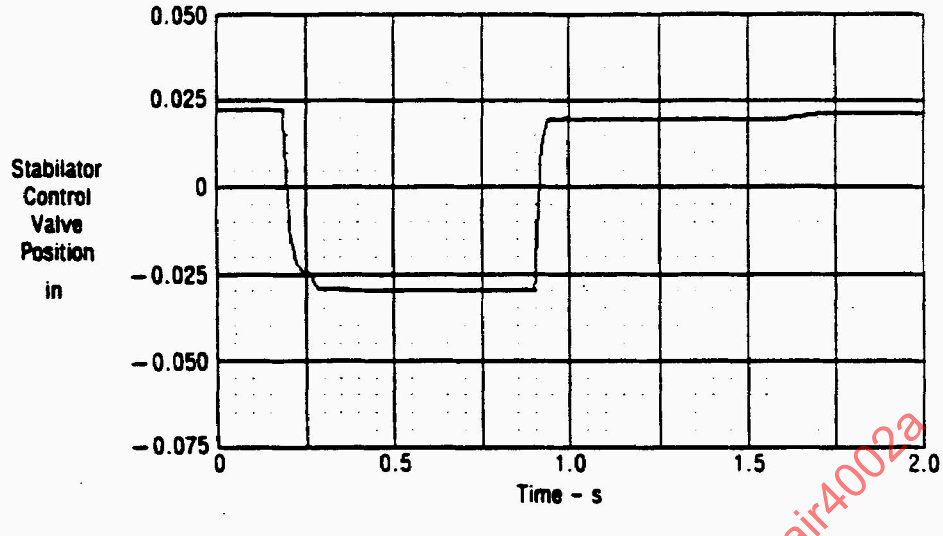
7.2.7 Figure 33 presents the main ram position versus time. A sharp sawtooth occurs on valve reversal that contributes to the maximum transient that can be generated. It is important to note that the control is open loop and the actuator bottoms out extended demonstrating the transients associated with piston bottoming.

7.2.8 Figure 34 presents the pump outlet pressure versus time. There are two peaks. The first occurs with the reversal and is approximately 500 psi above output pressure (8100 psi). The second occurs on piston bottoming with approximately 250 psi overshoot.

7.2.9 The actuator pressure (immediately ahead of the System 1 portion of the valve) versus time is presented in Figure 35. The peak pressure at reversal is 7000 psi propagating from a base pressure of 4500 psi (2500 psi delta). The base is high due to the System 1 piston unbalance; in the other direction of motion, the base is approximately 3000 psi as shown. The overshoot on piston bottoming is approximately 500 psi. The pressure immediately ahead of the System 2 portion of the valve is presented in Figure 36. The peak pressure at reversal is 5000 psi, propagating from a base of approximately 2500 psi (2500 psi delta). The piston bottoming transient overshoot is approximately 500 psi. There appears to have been overkill in controlling the valve reversal transients. Asymmetry, valve drop, and local velocity reduction techniques can possibly be relaxed and still maintain transients below design allowables.

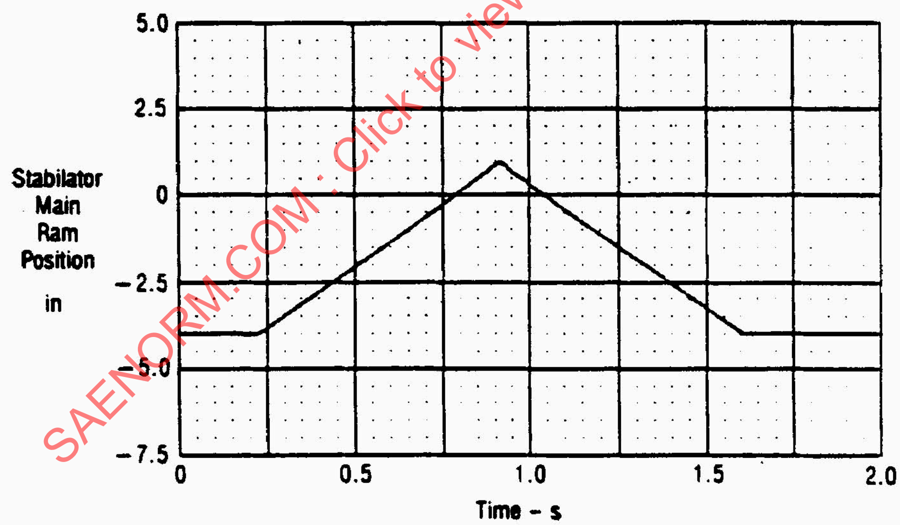
7.3 Reduction of Water Hammer Transients - 8000 psi Utility Systems:

7.3.1 An approach to controlling utility subsystem transients is presented here. For further information, see Reference 34 in Appendix A.



No Load Valve Reversal @ 103°F and 3,800 RPM
Extend to Mid to Extend

FIGURE 32 – CTFE Fluid 8000 psi Demonstration Test - Valve Position Versus Time



No Load Valve Reversal @ 103°F and 3,800 RPM
Extend to Mid to Extend

FIGURE 33 – CTFE Fluid 8000 psi Demonstration Test - Main Ram Position Versus Time

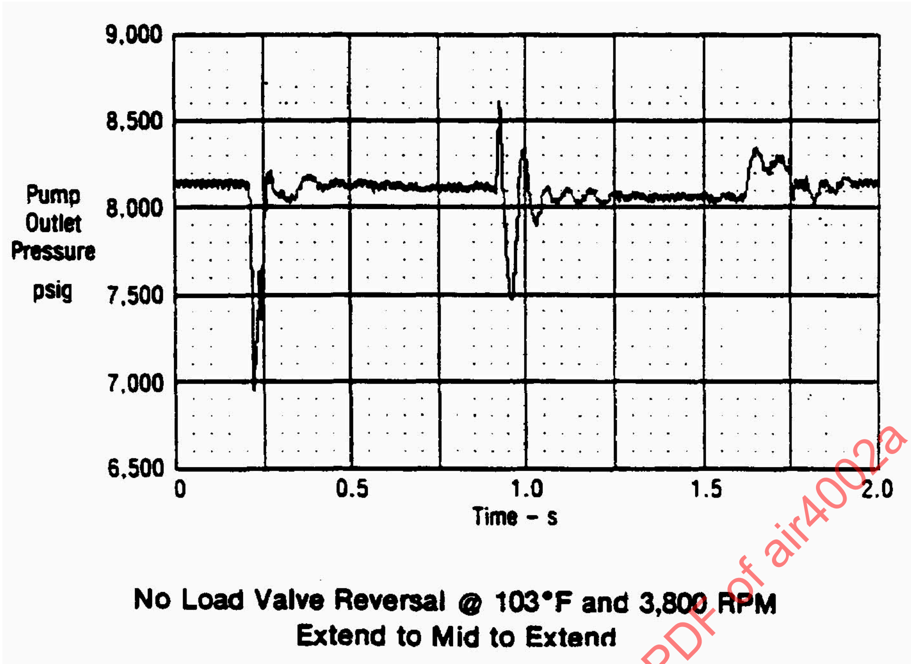


FIGURE 34 – CTFE Fluid 8000 psi Demonstration Test - Pump Outlet Pressure Versus Time

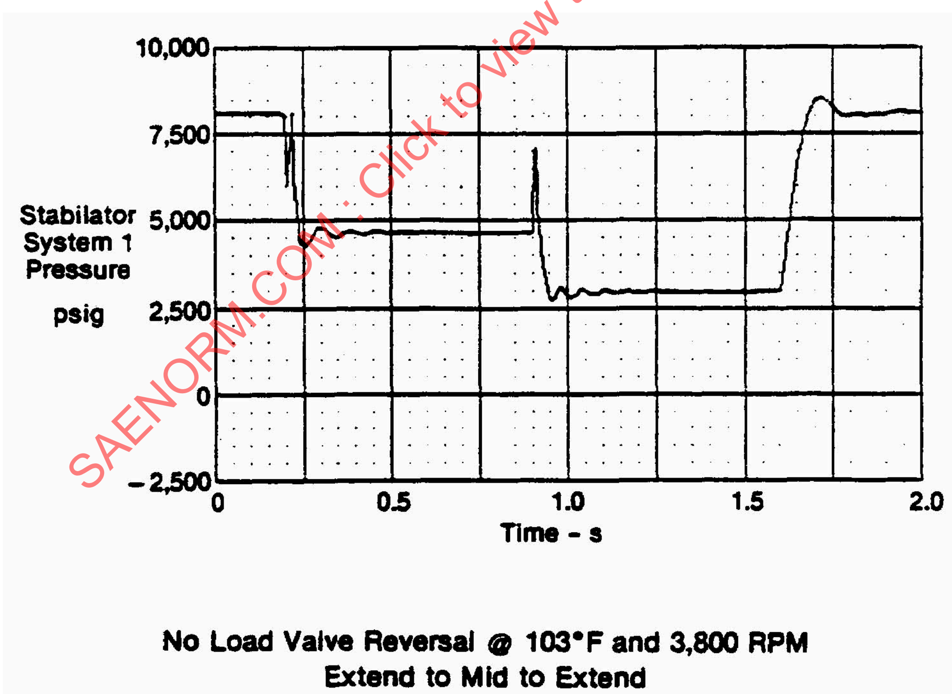


FIGURE 35 – CTFE Fluid 8000 psi Demonstration Test - Actuator Pressure Versus Time

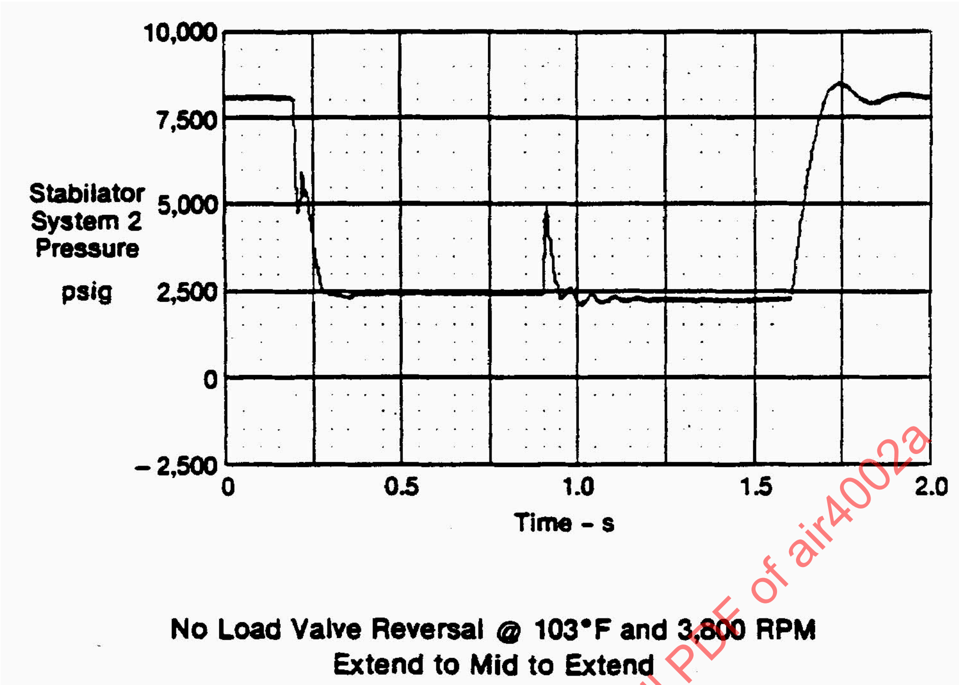


FIGURE 36 – CTFE Fluid 8000 psi Demonstration Test - System 2 Pressure Versus Time

7.3.2 The F-15 utility central system and speed brake subsystem HYTRAN model was developed and exercised to evaluate water hammer using the CTFE fluid. The HYTRAN block diagram is presented in Figure 37.

7.3.3 Valve performances of 20 and 50 ms were analyzed. In addition, nonlinear versions were analyzed. The operating time refers to the time required from initiation to completion of orifice opening.

7.3.4 Valve closure that generates upstream transients was of primary interest. Also, pressure-side transients due to actuator bottoming and return-side transients due to releasing stored energy were considered.

7.3.5 A subsystem operating cycle was established that provided answers to the questions posed. Figure 38 presents the operating cycle used in the HYTRAN simulation of the F-15 speedbrake subsystem. The valve is open to extend the actuator, then closed after a partial extension. This portion of the cycle provides the upstream transient pressure results desired.

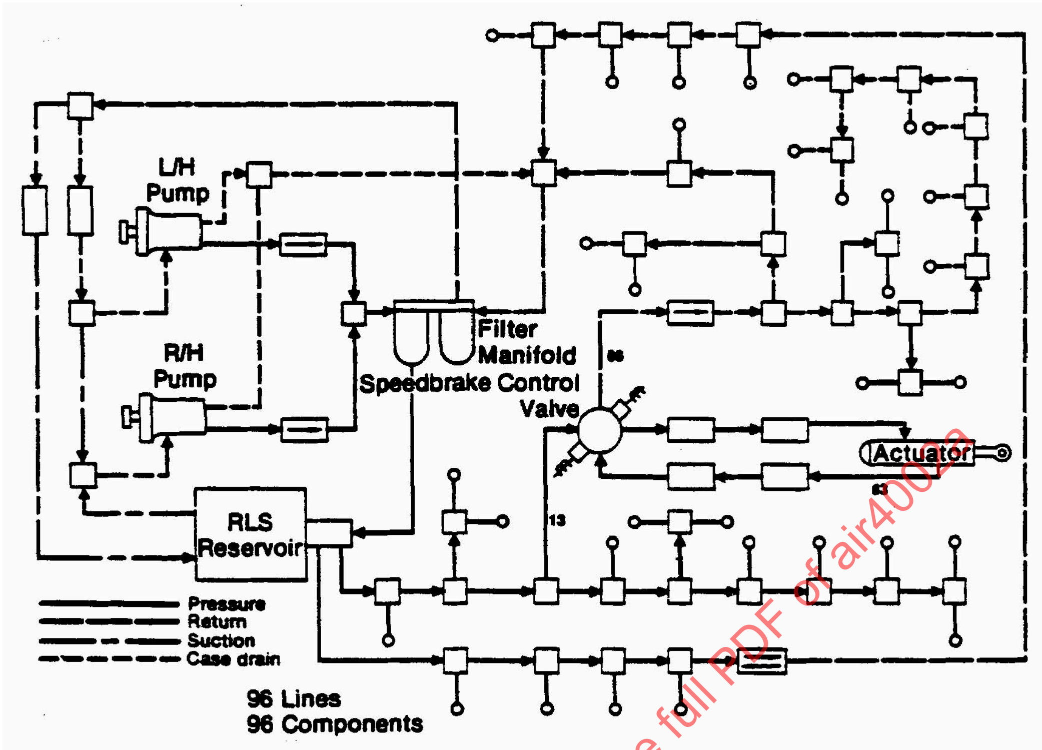


FIGURE 37 – F-15 Utility Hydraulic System Speedbrake Test Circuit HYTRAN Line and Component Arrangement

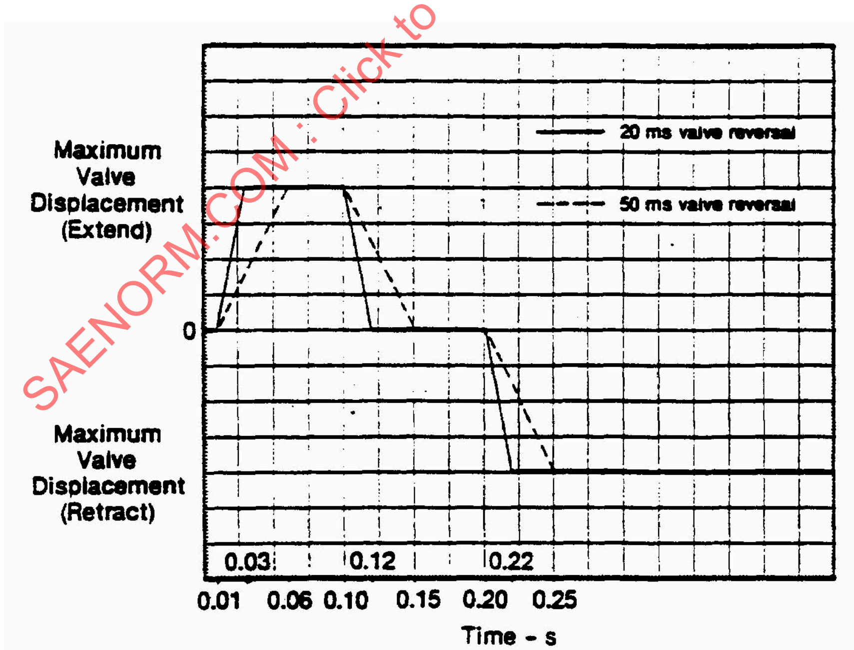


FIGURE 38 – F-15 Speedbrake Operating Cycle - HYTRAN Simulation

7.3.6 The subsequent initiation of the actuator retraction provides return transient results since the pressure trapped in the actuator during hold is dumped to return. Finally, maintaining the control valve in an actuator retract position gives the actuator bottoming transient. Both 20 and 50 ms operating time valves were used.

7.3.7 Nonlinear valve opening and closing characteristics can be powerful in controlling transients. Therefore, linear and nonlinear valves were evaluated. Figure 39 presents the linear and nonlinear characteristics used.

7.3.8 The HYTRAN simulation results are quite interesting. Upstream transients can be modified by controlling valve closing time or by valve nonlinearity. Figure 40 presents a pressure-versus-time prediction of system pressure immediately upstream of the valve.

7.3.9 The valve configuration used was linear, operating at 20 ms time for full valve stroke. The predicted peak was approximately 11 000 psi. Controlling the peak to 9600 psi maximum is the objective. Figure 41 presents predicted pressure immediately upstream of the actuator. The pressure peaks at 9500 psi on actuator bottoming.

7.3.10 For the next simulation, the valve operating duration was 50 ms and the linearity was maintained. Figure 42 presents the predicted pressure versus time printout results. The peak pressure was 9800 psi, a reduction of about 1200 psi.

7.3.11 The final simulation evaluated the use of the nonlinear valve at a 20 ms stroke. The upstream pressure transient characteristic predicted is presented in Figure 43. The predicted peak is 8200-8300 psi versus the desired 9600 psi maximum. From the above results one can conclude that the nonlinear concept is much more powerful than the valve time. In any event, upstream transient control for this type of system can be controlled with very acceptable state-of-the-art techniques.

7.3.12 The return transient predicted when trapped pressure is dumped into the return system is presented in Figure 44. The predicted peak immediately downstream of the valve is about 3500 psi. This seems reasonable and acceptable. The valve used in the simulation was also the 20-ms nonlinear valve.

7.4 Water Hammer Transients - Design Margins:

7.4.1 System transients appear to be controllable to 105 to 110% maximum of system rated pressure. (This compares to 135% accepted in 3000 psi systems, per MIL-H-5440, and 120% accepted for the Navy LHS 8000 psi Program using MIL-H-83282 fluid.)

7.4.2 In view of 7.4.1, it is suggested that design margins for the following design criteria be reviewed for margin reductions. The design recommended margins are shown in Table 7.

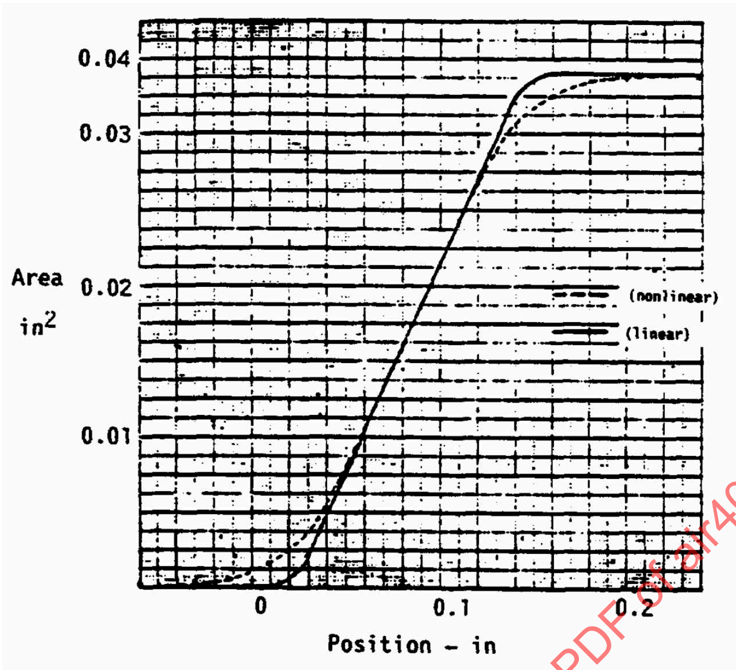


FIGURE 39 – Valve Characteristic Curves

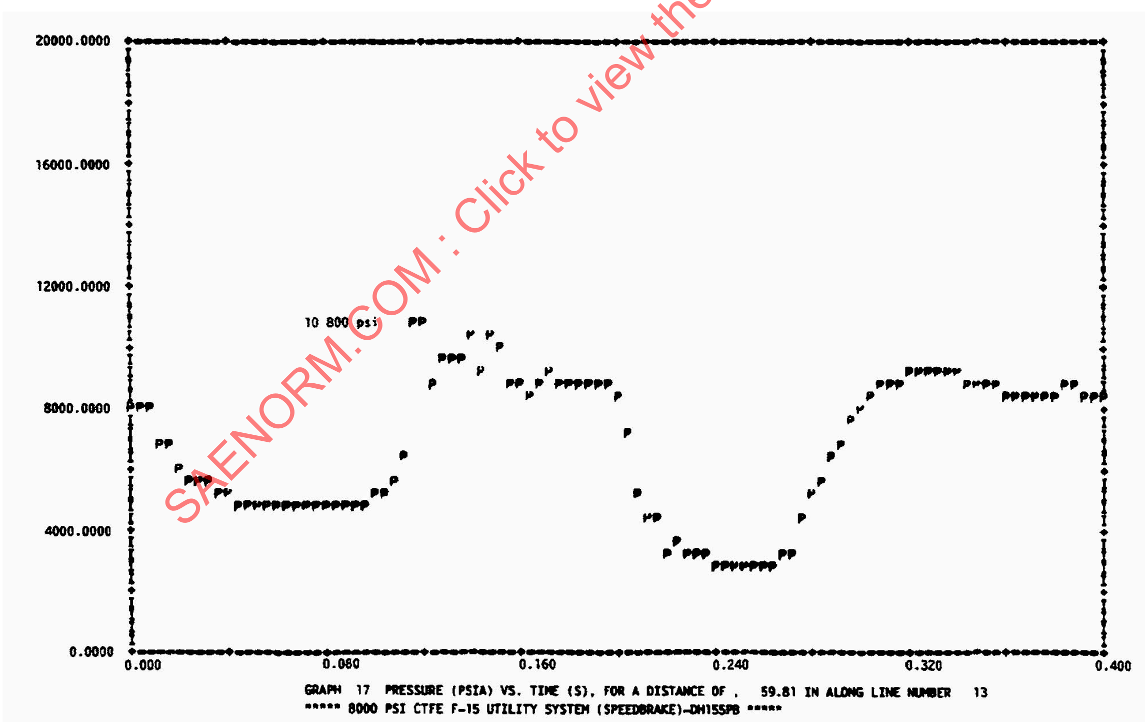


FIGURE 40 – Pressure Upstream of Linear Valve - 20 ms Valve Reversal

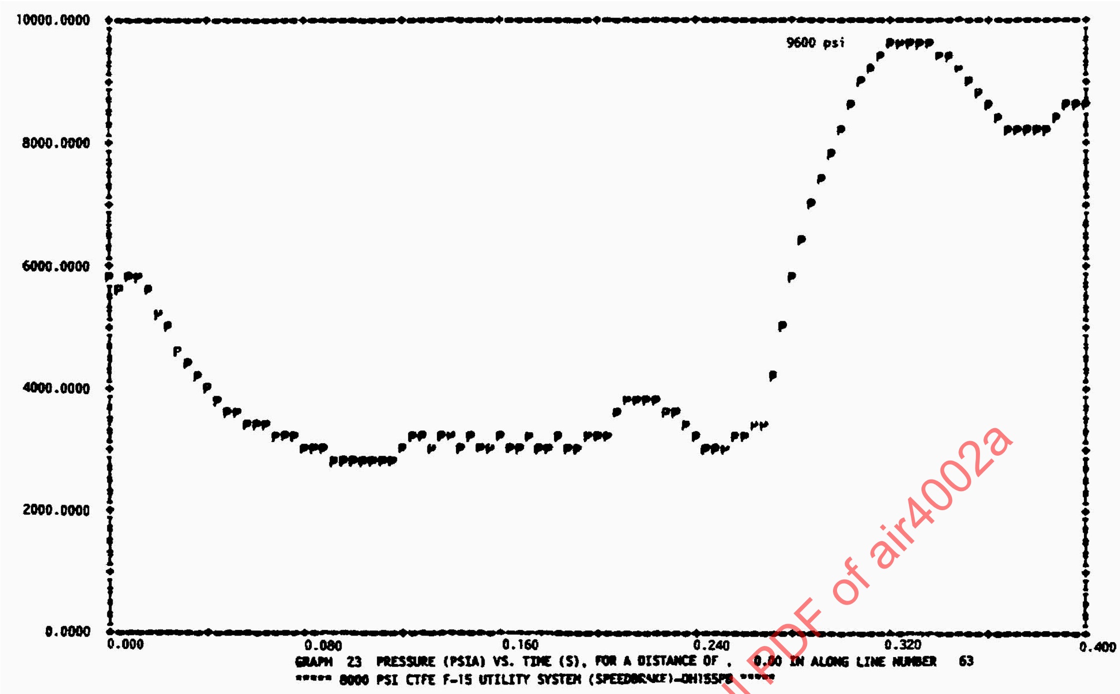


FIGURE 41 – Pressure Upstream of Actuator - 20 ms Valve Reversal

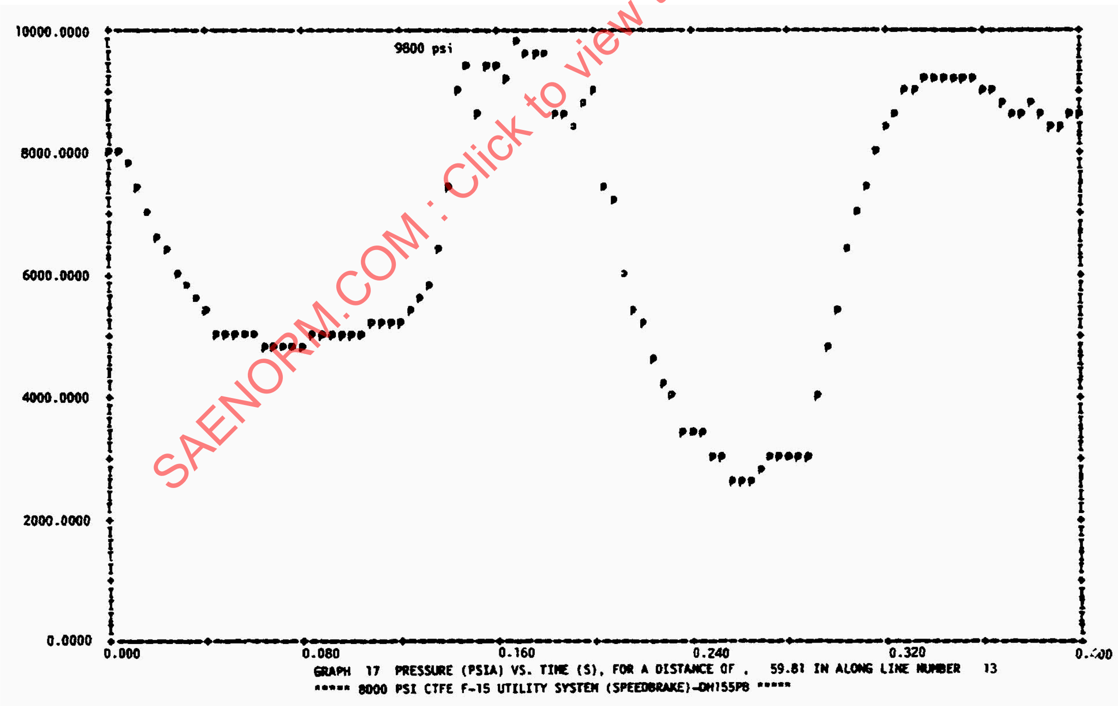


FIGURE 42 – Pressure Upstream of Valve - 50 ms Valve Reversal

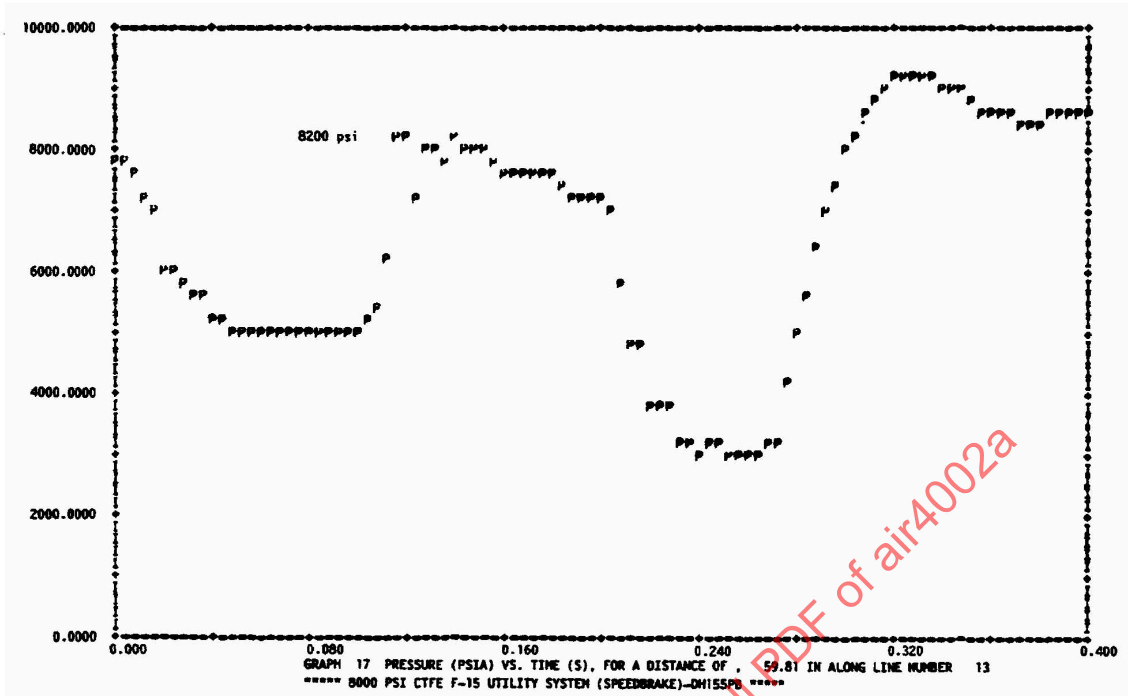


FIGURE 43 – Pressure Upstream of Valve - 20 ms Valve Reversal

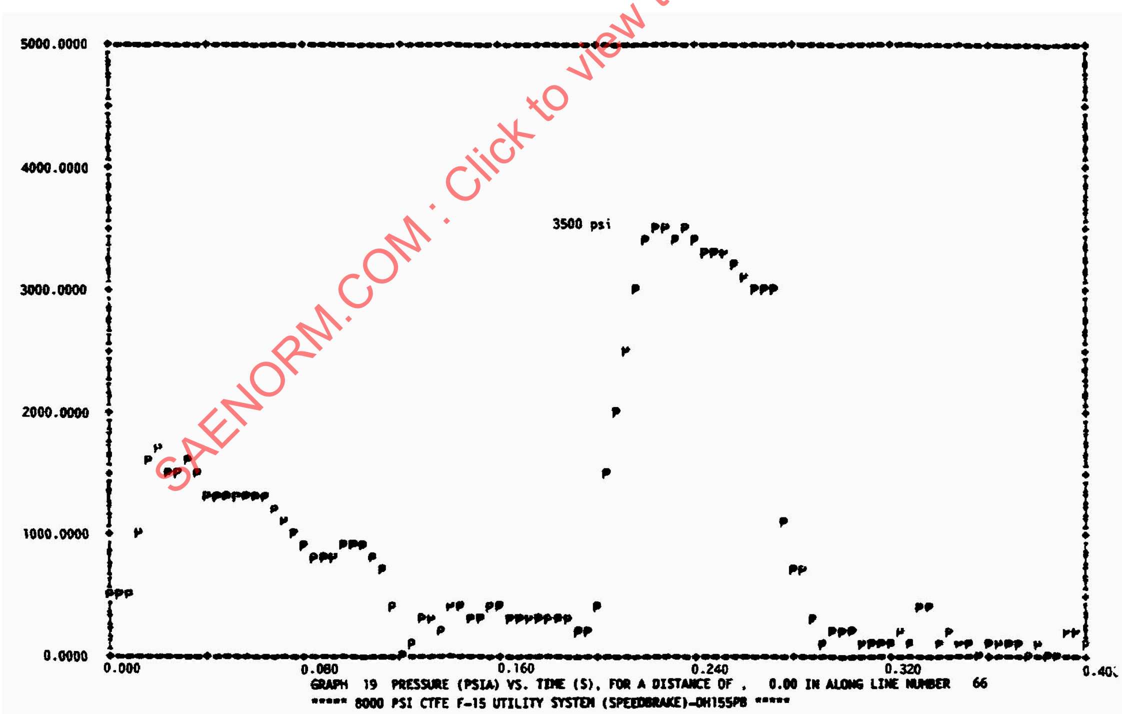


FIGURE 44 – Pressure Downstream of Valve (Return System) - 20 ms Valve Reversal

TABLE 7 – Design Margins

	3000 psi	8000 psi
Components		
Proof pressure	1.5	1.5
Burst pressure	2.5	2.0
Distribution System		
Proof pressure	2.0	2.0
Burst pressure	4.0	3.0
Transient Peak		
Maximum pressure	1.35	1.20

8. SELECTION OF 8000 psi PROOF AND BURST PRESSURES:

8.1 Proof and burst design factors for conventional 3000 psi hydraulic systems using MIL-H-83282 fluid are established in MIL-H-5440 and are listed. The factors are expressed in percent of operating pressures.

TABLE 8 – Proof and Burst Design Factors

	Tubing and Fittings (%)	Components (%)
Maximum Allowable Surge Pressure	135	135
Proof Pressure	200	150
Burst Pressure	400	250

8.2 The design of aircraft hardware is usually based on burst pressure (ultimate strength) rather than proof pressure (yield strength) because this results in more efficient design (less weight). The burst factors (B.F.) for an 8000 psi system were based on allowable pressure surges in a 3000 psi system and the following ratios.

$$\frac{B.F. - Surge\ Factor}{Surge\ Factor - 100\%} = \frac{Overall\ Safety\ Factor}{Surge\ Safety\ Factor} \qquad \text{(Eq. 3)}$$

For a 3000 psi system the ratios are:

$$Tubing\ and\ Fittings = \frac{4.00 - 1.35}{1.35 - 1.00}\,7.5 \qquad \text{(Eq. 4)}$$

$$Components = \frac{2.50 - 1.35}{1.35 - 1.00}\,3.3 \qquad \text{(Eq. 5)}$$

These ratios were used to establish the 8000 psi burst pressure factors. 8000 psi proof pressure factors remain the same as those used in 3000 psi systems.

8.3 The maximum pressure surge developed at 8000 psi in transmission lines with 25-ft/s fluid velocity average 120% of the operating pressure level. B.F. for an 8000 psi system were determined as follows:

$$Tubing\ and\ Fittings = \frac{B.F. - 1.20}{1.20 - 2.70}\,7.5 \qquad \text{(Eq. 6)}$$
$$B.F. = 2.70$$

$$Components = \frac{B.F. - 1.20}{1.20 - 1.00}\,3.3 \qquad \text{(Eq. 7)}$$
$$B.F. = 1.85$$

8.4 Adding in factors to allow for material handling, material quality control, and fabrication/installation techniques, the tubing and fitting B.F. was selected to be 3.0, and the component B.F. was selected to be 2.0. The 8000 psi safety factors are summarized in Table 9.

TABLE 9 – 8000 psi System Safety Factors

Maximum Allowable Pressure	Tubing and Fittings (%)	Components (%)
Surge	120	120
Proof	200	150
Burst	300	200

8.5 Factors of safety currently used in the design of various areas of military aircraft are given in Table 10. Note that the factor of safety for the design of hydraulic tubing is very conservative compared to other components sharing a similar responsibility for flight safety.

TABLE 10 – Current Systems Safety Factors

Design Area			Factor of Safety (%)
Structural elements			150
FC mechanisms			150
Electrical systems			
	Overload (5 min)		150
	Overload (1 s)		305
Environmental Systems			
	Proof		150
	Burst		200
Fuel Systems			
	Proof		150
	Burst		250
Hydraulic systems (3000 psi)			
	Components:	Proof	150
		Burst	250
	Tubing:	Proof	200
		Burst	400

9. TUBING AND FITTINGS:

9.1 Tubing Selection:

The following discussion is presented to emphasize the premise that a tubing B.F. of 3 for 8000 psi systems is equivalent, safety-wise, to the B.F. of 4 used in 3000 psi systems.

9.1.1 Special size tubing was fabricated for the LHS program. Use of this tubing permits the transmission of nearly twice the horsepower per pound of tubing at 8000 psi than at 3000 psi (for a given fluid velocity and tube material). Tubing wall thickness was determined using a burst pressure of 24 000 psi (12 000 psi is used for 3000 psi systems). The B.F. used for 8000 psi tubing was based principally on the fact that pressure surges encountered in 8000 psi systems are smaller, percentage-wise, than in 3000 psi systems. Surprisingly, the B.F. of three provides nearly twice the pressure safety margin at 8000 psi than the B.F. of four does at 3000 psi, as shown in Table 11.

TABLE 11 – Tubing Design Margins

Operating Pressure (psi)	Max Allowable Press. Surge (%)	Max Allowable Press. Surge (psi)	Burst Pressure, (psi)	Pressure Safety Margin (psi)
3000	135	4050	12 000	7 950
8000	120	9600	24 000	14 400

9.1.2 Comparison of stresses in 1/4 x 0.020 304 CRES tubing typically used in 3000 psi systems and 1/4 x 0.025 21-6-9 CRES tubing proposed for 8000 psi systems are compared in Table 12.

TABLE 12 – Stress Comparisons

Operating Pressure (psi)	Tube Size	CRES Material	Material Ultimate Strength (psi)	Hoop Stress at Operating Pressure (psi)
3000	1/4 x 0.020	304	105 000	17 250
8000	1/4 x 0.025	21-6-9	142 000	36 000

9.1.3 Fatigue properties of 1/4 x 0.020 304 CRES and 1/4 x 21-6-9 CRES have been evaluated by Resistoflex using the rotary flexure method. Tubing life appears to be infinite (more than 10 M cycles) when the combined stress level (bending + axial stress) is below 48 000 psi. This is considered a satisfactory design maximum for most applications.

9.1.4 The human element in fabricating and installing tube assemblies can sometimes lessen the integrity of the tubing. Tolerances on tube diameters and material imperfections can reduce tubing strength. The B.F. used for 3000 psi tubing includes these influences and results in a thicker wall than would be required by pressure alone. Tubing for 8000 psi systems has thicker walls and, thus, inherently has the extra ruggedness required for handling. This aspect permits a more efficient design, weight-wise.

9.1.5 Sizes of the titanium tubing utilized in 8000 psi systems are shown in Table 13.

TABLE 13 – Titanium Tubing Sizes

Dash No.	Outside Diameter (in)	Wall Thickness (in)
-3	3/16	0.020[1]
-4	1/4	0.026
-5	5/16	0.032
-6	3/8	0.038
-7	7/16	0.045
-8	1/2	0.051
-10	5/8	0.064

[1] 0.035 wall thickness is currently used for coiled tube applications.

9.1.6 Figure 45 shows that at lower operating pressure levels, for example 1000 psi, large factors of safety are required to design tubing that is rugged enough for handling. This unwanted effect disappears at 8000 psi.

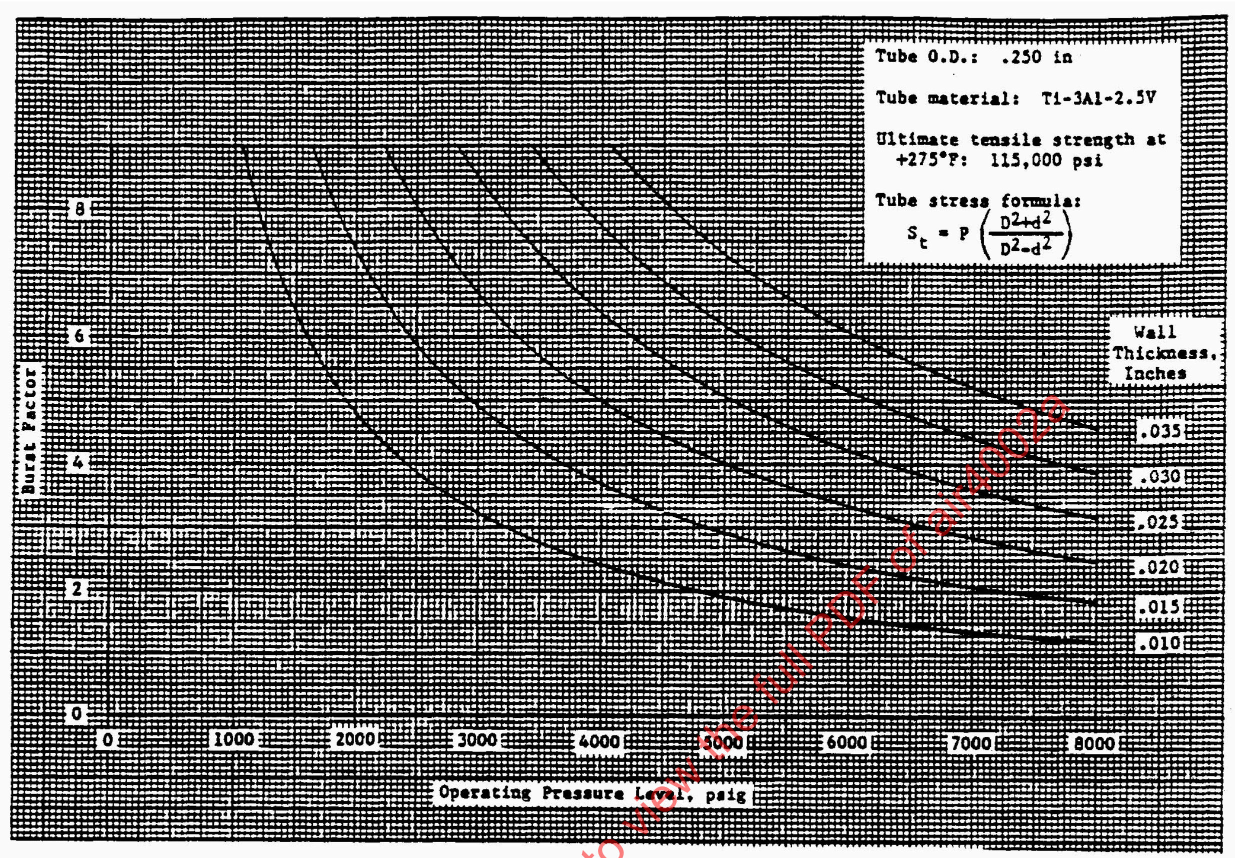


FIGURE 45 – Burst Factors for -4 Size Titanium Tubing

9.2 Coiled Tubing:

9.2.1 For many years the only devices available to transmit hydraulic power across moving joints were hoses, swivels, extension units, or a combinations of these devices. While great strides in reliability and compactness have been made in these designs, there remained a need for a method that had no moving joints, required no elastomeric seals, was compact, and was low in weight. In 1955 and 1957, significant work was done to define the stress analysis and design criteria for straight tubing loaded in torsion and coiled tubing loaded in bending. The tubing material was corrosion resistant steel. Many successful plumbing installations have been based upon these efforts and other pioneering work conducted in the early 1940’s.

9.2.2 The Rockwell/Navy LHS program determined that system reliability would be improved if extension units and swivels, both containing elastomeric seals, were replaced. The extension units provide a flexible fluid connector for moving-barrel actuators with linear motion. Swivels provide a flexible fluid connector for moving-barrel actuators with motion about a pivot. The coiled tube configurations available from the 1955 work, and documented in ARP584, are applicable only to motion about a pivot. The material is 18-8 CRES steel which has a relatively high torsional modulus. Only a single coil diameter was recommended for each tube size. The minimum tube size considered was 1/4 in.

9.2.3 Coiled tube technology has been extended by the introduction of new configurations useable with linear or pivotal motion, use of 3/16 in diameter tube, recommendations for three titanium alloy tube materials, and design equations allowing choice of coil mean diameter. Also, safe tube-wall thickness (0.035 in) is recommended for system pressure up to 8000 psi.

9.2.4 The basic configurations evaluated were the helical and tri-coil (see Figures 46 and 47).

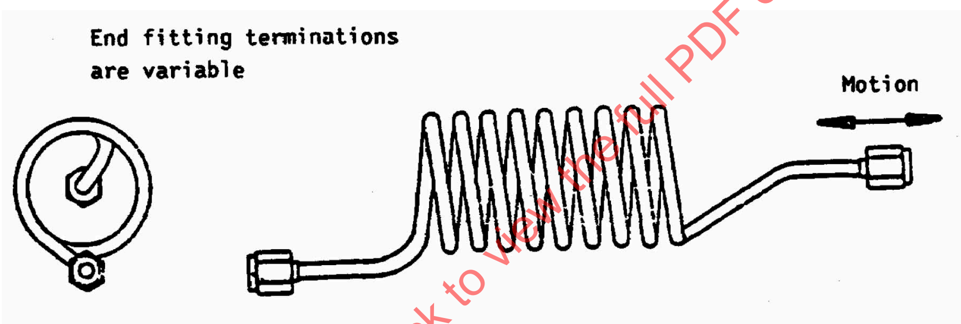


FIGURE 46 – Coiled Tube Helical Configuration

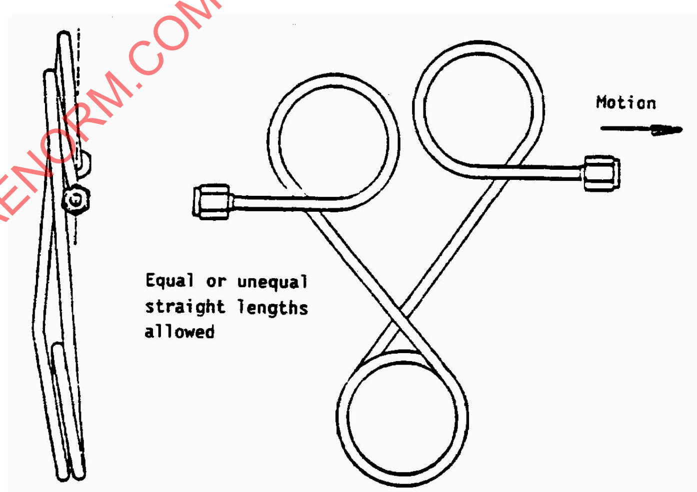


FIGURE 47 – Coiled Tube Tri-Coil Configuration

9.2.5 The materials selection study used the ratio of tensile strength to torsional modulus as the function to be maximized. A material with the highest ratio is considered a good spring material. The materials selected were cold-worked, stress-relieved Ti 3Al-2.5V and annealed Ti 3Al-4V. Material properties such as ultimate shear strength and torsional modulus for titanium alloys are not readily available.

9.2.6 Undamped configurations showed amplification of input acceleration. Relatively simple damping methods attenuated the amplification of random vibration. In the brief tests conducted, the use of low-density foams and plastic ties were moderately successful, but further work is required.

9.2.7 Fabrication of coils was generally very straight-forward. Difficulty was encountered in the forming of the bends from the helical coil to the straight lengths going to the end fittings. The hand forming techniques used would be acceptable for the small quantities involved in a test and development program, but would not be satisfactory for production of tubing in greater quantities.

9.3 Fittings:

9.3.1 In the 8000 psi systems four fitting configurations are currently employed. Size availability is quite limited.

a. Rosan Boss Fittings: Used as connectors on actuators. They require an O-ring seal.

b. Resistoflex (Dynatube) Reconnectable Fittings: Used for separable movable fittings. They are internally swaged using special tooling. Lip seal technology is employed in the separable joint.

c. Deutsch (Permaswage): Used for permanent and removal fittings. They are externally swaged and use lip seals in separable connections. Weepage has been experienced in the -3 sizes in coiled tubing installations. (3/16 x 0.035 wall titanium). It has been determined that the fittings used were designed and qualified for 3000 psi systems that use 0.020 wall thickness titanium. Under the conditions in which the Deutsch fittings were used, they did perform very well. Deutsch has designed and intends to qualify fitting systems for 8000 psi use.

d. Advanced Metal Components Inc. (Cryofit): Used for permanent fittings. They are also suitable to repair lines.

9.4 Hoses:

9.4.1 Hoses have been developed that are usable with 8000 psi hydraulic systems. They are compatible with either MIL-H-83282 or CTFE fluids. Sizes are available from -3 size to -8 size. Lengths in use are from 30 in to 15 ft. Hoses are designed for proof pressures of twice operating pressure (16 000 psi) and burst pressures of three times operating pressure (24 000 psi).

10. ACTUATION SYSTEM:

10.1 Servovalve Null Leakage:

10.1.1 Total power generation capacity of hydraulic systems in modern military aircraft commonly ranges up to 400 hp. These systems generally have up to 100 spool and sleeve type modulating control valves. Considerable industry effort is being expended in this area, and results will be reported in future updates of this document.

10.2 Actuator Sizing:

10.2.1 A typical, highly loaded, control surface actuator was analyzed to determine actuator total weight based on operating pressure level. A dual system, tandem, all steel actuator was assumed. The cylinder walls were sized for burst, and the actuator assembly was sized for column buckling by varying the horn radius to locate the critical column conditions. Hinge moment, angular surface travel, and rate were assumed constant. The actuator design was programmed for a digital computer that performed iterations and converged upon an optimum design for each of the four operating pressures as follows:

a. 3 000 psi

b. 6 000 psi

c. 9 000 psi

d. 12 000 psi

10.2.2 The actuator weights and cylinder O.D. areas determined were used to construct the curves presented on Figure 48. The pressure at which the benefits of higher operating pressure levels being to diminish occurs at approximately 8000 psi.

11. PUMP DEVELOPMENT:

11.1 Various pump designs have been developed thus far for 8000 psi programs. The main suppliers of pump hardware for aircraft application are Abex Corporation and Vickers. All pumps are axial piston designs with pressure compensation and variable flow delivery.

11.2 A unit was developed by Abex for the Navy in 1970. The pump, M/N AP6V-57, was intended for laboratory test purposes. The design was adapted from an Abex built pump used on the Boeing 747 airliner. Bearing loads and stress levels in the AP6V-57 unit were approximately the same as those in the Boeing pump. Rated delivery at 4000 rpm was 14 gpm at 7850 psi. The pressure compensator was adjustable from 3000 to 9000 psi. The AP6V-57 pump design has been updated several times to meet changing needs and incorporate improvements. One version of this unit, Abex M/N AP6VH-3, is currently being used in Air Force CTFE programs. Estimated total operating time on the numerous pumps currently in use is 3000 h.

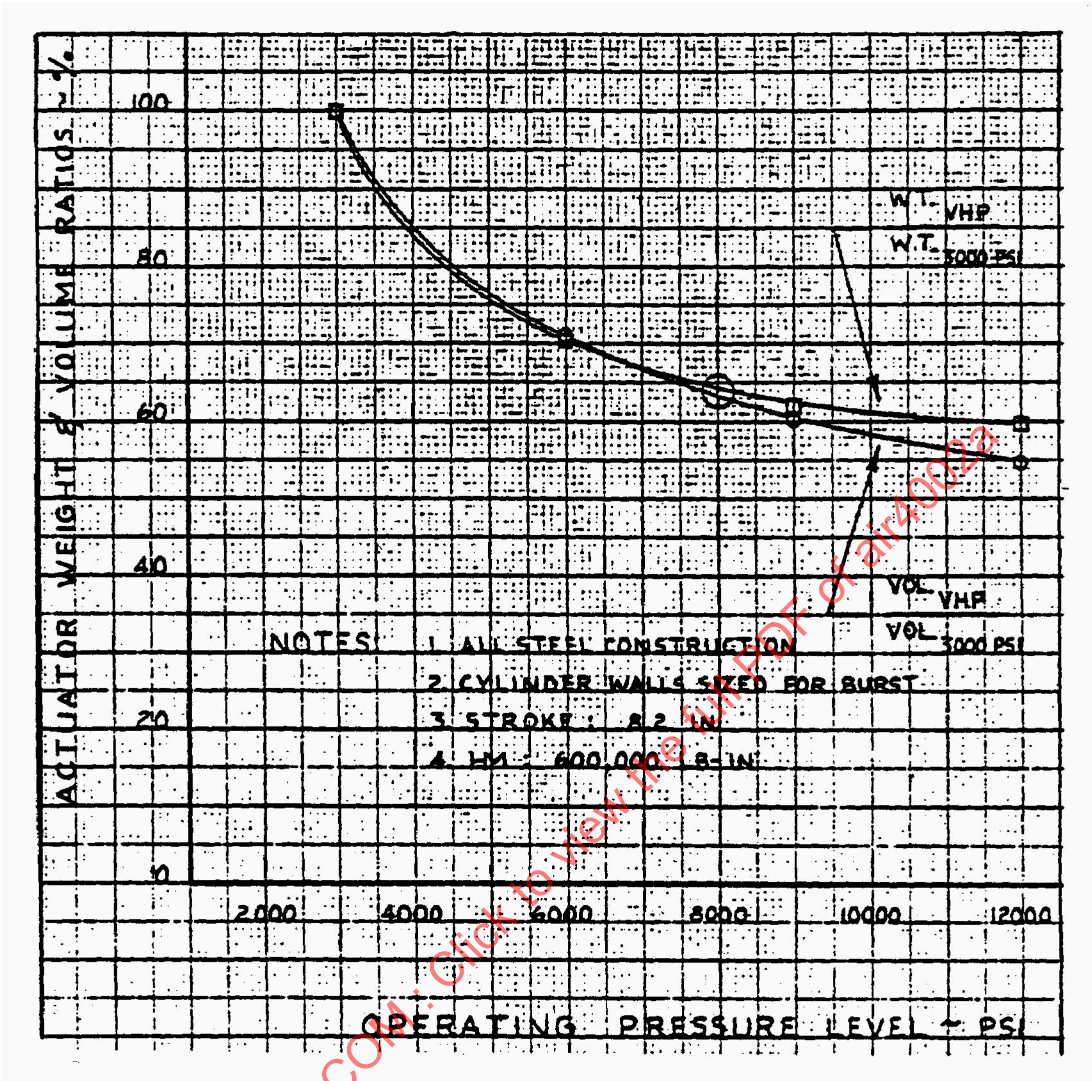


FIGURE 48 – Optimized Control Surface Actuator Another pump developed was Abex M/N AP1V-106. This unit was built for the first flight test program involving 8000 psi hardware for the Rockwell/Navy T-2C program. Rated delivery was 3 gpm at 7300 rpm and 7850 psi.

11.3 A pump developed by Vickers in 1980 is currently being used on the Navy A-7E LHS simulator at Rockwell. The pump, identified as M/N PV3-047, has a 10-gpm rated output at 5900 rpm and 7800 psi. PV3-047-3 is the current model. The unit is designed to meet envelope and engine mounting requirements of the Vought A-7E Corsair.

11.4 Pumps for ground applications have been built by Denison Hydraulics Inc. for use on 8000 psi stationary test benches and ground support equipment. This unit is a version of the Gold Cup series modified for use at 8000 psi. Rated flow is 20 gpm at 1200 rpm. The compensator is adjustable from 1000 to 9000 psi. Pump electric servocontrol provides variable flow delivery.

11.5 Dynex/Rivett Inc. is also providing an 8000 psi 9 gpm check valve design pump for use on a ground support stand for the Rockwell/Navy program (Section 12).

12. GROUND SUPPORT EQUIPMENT:

12.1 A Navy AHT-63 (electric driven) ground test stand has been modified for service at 8000 psi. This unit, which has a capacity of 9 gpm at 7900 psi, has been demonstrated in conjunction with the Navy A-7E lightweight hydraulic simulator.

12.2 It was necessary to replace the following 3000 psi components in the high pressure circuit with 8000 psi components:

a. Pump

b. Check valve

c. Filter

d. Gage

e. Disconnect

f. Bypass valve

g. Relief valve

h. Lines, fittings, and hoses

13. SAFETY:

13.1 Concerns have been expressed about the safety aspects of operating at 8000 psi. Early in the high pressure program development efforts, the Navy conducted gunfire tests of both 3000 and 8000 psi systems for comparison purposes. The results showed no abnormal differences between the two pressure levels. In both instances, when the bullets severed the lines, a mist-like spray occurred. This very quickly disappeared as the pressure dropped in the systems. Due to the smaller fluid volume in the 8000 psi system (approximately half), the fluid depleted more rapidly than the fluid in the 3000 psi system at low flow and low pressure, much like trapped fluid in a garden hose when the nozzle is depressed.

13.2 Another series of tests were conducted by putting a 0.008 in diameter hole in the end of a fitting attached to a 3/16 in diameter line. The resulting sprays, at both 3000 psi and 8000 psi, were remarkably similar. It was difficult to maintain a spray for more than a few seconds due to the inability of the supply pump to maintain the pressure with a depleting volume of fluid.

13.3 During the course of operation of the LHS (over 1200 h) several failures occurred in hardware (fittings, actuators, and other hardware). The mode was an initial weep at the failure point, followed by drops, then low pressure flow.

13.4 The Air Force (AFWAL/POOS) conducted a series of tests using CTFE fluid at 8000 psi. A fitting failure resulted in a fan-like spray that dissipated as the fluid depleated. In another instance, a pinhole leak resulted in a spray. A paper towel was held within 6 in of the leak source. The towel was saturated with fluid but not penetrated by the spray.

13.5 It can be concluded that safety is no more a problem at 8000 psi than at 3000 psi. Normal safety practices should be observed when working with any pressure in a closed system.

14. LIST OF ABBREVIATIONS:

AFCS	Automatic Flight Control System
AFWAL	Air Force Wright Aeronautical Laboratories
ASTM	American Society for Testing Materials
B.F.	Burst Factor
CTFE	Chlorotrifluoroethylene
EPDM	Ethylene Propylene Diene Monomer
FC	Flight Control
HYTRAN	Hydraulic Transient Analysis
LHS	Lightweight Hydraulic System
LH-UHT	Left Hand - Unit Horizontal Tail
ms	Millisecond
NADC	Naval Air Development Center
P_k	Probability of Kill

14. (Continued):

PNF	Phosphonitrilic Fluoroelastomer
RFI	Roll Feel Isolation
RH-UHT	Right Hand - Unit Horizontal Tail

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