SAE Aerospace An SAE International Group	AEROSPACE INFORMATION REPORT	SAE AIR4543	REV. B
		Issued 1992-01 Revised 2010-11 Superseding AIR4543A	

(R) Aerospace Hydraulics and Actuation Lessons Learned

RATIONALE

This AIR is revised to:

- Reformat the Lessons Learned from Revision A to conform with the current SAE Aerospace Technical Report Style Manual, dated March 2010.

- Restructure the Revision A material to conform with a Table of Contents structure approved by the responsible A-6 committee during the Fall 2008 A-6 meeting.

TABLE OF CONTENTS

1. SCOPE

This SAE Aerospace Information Report (AIR) contains Lessons Learned from aerospace actuation, control and fluid power systems technologies. The lessons were prepared by engineers from the aerospace industry and government services as part of SAE Committee A-6, Aerospace Fluid Power, Actuation, and Control Technologies, and were presented to the A-6 during meetings held from 1989 through 1999. The document is organized into five sections covering systems, actuation, hydromechanical components, electrical components and miscellaneous, each further divided into subsections. The lessons are presented in a concise format of Problem, Issue, Solution and Lesson Learned, often with accompanying descriptive diagrams and illustrations for clarity and understanding.

Because of the potential growth in the size of the document as new lessons are published, those presented to the A-6 Committee in 2000 and later years are planned to be released in separate slash number documents, AIR4543/1, AIR4543/2 and so on.

1.1 Further Information About this Revision

The attempt has been made to transfer all of the technical information unchanged from the Revision A to this Revision B document. Several spelling and other typos have been corrected and a duplicate of one lesson was deleted. Two other lessons, which have been revised and repeat presented to A-6 on dates close to the publication for this revision, have also been deleted and will reappear in the first slash number document.

NOTE: 1. The Lessons Learned contained in this document were first published a minimum of 11 years before the publication date. The reader is advised that some may have been overtaken by technology and developing industry practice.

2. The allocation of the lessons to the various categories is typically obvious, because of their focused content, but occasionally arbitrary because of mixed content. Text searches are suggested to ensure that all lessons of interest are found.

2. REFERENCES

2.1 Applicable Documents

The following publications form a part of this document to the extent specified herein. The material in this document is informational and historical. Many of the Lessons Learned were originally published many years before the published date of this revision and the incident dates and relevant issues of the referenced publications are not known. Some of them have been superceded by other standards and this information is provided by parenthetical statements in the body of the document. Both the superceded publications and those current at the time of publication are listed below.

2.1.1 SAE Publications

Available from SAE International, 400 Commonwealth Drive, Warrendale, PA 15096-0001. Tel: 877-606-7323 (inside USA and Canada) or 724-776-4970 (outside USA), www.sae.org.

AIR1243 Anti Blow-By Design Practice for Cap Seals

AIR1569 Handling and Installation Practice for Aerospace Hose Assemblies

AMS-P-83461 Packing, Preformed, Petroleum Hydraulic Fluid Resistant, Improved Performance at 275 °F (135 °C)

AMS-R-83485 Rubber, Fluorocarbon Elastomer, Improved Performance at Low Temperatures

ARP4150 Procedure for Inspection of Inservice Airborne Accumulators for Corrosion and Damage

ARP4378 Aerospace - Accumulator, Hydraulic, Welded Bellows, Factory Precharged

ARP4379 Aerospace - Accumulator, Hydraulic, Cylindrical, Piston Separated

ARP4553 Aerospace - Accumulator, Hydraulic, Self-Displacing

AS604 Hose Assembly, Polytetrafluoroethylene, Metallic Reinforced, 3000 psi, 400 °F, Heavyweight, Hydraulic

AS620 Hose Assemblies, Convoluted Polytetrafluoroethylene Metallic Reinforced, High Temperature, Medium Pressure, Aircraft

AS1241 Fire Resistant Phosphate Ester Hydraulic Fluid for Aircraft

AS1339 Hose Assembly, Polytetrafluroethylene, Metallic Reinforced, 3000 psi, 400 °F, Lightweight, Hydraulic and Pneumatic

AS4059 Aerospace Fluid Power - Cleanliness Classification for Hydraulic Fluids

AS4623 Hose Assembly, Polytetrafluoroethylene, Para-Aramid Reinforced, 3000 psi, 275 °F, Heavy Duty, Aircraft Hydraulic Systems

AS4716 Gland Design, O-Ring and Other Elastomeric Seals

AS5440 Hydraulic Systems, Aircraft, Design and Installation Requirements For

AS8775 Hydraulic System Components, Aircraft and Missiles, General Specification For

AS28775 Packing, Preformed, - MS28775 O-Ring

AS83461/1 Packing, Preformed, MS83461 O-Ring

AS83461/2 Packing, Preformed, MS83461 Straight Thread Tube Fitting Boss

2.1.2 AIA Publications

Available from Aerospace Industries Association, 1000 Wilson Boulevard, Suite 1700, Arlington, VA 22209-3928, Tel: 703-358-1000, www.aia-aerospace.org.

NAS1638 Cleanliness Requirements of Parts Used in Hydraulic Systems (Inactive for new design)

2.1.3 ASTM Documents

Available from ASTM International, 100 Barr Harbor Drive, P.O. Box C700, West Conshohocken, PA 19428-2959, Tel: 610-832-9585, www.astm.org.

ASTM F1940 Standard Test Method for Process Control Verification to Prevent Hydrogen Embrittlement in Plated or Coated Fasteners

2.1.4 FAA Publications

Available from Federal Aviation Administration, 800 Independence Avenue, SW, Washington, DC 20591. Tel: 866-835-5322, www.faa.gov.

14CFR Part 25 Code of Federal Regulations, Part 25 Airworthiness Standards: Transport Category Airplanes

2.1.5 Joint Aviation Authorities Committee Publications

Available from Global Engineering Documents, 15 Inverness Way East, Englewood, Colorado, 80112-5776. Tel: 1-800-854-7179 (inside USA and Canada) or 1-303-397-7956 (outside USA), http://global.ihs.com

JAR 25 Joint Airworthiness Requirements, Large Aeroplanes

2.1.6 US Government Publications

Available from the Document Automation and Production Service (DAPS), Building 4/D, 700 Robbins Avenue, Philadelphia, PA 19111-5094, Tel: 215-697-6257, http://assist.daps.dla.mil/quicksearch/.

MIL-DTL-5498 Accumulators, Hydraulic, Cylindrical 3000 psi, Aircraft

MIL-F-8815 Filter and Filter Elements, Fluid Pressure, Hydraulic Line, 15 micron absolute and 5 micron absolute, Type II Systems, General Specification for

MIL-F-8815/2 Filter, Fluid, Pressure, Hydraulic Line, 3000 psi, Absolute 15 Micron, Style B, Non Bypass, -65 Degrees to +275 Degrees F (Inactive for new design.)

MIL-F-24402 Filters (Hydraulic), Filter Elements (High Efficiency), and Filter Differential Pressure Indicators, General Specification for

MIL-G-5514 Gland Design; Packings, Hydraulic, General Requirements for (Inactive for new design)

MIL-P-25732 Packing, Preformed, Petroleum Hydraulic Fluid Resistant, Limited Service at 275 Deg. F (135 Deg. C) (Inactive for new design)

MIL-PRF-680 Degreasing Solvent

MIL-PRF-5606 Hydraulic Fluid, Petroleum Base; Aircraft, Missile and Ordnance (Inactive for new design)

MIL-PRF-6083 Hydraulic Fluid, Petroleum Base, for Preservation and Operation

MIL-PRF-17331 Performance Specification, Lubricating Oil, Steam Turbine and Gear, Moderate Service

MIL-PRF-46170 Hydraulic Fluid, Rust Inhibited, Fire Resistant, Synthetic Hydrocarbon Base, NATO Code No. H-544

MIL-PRF-81836 Cleanliness Requirements of Parts Used in Hydraulic Systems

MIL-PRF-83282 Hydraulic Fluid, Fire Resistant Synthetic Hydrocarbon Base; Aircraft,

MIL-PRF-87257 Hydraulic Fluid, Fire Resistant Synthetic Hydrocarbon Base; Aircraft

MIL-R-25988 Rubber, Fluorosilicone Elastomer, Oil- and Fuel Resistant, O-Rings

MS27595 Retainer, Packing Backup, Continuous Ring, Polytetrafluoroethylene (Inactive for new design)

MS28774 Retainer, Packing Backup, Single Turn, Polytetrafluoroethylene (Inactive for new design)

2.1.7 Referenced Publications Now Canceled

The following documents are referenced in the legacy Lessons Learned contained in this document. In each case they were valid when the Lesson was first published but should now be disregarded since they have been canceled and usually superceded. The superceding documents are provided in the body of the Lesson through parenthetical notes and listed in the preceding subsections.

NOTE: The original references have been retained so that the reader can access the author's intent if any of the lessons are found to conflict with the superceding document.

Canceled SAE Publications

AIR4150 Inspection of Inservice Airborne Accumulators for Corrosion and Damage

Canceled Government Specifications and Standards

MIL-H-5440 Hydraulic Systems, Aircraft, Design and Installation Requirements for

MIL-H-8775 Hydraulic System Components, Aircraft and Missiles, General Specification For

MIL-P-83461 Packing, Preformed, Petroleum Hydraulic Fluid Resistant, Improved Performance at 275 deg F (135 deg C)

MIL-P-87175 Packing, Preformed, Petroleum Hydraulic Fluid Resistant, Phosphonitrilic Fluoroelastomer

MIL-R-83485 Rubber, Fluorocarbon Elastomer, Improved Performance at Low Temperatures

MS28775 Packing, Preformed, Hydraulic, +275 deg F ("O" Ring)

P-D-680 Dry Cleaning and Degreasing Solvent

2.2 Definitions

Refer to ARP4386.

3. TECHNICAL INFORMATION

3.1 Systems Lessons Learned

3.1.1 All Systems

This subsection addresses lessons generally applicable to systems of all types.

3.1.1.1 Biased Trade Studies

PROBLEM: Trade Studies Biased by Omissions

ISSUE: Trade studies often fail to assess all influences on the air vehicle. Frequently in such trade studies the following issues are not compared:

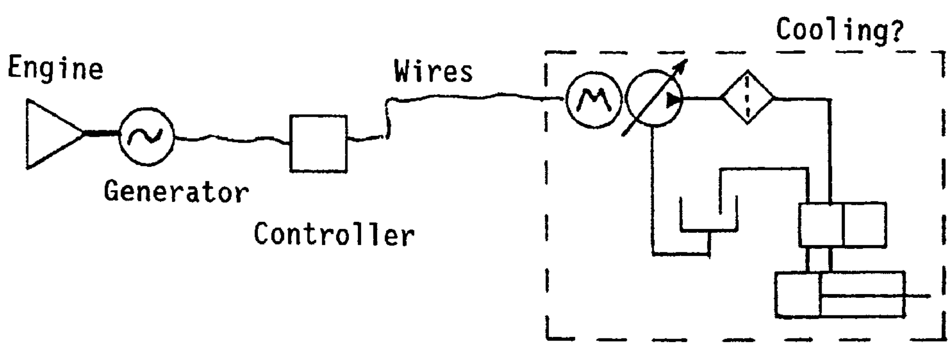
a. Cooling

b. Power transmission

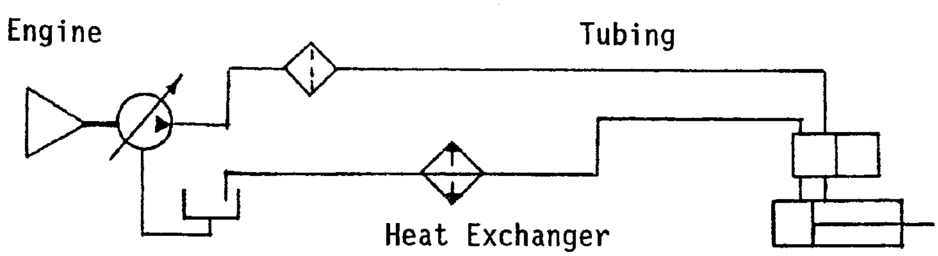
c. Power conversion

ILLUSTRATION:

Distributed hydraulic packages:



Centralized hydraulic system



SOLUTION: Conduct the trade study from the point of original power generation (usually the engine) to the point of final utilization.

3.1.2 Flight Control Systems

3.1.2.1 Effect of JAR/FAR 25.671 Requirement

PROBLEM: Spoiler surface operation not compliant with JAR/FAR 25.671 requirement.

ISSUES:

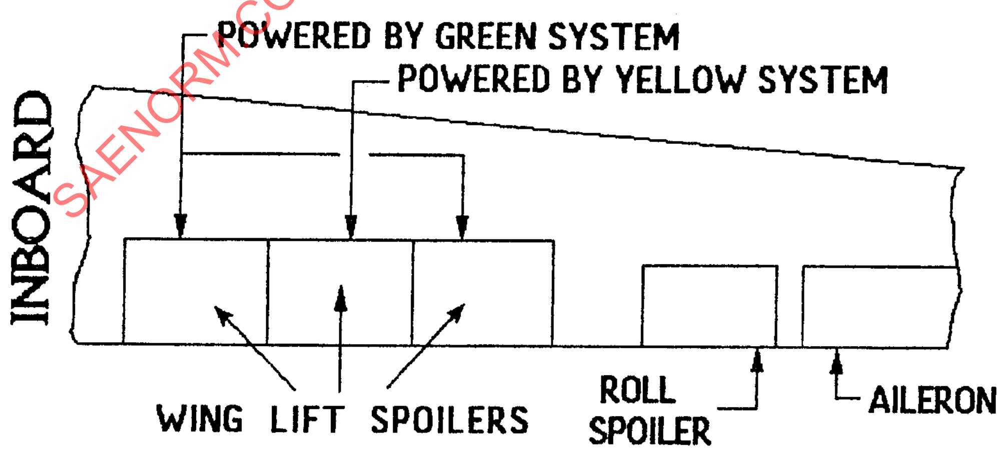
a. Noncompliance with JAR/FAR 25.671.

[The reference "JAR/FAR 25.671" is to paragraph 25.671 from 14CFR Part 25 and JAR 25 Ed.]

b. A single failure of a flight control surface attachment point could cause a catastrophic event.

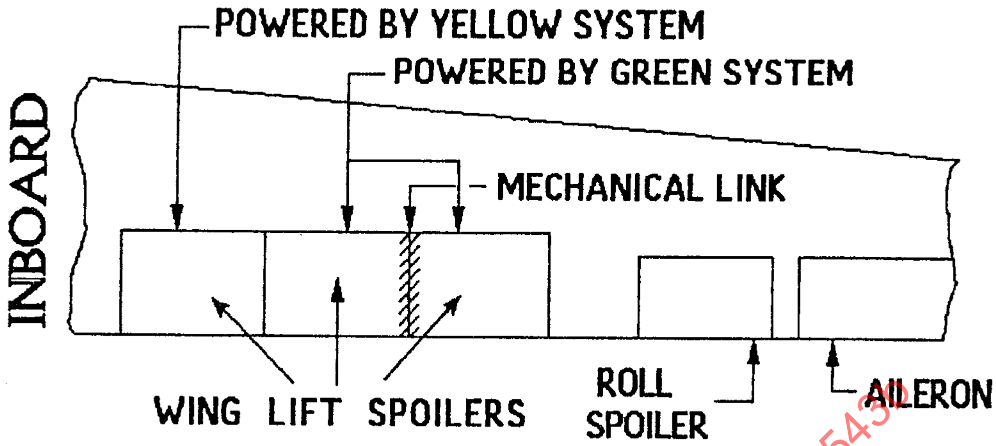
c. The center or outboard ground lift spoiler panel could deploy, following the failure of the eye end attachment point for example, such that an inadvertent rolling moment could not be controlled.

ILLUSTRATION:



SOLUTION: Reroute hydraulic supplies to the lift spoilers and link the center and outboard panels.

ILLUSTRATION:



3.1.2.2 Failure of all Four Hydraulic Systems

PROBLEM: Failure of all four hydraulic systems due to blowout of the aft bulkhead. (Normal flight control was lost and the airplane crashed into a mountain.)

ISSUES:

a. The bulkhead was replaced and mistakenly fastened with a single row of rivets rather than the required double row.

b. The released cabin air over pressurized the fin, causing failure of the hydraulic lines.

SOLUTION:

a. The access port from the tail section (aft of the bulkhead) was closed.

b. A hydraulic fuse was installed in one of the four hydraulic pressure lines leading to the fin.

3.1.2.3 Flight Control Attachment Bolt Failure

PROBLEM: Flight control attachment bolt failure – 3/16 in.

ISSUE: Critical flight control linkage bolt joints using 3/16 in bolts may be over torqued and stay together but fail later in service.

SOLUTION: Use 1/4 in or larger diameter bolt for all critical linkage joints.

3.1.3 Power Distribution Systems

3.1.3.1 Power Conversion Efficiency

PROBLEM: Multiple power conversions degrade efficiency.

ISSUE: Efficiency is lost for each of the following power conversions:

a. Mechanical

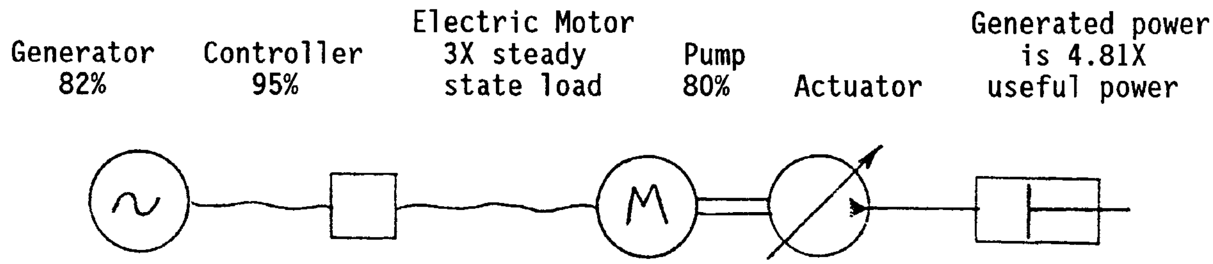
b. Electrical

c. Hydraulic

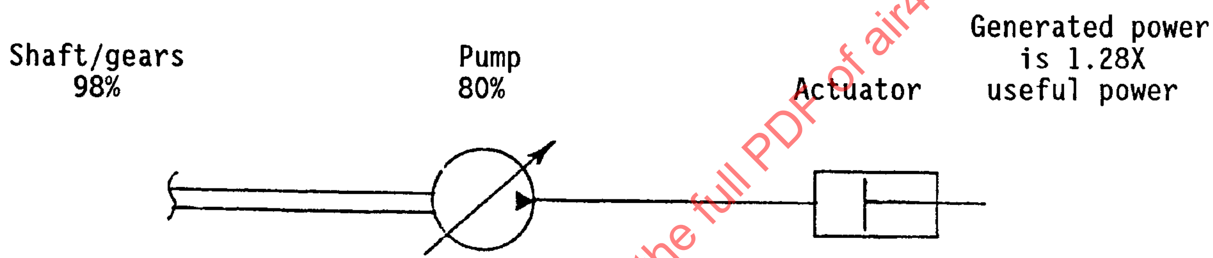
d. Pneumatic

ILLUSTRATION:

Hydraulic power from an all-electric engine:



Hydraulic power from the engine shaft:



SOLUTION: Minimize the number of energy conversions between source and use.

3.1.3.2 Pump Overheat and Rapid Wear

PROBLEM: Pump overheat and rapid wear installed in aircraft system.

ISSUE:

a. Pump case drain line too small

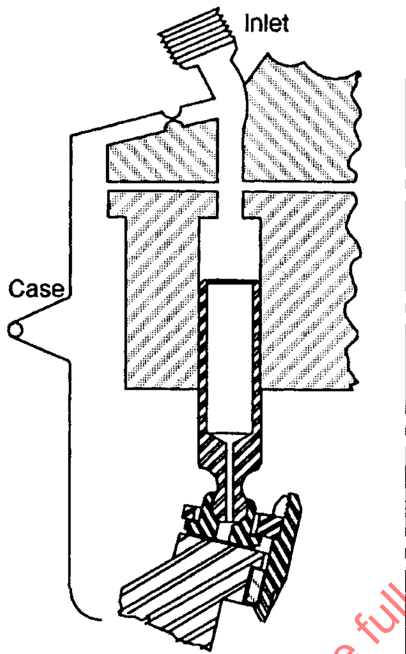
b. Creates high back pressure (case)

c. Creates high discharge pressure (flow)

d. Overheat (pump)

e. Excessive pump shoe wear

ILLUSTRATION:



SOLUTION:

a. Increase line size (case drain)

b. Use bypass type filters

c. Reroute line to reduce pressure drop

d. Internally drain (to inlet) if minimum system demand available

3.1.3.3 Excessive Pump Pressure Ripple

PROBLEM: Hydraulic system life and noise degraded by excessive pump pressure ripple.

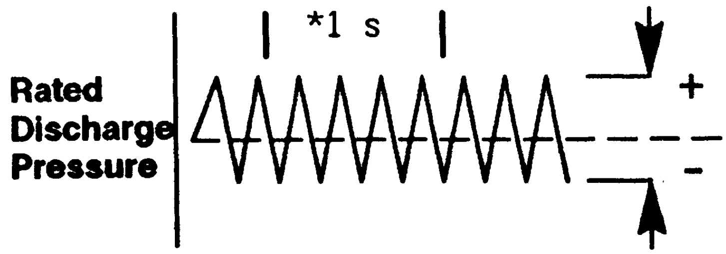
ISSUE:

a. Line fatigue

b. Excessive wear (moving parts)

c. Vibration/noise

ILLUSTRATION:



NOTE: Ripple amplitude: Acceptable +/- 10%, Desirable +/- 5 %, Achievable +/- 2%

$$\text{Ripple Frequency} = \left(\frac{\#\text{Pistons} \times \text{rpm}}{60}\right)\text{cps}$$

SOLUTION:

a. Generate dynamic analysis of total system

b. "Tune" pump to match system compliance

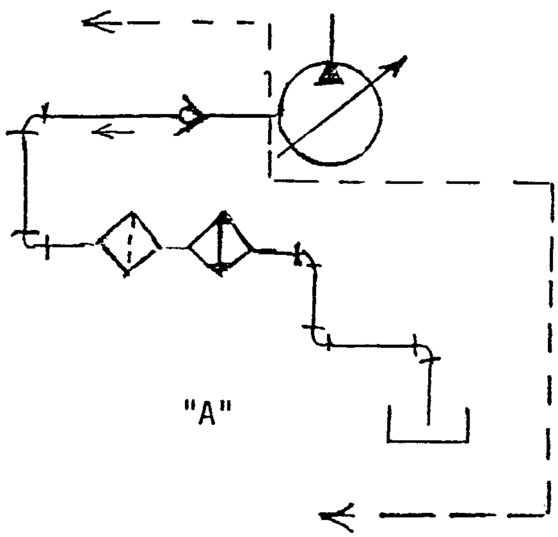
c. Provide matched attenuator (pump to system)

3.1.3.4 Pump or Motor Case Drain Line Design

PROBLEM: Pump or motor life reduced by case drain line design.

ISSUE: Restrictions in the hydraulic pump/motor case drain plumbing can cause high pressure and subsequent failure of the pump/motor case. Restrictions can be caused by too small diameter tubing, long tubing lengths, or improper component installations. The restrictions can result in high line pressure drop, resulting in high pressures in the pump or motor case. Pump and motor cases are generally designed to a 500 lbf/in^2 (3450 kPa) proof pressure or less.

ILLUSTRATION: - Typical "Case Circuit":



SOLUTION:

a. Analyze the installation and determine the pressure drop of the pump or motor case drain plumbing from the case to the reservoir. The analysis should include evaluations considering delta pressure of all possible case drain flow and fluid temperature combinations, case drain filter variables, transient flows, and component installation.

    1. The pressure drop considerations. A, include:

    2. Line length, line size, bends, fittings,check valve, reservoir, heat exchanger

    3. The total pressure drop A summation must be the same or lower than the pump/motor case allowed design operating pressure.

    4. All case drain temperature/flow combinations must be considered in pressure drop calculation. Some design margin should be included to reduce the effects of pressure spikes in the pump or motor case.

b. Use maximum size tubing and short line lengths. Plumbing to the hydraulic return lines should be considered if decreased pressure levels are attained.

c. Conduct the test if possible to verify the analysis prior to design freeze.

3.1.3.5 Pump Cavitation

PROBLEM: Pump cavitation causing hydraulic system failures.

ISSUE:

a. High levels of pressure oscillation have resulted from:

    1. Poor maintenance (excessive air in system)

    2. Pump cavitation due to system malfunction

b. Bracket and hydraulic line failures between the engine driven pumps and the filter module have occurred.

SOLUTION:

a. Install an attenuator in the pump outlet line to reduce oscillations during cavitation.

b. Tune the attenuator to the system configuration.

c. Remove all free air from system.

3.1.3.6 Pump Suction Line Bursting

PROBLEM: Pump suction line bursting.

ISSUE:

a. On the initial engine run of a twin engine aircraft the R/H pump suction line burst.

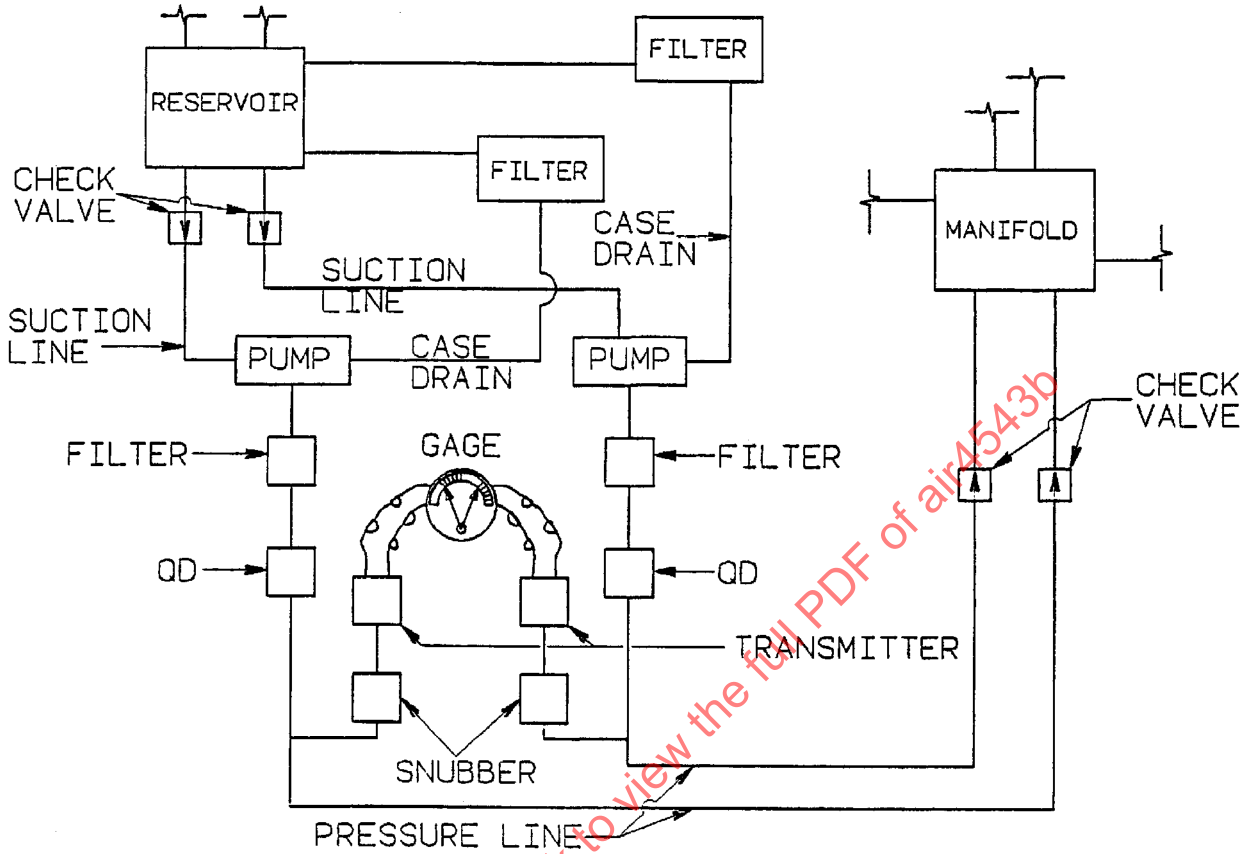
b. The line was replaced and the R/H engine started - no problem.

c. The L/H engine was started and the R/H suction line burst.

d. The line was replaced and the L/H engine started - no problem.

e. The R/H engine was then started and the R/H suction line burst.

ILLUSTRATION:



SOLUTION:

a. The R/H pump compensator was found to have a slightly lower pressure setting than required.

b. A failed pump outlet check valve created a reverse pumping situation on the R/H side.

c. Pump and check valve were replaced.

3.1.3.7 Hydraulic System Noise

PROBLEM: Hydraulic system noise.

ISSUES:

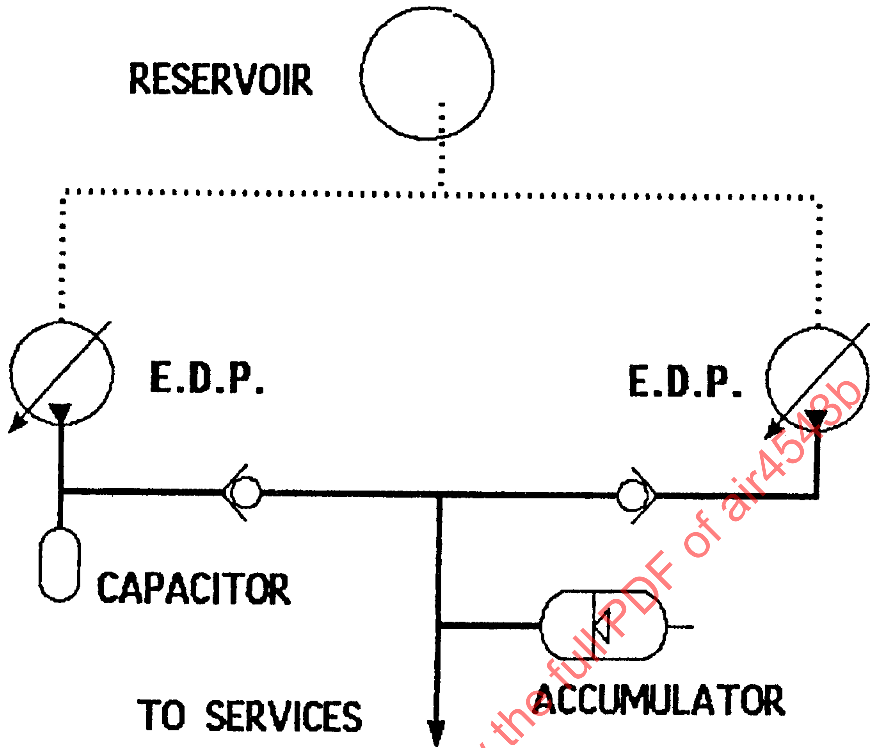
a. With a twin engine driven pump (E.D.P.) hydraulic system, there were complaints of noise, although not encountered with:

   1. Earlier production aircraft;

   2. When system is pressurized by ground rigs.

b. Resonance condition occurred in the shortest line downstream of E.D.P. due to the reduced volume of the high pressure part of the hydraulic system.

ILLUSTRATION: - Simplified System Diagram



SOLUTIONS:

a. Initially increase the accumulator size to that previously fitted.

b. Finally, revert to a smaller accumulator and add a “capacitor” (a small pressure vessel (22 in^3) (360 cm^3) to the system).

3.1.3.8 Low Pressure Cavitation During Valve Endurance Test

PROBLEM: Low pressure cavitation during valve endurance test.

ISSUE: As the demand valve cycled, the return line in the test circuit would pull a vacuum when flow was abruptly shut off from 25 gal/min (1.6 L/s). The valve cap eroded through in about 35 000 cycles. The area of the valve which eroded was exposed only to 60 lbf/in^2 (410 kPa) maximum and a negative 10 lbf/in^2 (70 kPa).

SOLUTION:

a. A relief valve set at 150 lbf/in^2 (1030 kPa) was put into the return line of the circuit to maintain a positive pressure in the area. This solved the problem.

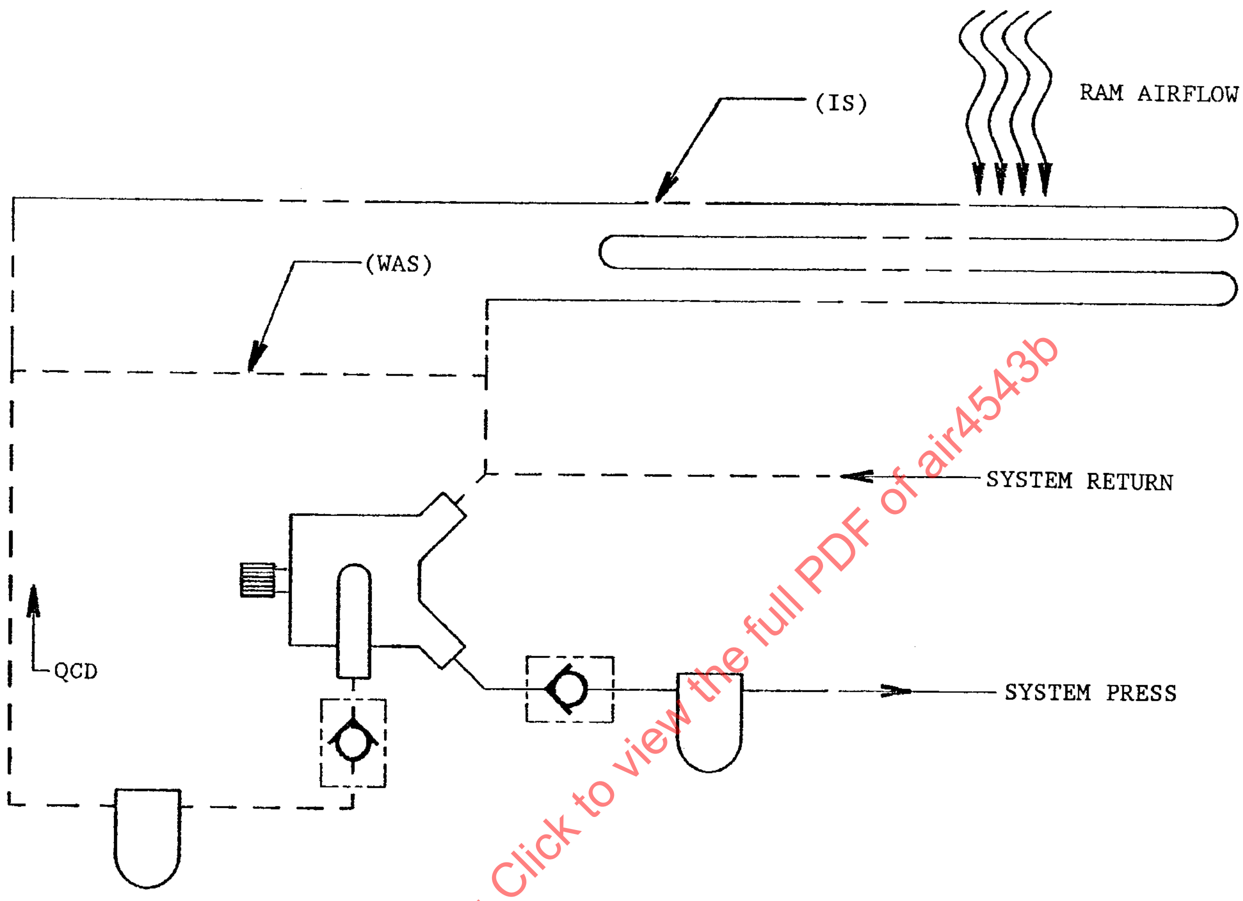
b. A later metallurgical analysis confirmed the low pressure cavitation.

3.1.3.9 System Overheat

PROBLEM: System overheat caused by inefficient pump and lack of heat exchanger.

ISSUE: System can exceed 275 °F (135 °C)

ILLUSTRATION:



SOLUTION:

a. Revise the specifications to buy the pump with guaranteed efficiency and case drain flow over the life of the pump.

b. Increase the case drain line diameter and the filter housing size.

c. Reroute the case drain line to increase surface area and also expose to area with ram air.

3.1.3.10 Spurious Hydraulic Reservoir Warning

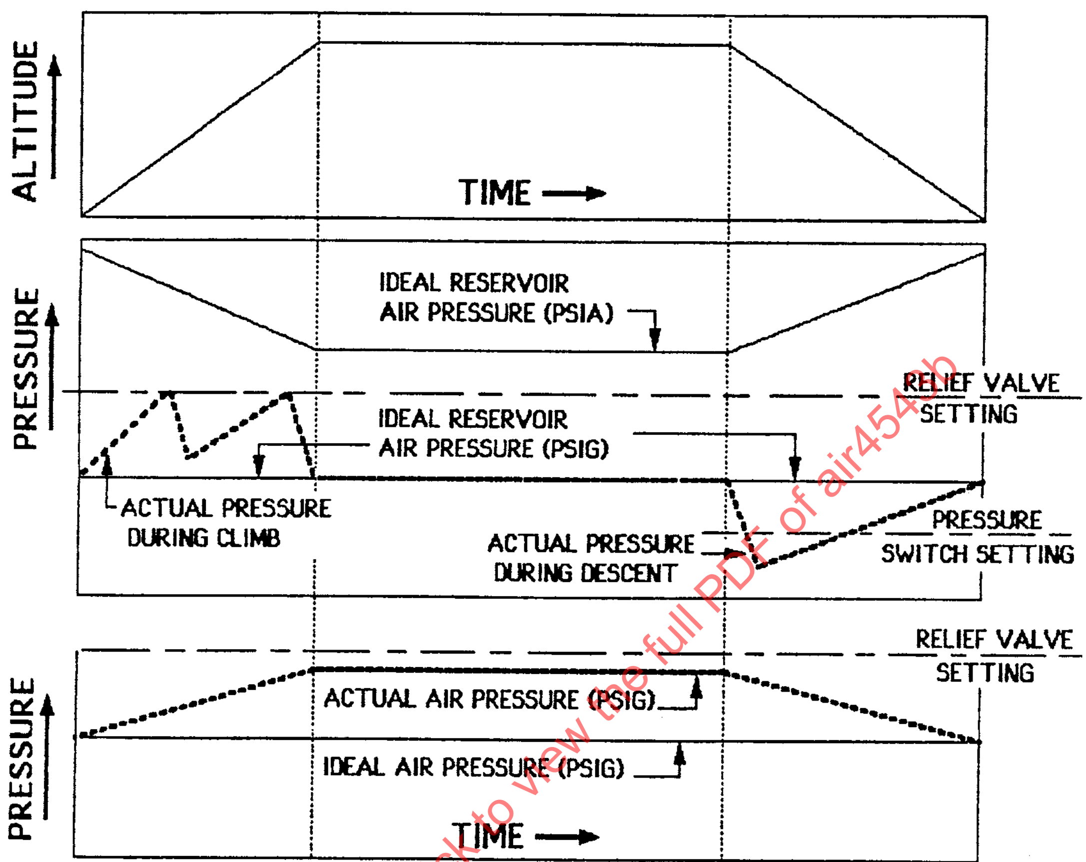
PROBLEM: Spurious hydraulic reservoir low air pressure warning.

ISSUE:

a. During aircraft descent, hydraulic reservoir low air pressure warnings occurred although the reservoir pneumatic pressurization system was fully serviceable.

b. The engine bleed air pressure with engines at flight idle was less than the pressure switch setting.

ILLUSTRATION:



SOLUTION: Either reset the pressure switch to be less than the minimum bleed pressure or set the relief valve so that it does not operate during aircraft climb.

3.1.3.11 Tire Burst in the Wheel Well

PROBLEM: Hydraulic system failures due to tire burst in the wheel well.

ISSUE: FAR 729 (f) Requirement: Protection of equipment in wheel wells. Equipment that is essential to safe operation of the airplane and that is located in wheel wells must be protected from damaging effects of (1) a bursting tire, unless it is shown that a tire cannot burst from overheat; and (2) a loose tire tread, unless it is shown that a loose tire tread cannot cause damage. Tire overheating due to hard braking or dragging brake prior to takeoff causes the tire to blow out at a weak spot in the tread. The direct jet impingement and the wheel well overpressure can fail hydraulic tubing and blow off the wheel well doors.

[The reference "FAR 729" is to paragraph 25.729 from 14CFR Part 25 Ed.]

SOLUTION: Since this event cannot be prevented, critical hydraulic components and tubing must be located and routed to provide the best possible protection. In aircraft with multiple hydraulic systems, system separation is essential in order to meet requirements. One has to remember that the energy levels due to a tire or wheel blow out are potentially so high that shielding may not be effective and may even cause more damage than the tire blow out. System separation around wheels and tires is a must for all operating aircraft systems such as pneumatics, fuel, controls, electrical, avionics.

3.1.3.12 Failure of Both Hydraulic Systems due to Landing Gear Failure

PROBLEM: Failure of both hydraulic systems due to failure of a landing gear link.

ISSUE: When the link failed with the gear retracted in flight, it swung into tubing in both hydraulic systems causing their failure, and loss of the aircraft.

SOLUTION: Provide adequate system separation so that no such equipment failure can damage all hydraulic systems.

3.1.3.13 Inadvertent Actuation of Switches and Valves in High Response Servoactuators

PROBLEM: Inadvertent actuation of ΔP switches or ΔP operated valves in servoactuators for high response rate flight controls.

ISSUE: A pressure spike occurs in the return system at the servo, due to the initiation of a high return flow rate from the servoactuator.

SOLUTION:

a. Return system tubing should be sized with consideration for ΔP required to accelerate the fluid.

b. ΔP switches and ΔP operated valves should be designed with damping to prevent problems from pressure transients.

3.1.3.14 Hydraulic Component Accessibility

PROBLEM: Servicing difficult because of poor hydraulic component accessibility.

ISSUE: Excessive time is required to service frequent maintenance components such as reservoirs, accumulators and system filters in the wheel well or engine compartment.

SOLUTION: In arriving at a location for installation of reservoirs, accumulators and system filters one must keep in mind the need for servicing these components. Provide good accessibility to permit the servicing in the shortest time possible. Keep in mind never to require removal of components in order to service another component.

3.1.4 Utility Control Systems

This subsection applies to systems that provide control of non-critical functions for aircraft and other vehicles.

3.1.4.1 Hand Pump Suction Line

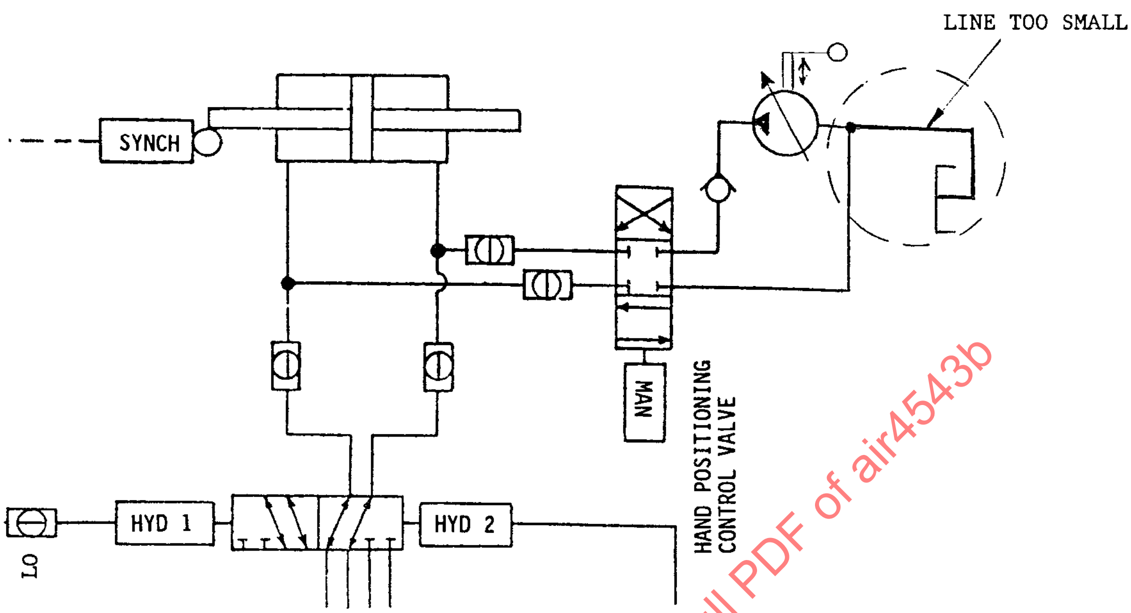
PROBLEM: Inadequately sized hand pump suction line causes unacceptable performance.

ISSUES:

a. The hand pump doesn't operate the submarine steering cylinder at the required rate.

b. The pump operates satisfactorily in the shop but not when installed in the ship.

ILLUSTRATION:



SOLUTION: The variable displacement hand pump discharges fluid as the pump handle is stroked in each direction. However, analysis indicated that the pump takes suction only when the pump is stroked in one direction. As a result, considerable flow is required from the unpressurized reservoir during the suction stroke even though the cylinder is balanced (equal volume for each direction of travel). The diameter of the make-up line from the reservoir to the pump suction had been sized for minimal flow and had to be increased to obtain rated pump capacity.

3.1.4.2 Sequencing of System Functions

PROBLEM: Unwanted simultaneous or out-of-order sequencing of system functions.

ISSUE: Failures in the electrical control logic (limit switches, coils, relays) and wiring.

SOLUTION: Mechanical sequencing of hydraulic valves

a. Push-pull rods

b. Bell cranks

c. Cables

[Note that this belief in mechanically controlled processes was widely held in the 70s but in later years, as technology advanced, gave way to the almost universal use of electronic control Ed.]

3.2 Actuation Lessons Learned

3.2.1 Electric Servoactuators and Actuators

This subsection will contain Lessons Learned in the design and development of electrically powered servoactuators and actuators.

3.2.2 Hydraulic Servoactuators and Actuators

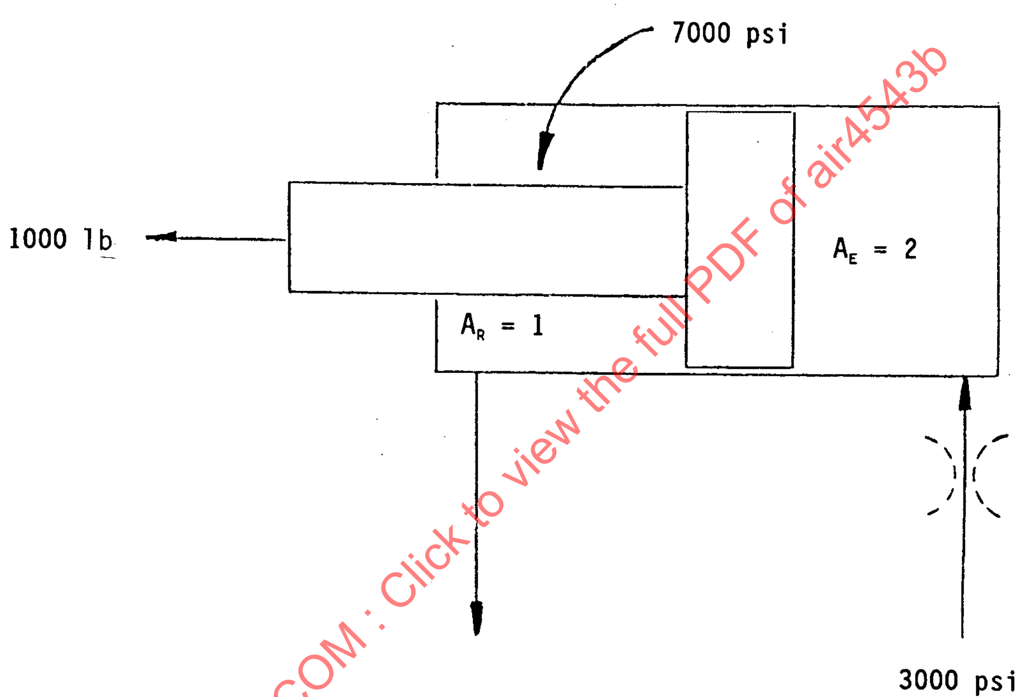
This section contains Lessons Learned relating to hydraulically powered servoactuators and actuators.

3.2.2.1 Pressure Intensification

PROBLEM: Pressure intensification in extending cylinder.

ISSUE: There have been many cases where designers failed to recognize the high pressure that can occur on the retract side of a cylinder due to pressure intensification and/or an aiding load.

ILLUSTRATION:



NOTE: The illustration values of 7000 lbf/in^2, 3000 lbf/in^2 and 1000 lbf are equivalent to 48 300 kPa, 20 700 kPa and 4450 N, respectively.

SOLUTION: Include restrictor in the extend line so that the extend pressure is reduced to an acceptable level.

3.2.2.2 Servovalve Separation from the Manifold

PROBLEM: Servovalve separation from the manifold due to servoactuator velocity limiting.

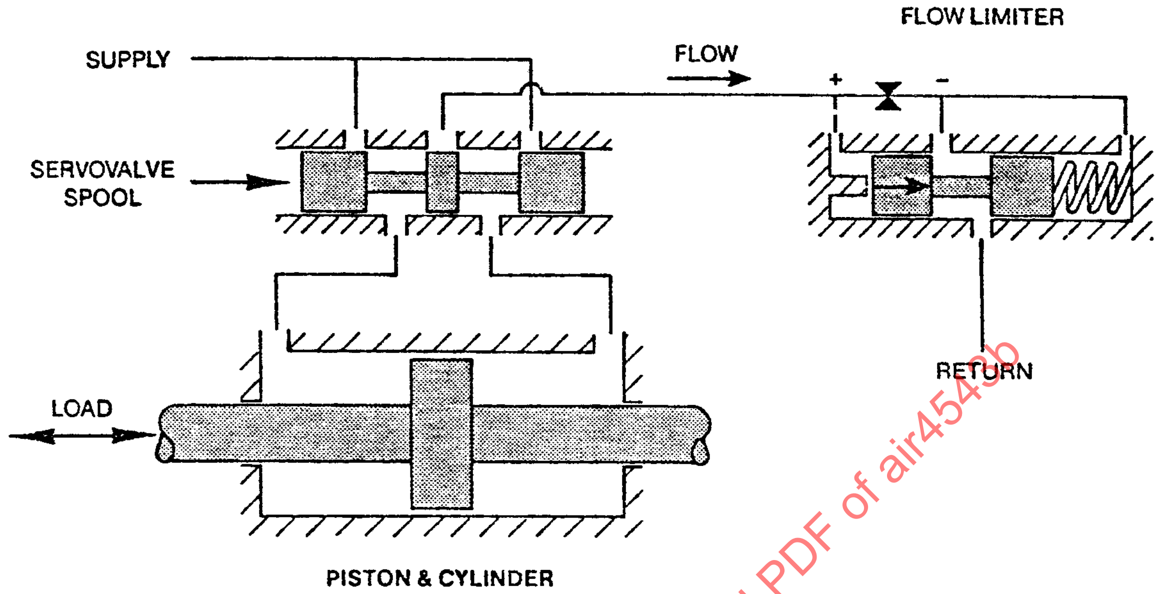
ISSUE:

a. During the countdown for the initial launch of the Gemini two-man space vehicle, the primary servovalve of the dual redundant thrust vector control actuator blew off the manifold.

b. The actuator flow limiter used to restrict excessive thrust vector control rate caused high back pressure during engine start transient.

c. The high impulse loads can create transient back pressure on the servovalve. Alternate configurations for limiting actuator velocity should be considered if impulse loads cannot be suppressed.

ILLUSTRATION:



SOLUTION:

a. A quick fix was to beef-up the servovalve body and mounting.

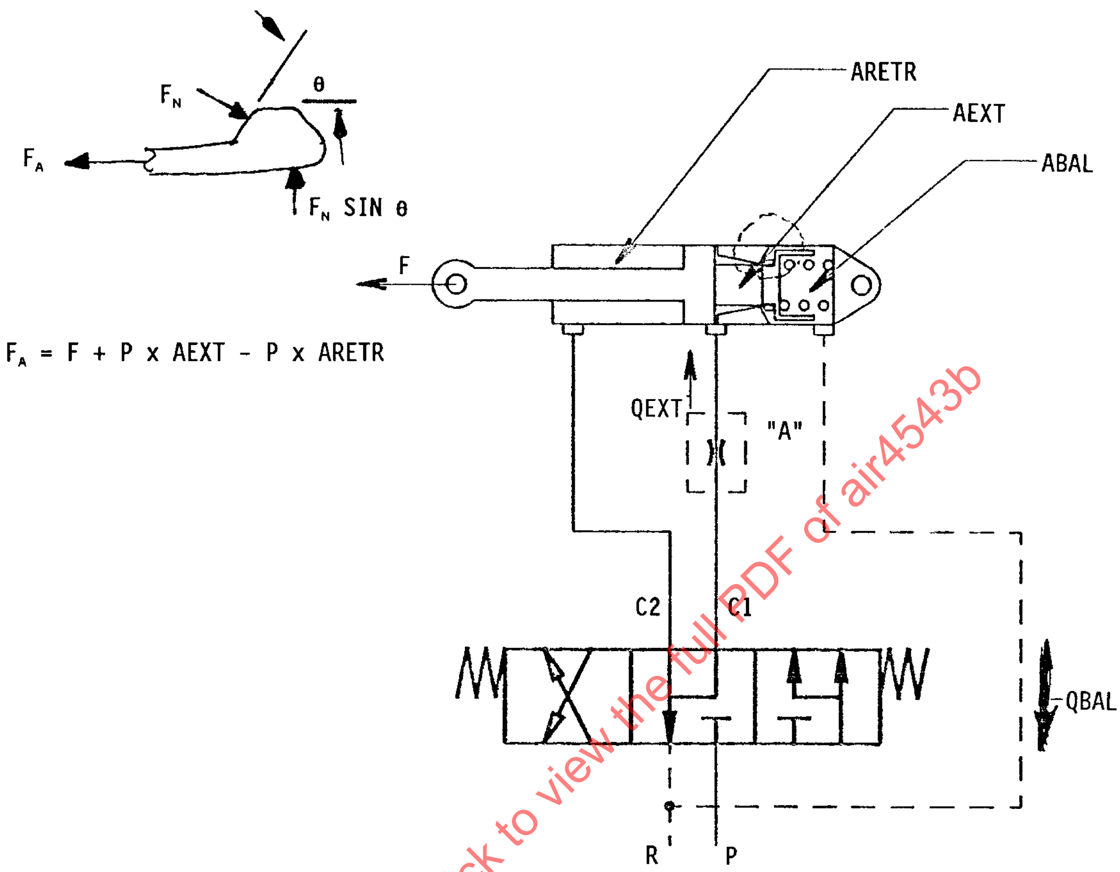
b. A long-term solution has been to add cylinder bypass (pressure relief) valves to dissipate energy of hard engine/rocket motor starts.

3.2.2.3 Galling of the Cylinder Lock Mechanism

PROBLEM: Galling of the cylinder lock mechanism when unlocked under high loads.

ISSUE: The contact area approaches zero at the point of lock release.

ILLUSTRATION:



SOLUTION:

a. Size port and line to ABAL for high transient flow.

b. Relieve external F on fingers by application of retract pressure before application of extend pressure.

c. Choke the inlet flow QEXT at "A" to reduce the rate of pressure rise to AEXT.

d. Use one selector valve per actuator.

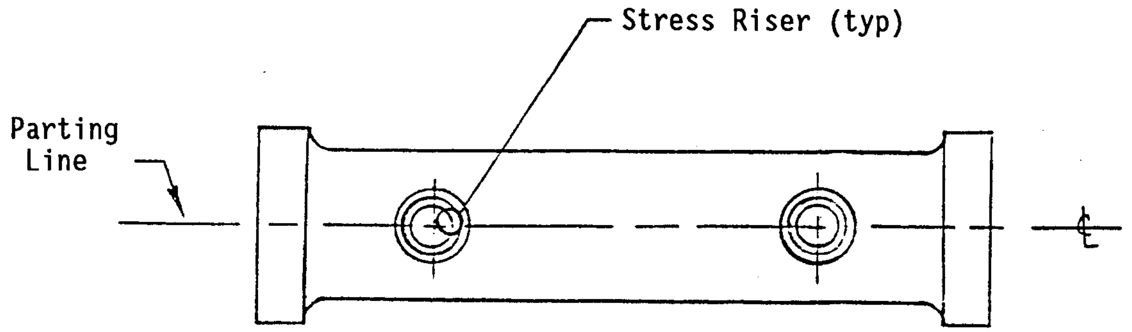
e. Guide and lock piston hardness to be much higher than lock fingers.

3.2.2.4 Parting Plane Fatigue Failures in Cylinders

PROBLEM: Fatigue failures in cylinders with parting plane through ports.

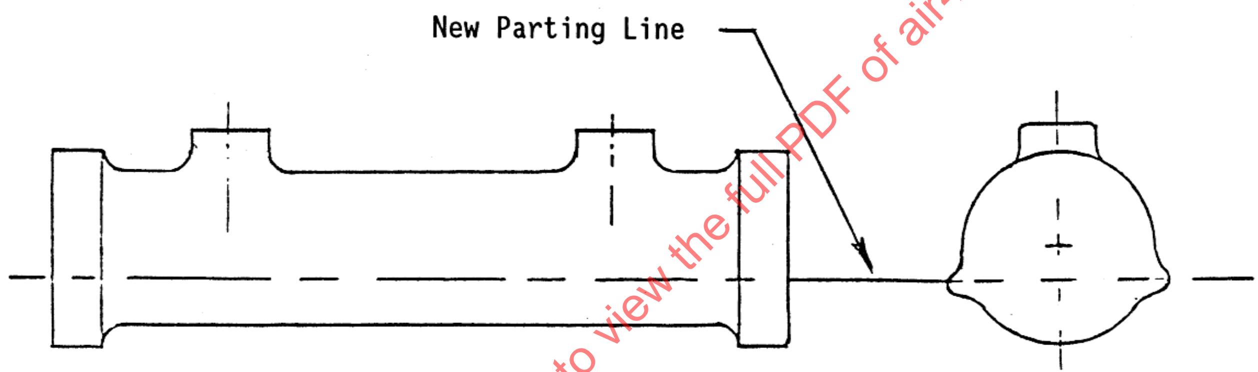
ISSUE: The parting line placement on the cylinder centerline for "shallow draw" forging creates stress risers at the ports and at the thinnest section of the cylinder. This can result in fatigue failures or shortened life (see Illustration 1).

ILLUSTRATION: - 1



SOLUTION: Relocate the parting line off the cylinder centerline and completely off the ports (see Illustration 2).

ILLUSTRATION: - 2

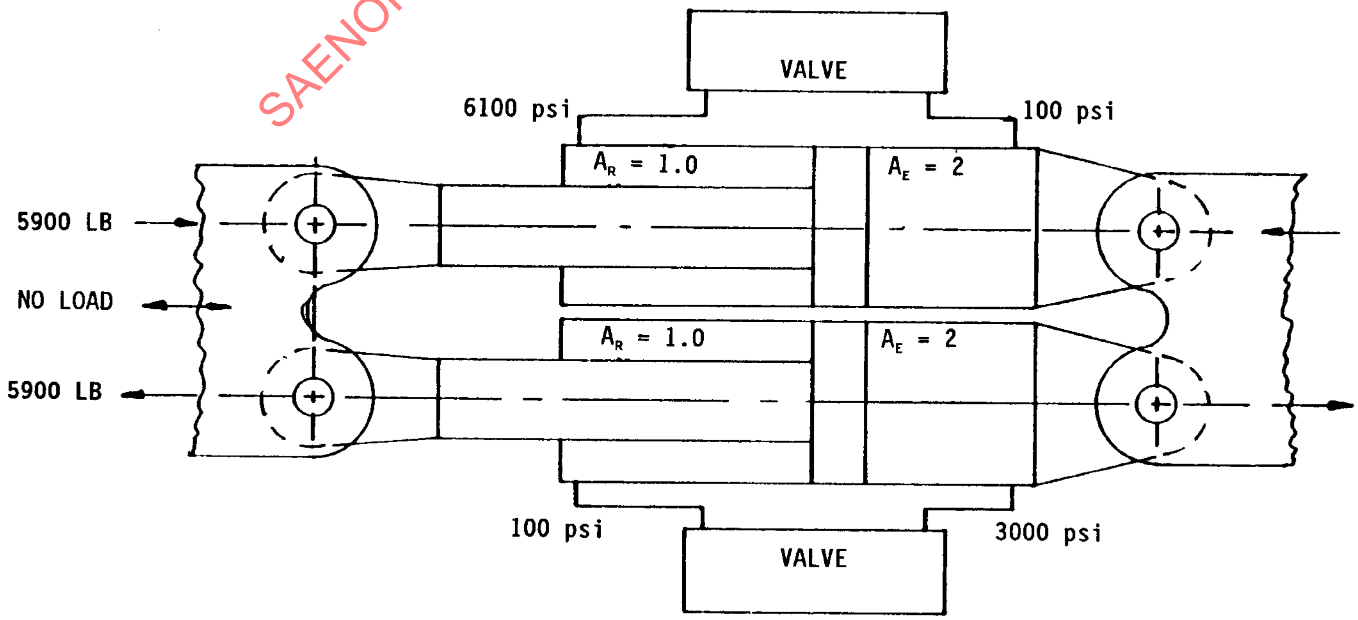


3.2.2.5 Fatigue Damage to Dual Side-by-Side Cylinders

PROBLEM: Fatigue damage to loaded structure and cylinders of side-by-side cylinders.

ISSUE: Force fight in side-by-side cylinders driven by separate valves.

ILLUSTRATION:



NOTE: The pressures and loads, 6100 lbf/in^2, 3000 lbf/in^2, 100 lbf/in^2 (two places) and 5900 lbf (two places), referenced in the illustration are equivalent to 42 100 kPa, 20 700 kPa, 690 kPa (two places) and 26 200 N (two places), respectively.

SOLUTION:

a. Use a single tandem valve for control pressure matching.

b. Mechanically couple separate valves for control pressure matching.

c. Use pressure transducers and electronic equalization to match control pressures.

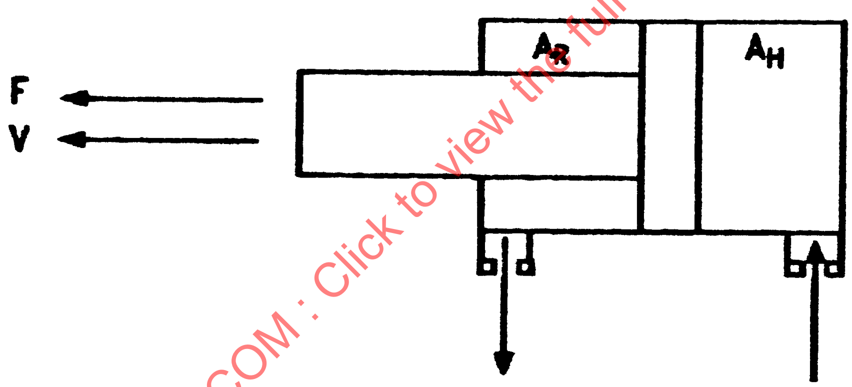
d. Use underlapped separate servovalves to match control pressures at null (may still have a dynamic force fight problem).

3.2.2.6 Velocity Control of Extending Actuator

PROBLEM: Damage caused by poor velocity control of extending actuator.

ISSUE: Velocity control restrictors must be sized to prevent overpressure and cavitation.

ILLUSTRATION:



SOLUTION:

$$F = \left[P_R + \frac{A_R{}^2V^2C_1{}^2}{C_R{}^2}\right]A_R - \left[P_s - \frac{A_H{}^2V^2C_1{}^2}{C_H{}^2}\right]A_H$$

CHECKS:

Equation not valid if $\frac{A_H{}^2V^2C_1{}^2}{C_H{}^2}$ exceeds P_S

F positive is aiding load

F negative is opposing load

$$\text{Rod end pressure} = \left[P_R + \frac{A_R{}^2V^2C_1{}^2}{C_R{}^2}\right]$$

$$\text{Head end pressure} = \left[P_s - \frac{A_H{}^2V^2C_1{}^2}{C_H{}^2}\right]$$

NOTE: C_R and C_H contain the restrictor sizes and are defined in the TERMS list.

FORCE EQUATION TERMS:

NOTE: Also applicable to the retracting case in 3.2.2.7.

F = Actuator force, lbf (positive F = aiding load, negative F = opposing load)
V = Actuator piston velocity, in/s
A_R = Rod end area, in^2
A_H = Head end area, in^2
C_1 = 0.26
P_R = Return pressure, lbf/in^2
P_S = System pressure, lbf/in^2
$C_R = 29.8 \times C_{DR} d_R^2 / \sqrt{S}$, $in^4/lbf^{0.5}.s^2$
C_{DR} = Coefficient of discharge, rod end restrictor
d_R = Diameter, rod end restrictor, in
$C_H = 29.8 \times C_{DH} d_H^2 / \sqrt{S}$, $in^4/lbf^{0.5}.s^2$
C_{DH} = Coefficient of discharge, head end restrictor
d_H = Diameter, head end restrictor, in
S = Fluid specific gravity

[The coefficients C_R and C_H have the units of area divided by the square root of mass density, identical to the units of the prefix to the standard orifice turbulent flow equation Ed.]

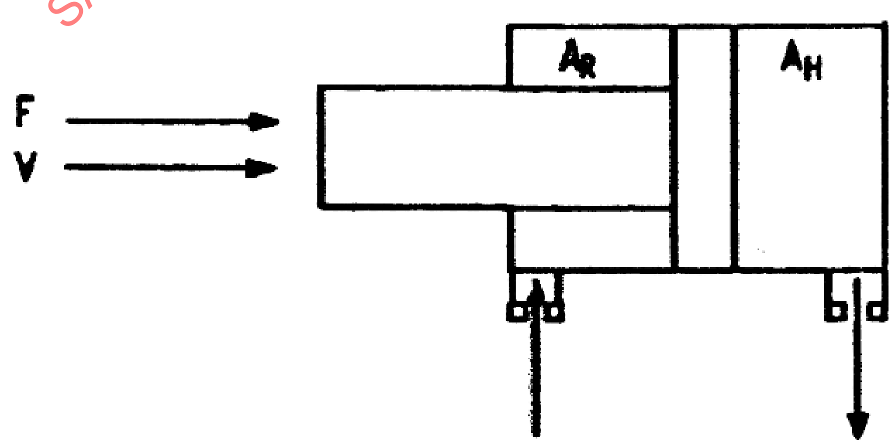
[The quadratic flow equations do not readily allow a single closed-end solution. Given the applied force and required velocity, restrictor diameter sizes can be chosen, in an iterative manner, to meet the performance required while preventing overpressure and cavitation of the rod end and head end cavities Ed.]

3.2.2.7 Velocity Control of Retracting Actuator

PROBLEM: Damage caused by poor velocity control of retracting actuator.

ISSUE: Velocity control restrictors must be sized to prevent overpressure and cavitation.

ILLUSTRATION:



SOLUTION:

$$F = \left[P_R + \frac{A_H{}^2 V^2 C_1{}^2}{C_H{}^2}\right] A_H - \left[P_s - \frac{A_R{}^2 V^2 C_1{}^2}{C_R{}^2}\right] A_R$$

CHECKS:

Equation not valid if $\frac{A_R{}^2 V^2 C_1{}^2}{C_R{}^2}$ exceeds P_S

F positive is aiding load

F negative is opposing load

$$\text{Rod end pressure} = \left[P_S - \frac{A_R{}^2 V^2 C_1{}^2}{C_R{}^2}\right]$$

$$\text{Head end pressure} = \left[P_R + \frac{A_H{}^2 V^2 C_1{}^2}{C_H{}^2}\right]$$

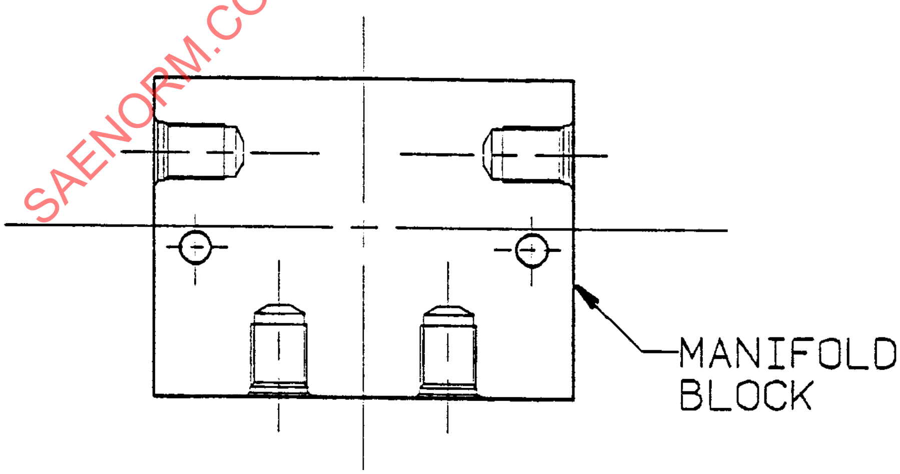
[The quadratic flow equations do not readily allow a single closed-end solution. Given the applied force and required velocity, restrictor diameter sizes can be chosen, in an iterative manner, to meet the performance required while preventing overpressure and cavitation of the rod end and head end cavities Ed.]

3.2.2.8 Aileron Failed to Operate

PROBLEM: Installed aileron actuator failed to operate.

ISSUE: No hydraulic oil was being delivered to the L/H aileron actuator. After extensive troubleshooting, it was found that a manifold block passage had not been drilled.

ILLUSTRATION:



SOLUTION: Although the manifold had passed inspection it was decided to replace it. Afterwards the aileron worked.

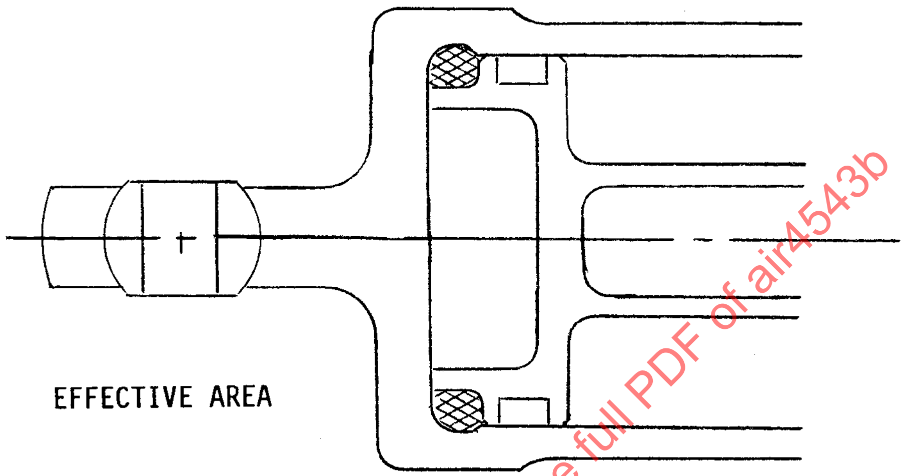
Never take for granted that any part is per the drawing.

3.2.2.9 Fully Retracted Actuator Fails to Extend

PROBLEM: Fully retracted actuator fails to extend against load.

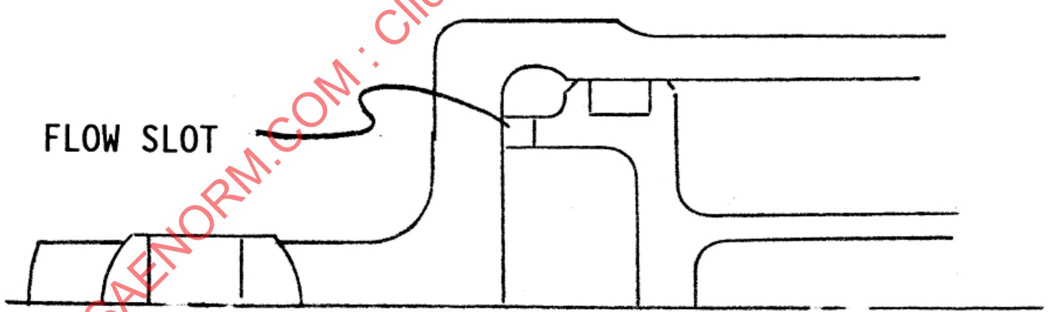
ISSUE: Piston retract stop ring acts as face valve, reducing extend area.

ILLUSTRATION:



SOLUTION: Add flow slot in retract stop ring to assure that full head area is pressurized.

ILLUSTRATION:



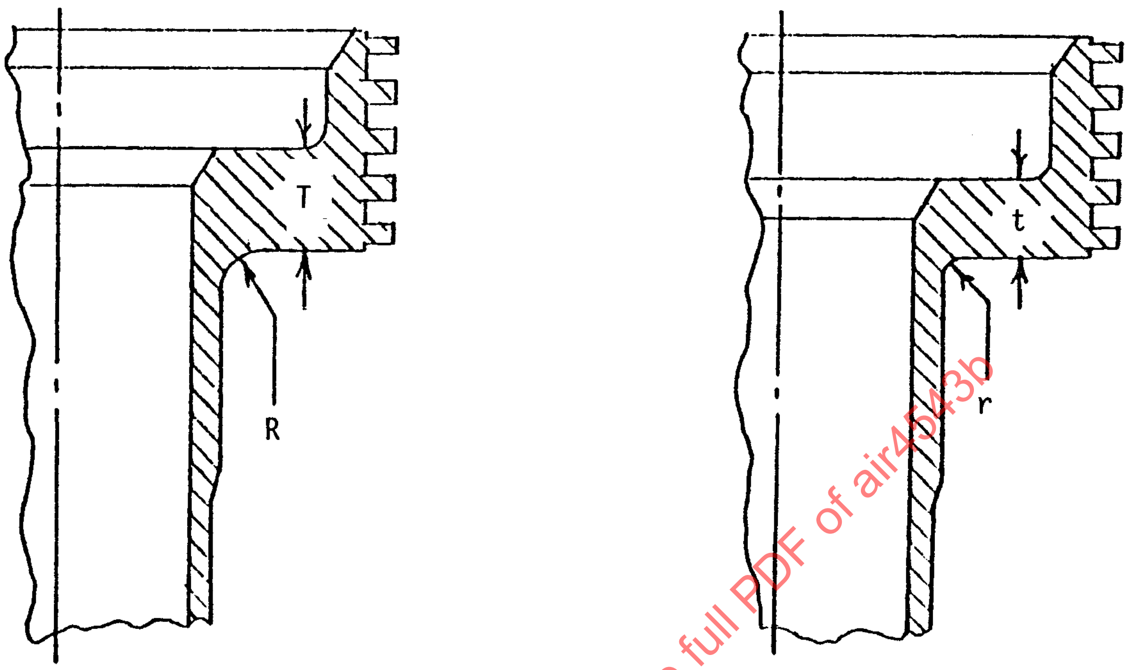
3.2.2.10 Piston Head Separation

PROBLEM: Piston head separation in service.

ISSUE: Hydraulic cylinder piston heads have separated from the piston rods at the filet between the head and rod.

ILLUSTRATION:

DO **DO NOT**



SOLUTION: Design pistons with ample wall thickness (T) and generous filet radii (R). Consideration must be given to peak transient loading. The type loading, whether pure tensile or combined tensile and bending, must be considered in relation to the stress distribution at the piston head filet to prevent stress risers.

3.2.2.11 Fatigue Failure of Internal Threads in Aluminum Components

[See AIR4543/1 for revised version. Ed.]

3.2.2.12 Ball Bearing Failures on Servoactuator Input Shaft

PROBLEM: Bearing balls supporting servo input shaft have come out of bearing races and jammed main valve spool causing stab to lock up.

ISSUE: Support bearing may be axially loaded (forced into position) during assembly of the servovalve, causing balls to come out of bearing races. Since the bearing is used in a backup input control, this failure can go undetected and is not visually observable.

SOLUTION:

a. Split outer race allowing higher ball retention force.

b. Caged bearing design if space permits.

3.2.2.13 Ball Bearing Fatigue Failure on Servoactuator Input Shaft

PROBLEM: Ball bearings, supporting servo input shafts, have had many fatigue failures.

ISSUE: Ball bearings are normally qualified by running them for a given number of rotations under their rated load. Bearings which support a servo input shaft are not used as they were qualified. The bearing oscillates through a few degrees - its entire life spent with the load on 1-3 balls in contact with a small arc on the inner and outer races.

SOLUTION: De-rate the bearing load capacity to account for this unusual usage. Assure that the actuator qualification verifies bearing endurance life.

3.2.2.14 Fly By Wire Actuator Pressure Fatigue Failures

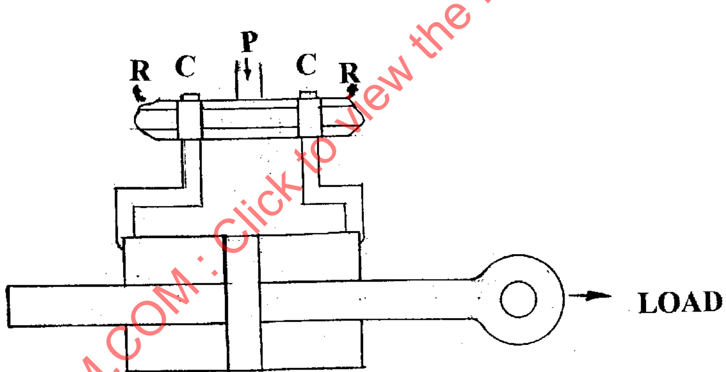
PROBLEM: Early pressure fatigue failures of fly by wire actuators.

ISSUE: Fly by wire servoactuator barrel and end closure pressure fatigue failures occurred in a very short period of use due to a combination of servovalve overlap conditions and continuous small movement command which kept the valve within the overlap region.

To meet low leakage requirements, the supplier provided valves with overlap - long overlap cylinder to return lands and shorter overlap pressure to cylinder lands, tending toward high "no load" null pressures. Combined with an aiding air load, the resisting cylinder pressure may reach a very high pressure value. For example: A constant 1500 lbf/in^2 (10 300 kPa) extend force pressure load could require the resisting cylinder chamber pressure to approach 4500 lbf/in^2 (31 000 kPa) in a 3000 lbf/in^2 (20 700 kPa) system when the valve is in the overlap region and the actuator is extending, depending on the ratio of the overlap lengths.

Combining this condition with small amplitude, low frequency computer generated commands while the aircraft is flying straight and level results in cylinder chamber pressure cycles in the resisting cylinder chamber approaching a 3000 lbf/in^2 (20 700 kPa) to 4500 lbf/in^2 (31 000 kPa) range while retracting and extending respectively. If the commands are continuous, even at low frequencies, a substantial number of high pressure cycles may be experienced in a short time even if there is virtually no load change.

ILLUSTRATION:



SOLUTION:

a. Minimize overlaps but specify overlap balance conditions if a low leakage is required.

b. Insure, via very slow cycle looped cylinder pressure monitoring, that there is a good balance of pressure and return land overlaps if overlaps are substantial, i.e., that the cylinder pressure remains within an approximate (P + R)/2 ± 500 lbf/in^2 (± 3500 kPa) range in the entire overlap region.

c. It is important to know what the FCC is requiring the actuator to do.

d. Minimize the frequency of small movement commands (system).

3.2.2.15 Uncommanded T-2C Flight Control System Inputs

PROBLEM: Uncommanded Inputs to the T-2C longitudinal flight control system.

a. The training command was experiencing flight incidents, in the form of un-commanded pitch inputs, which led to the grounding of the training fleet due to safety-of-flight concerns.

b. A task team consisting of Navy and industry and SAE experts was convened to study the problem.

ISSUE:

a. Longitudinal flight control actuator utilizes "scarf cut" main piston seals. Teflon cap seal with O-ring energizer in wide groove, 1% squeeze

b. Longitudinal flight control actuator utilizes force feedback by means of balance pistons. Free floating type.

SUSPECT CAUSES OF PROBLEM:

a. Main piston seal by-pass results in uncommanded actuator movement through unbalance of feedback pistons

b. Potential sticking of feedback pistons results in un-commanded inputs

SOLUTION:

a. Replace scarf cut main piston seal with uncut seal

b. Increase seal squeeze to prevent by-pass, 10 to 12%

c. Improve feedback piston design to reduce probability of sticking

d. More balancing grooves, 13 versus 4

e. Tighter clearance, 125 µin (3.2 µm)

f. Reduce jamming potential, reduce travel

g. Straightness control

VALIDATION OF SOLUTION:

a. Qualification test of new main piston seal design

b. Qualification test of improved feedback pistons

c. Flight test of reworked actuators with design improvements

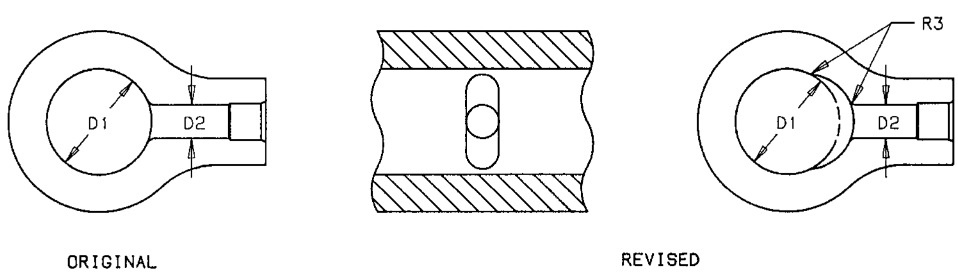
3.2.2.16 Intersecting Passages and Fatigue Life

PROBLEM: Low fatigue life at passage intersections.

ISSUE: Stress concentration decreases as intersecting hole diameter approaches bore diameter.

SOLUTION: Use "fly cut" to create effectively a larger diameter hole.

ILLUSTRATION:



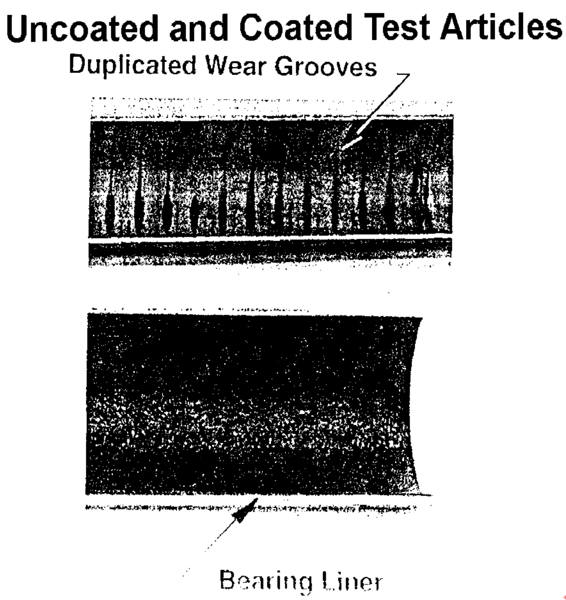
3.2.3 Mechanical Actuation

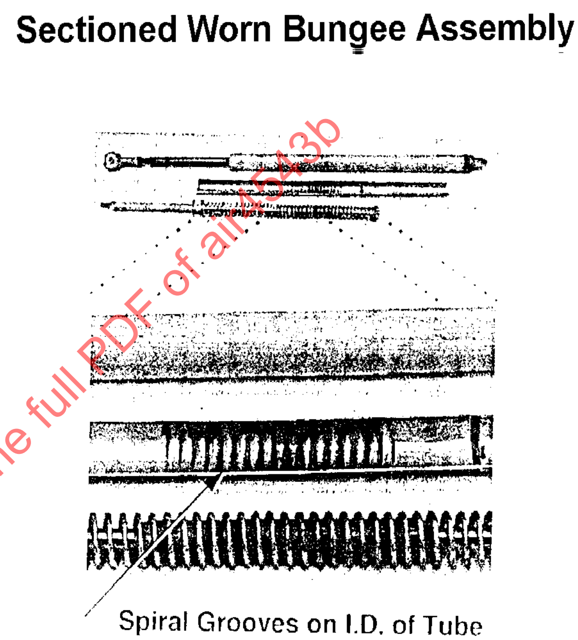
3.2.3.1 Long-Term Wear of Flight Control Bungee Assemblies

PROBLEM: Springs wearing through bungee tubes.

ISSUE: Flight control bungees sustaining severe wear of bungee tubes caused by vibration of internal springs.

ILLUSTRATION:





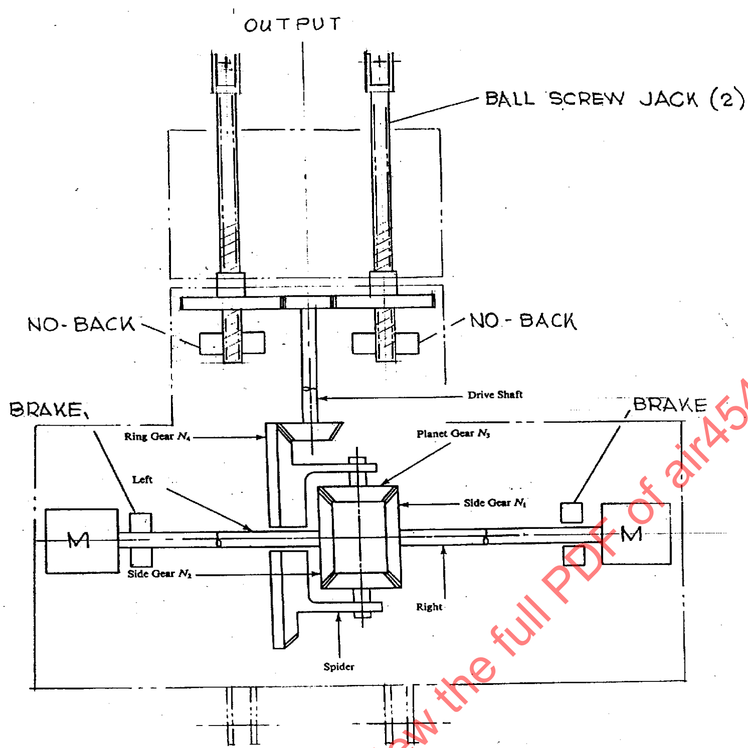
SOLUTION: Apply homogeneous bearing liner to ID of tubes to provide a wear resistant coating.

3.2.3.2 Latent Failures in the Design of Flight Control Components

PROBLEM: Latent failures in the design of flight control components.

ISSUE: Redundancy is frequently employed in the attempt to meet safety and reliability requirements of flight control systems or components. In order to meet the safety requirements, latent failures are sometimes inadvertently incorporated into the design. The subsequent reliability analysis may not properly take into account the actual failure rate of the system or component. In the pitch trim actuator the failure of the mechanical no-backs was undetected because the electro-magnetic brakes at the motors prevented the output from being back driven even after failure of the no-backs. A subsequent failure (open) in the gear train caused the output to be back-driven (the ball screws are reversible) by the air loads at a high rate with no restraint except the inertia of the components.

ILLUSTRATION:



SOLUTION: Carefully examine the component or system for latent failures. If found, the best solution is simply to alter the design and eliminate the latent failure mode. The next best approach is to properly take into account the exposure of the latent failure mode in determining the actual system failure rate. Hopefully the resultant net failure rate is acceptable for the application being considered. If not, a redesign is in order. Alternately, a reduction in the exposure time of the latent mode may be an option.

3.3 Mechanical and Hydromechanical Component Lessons Learned

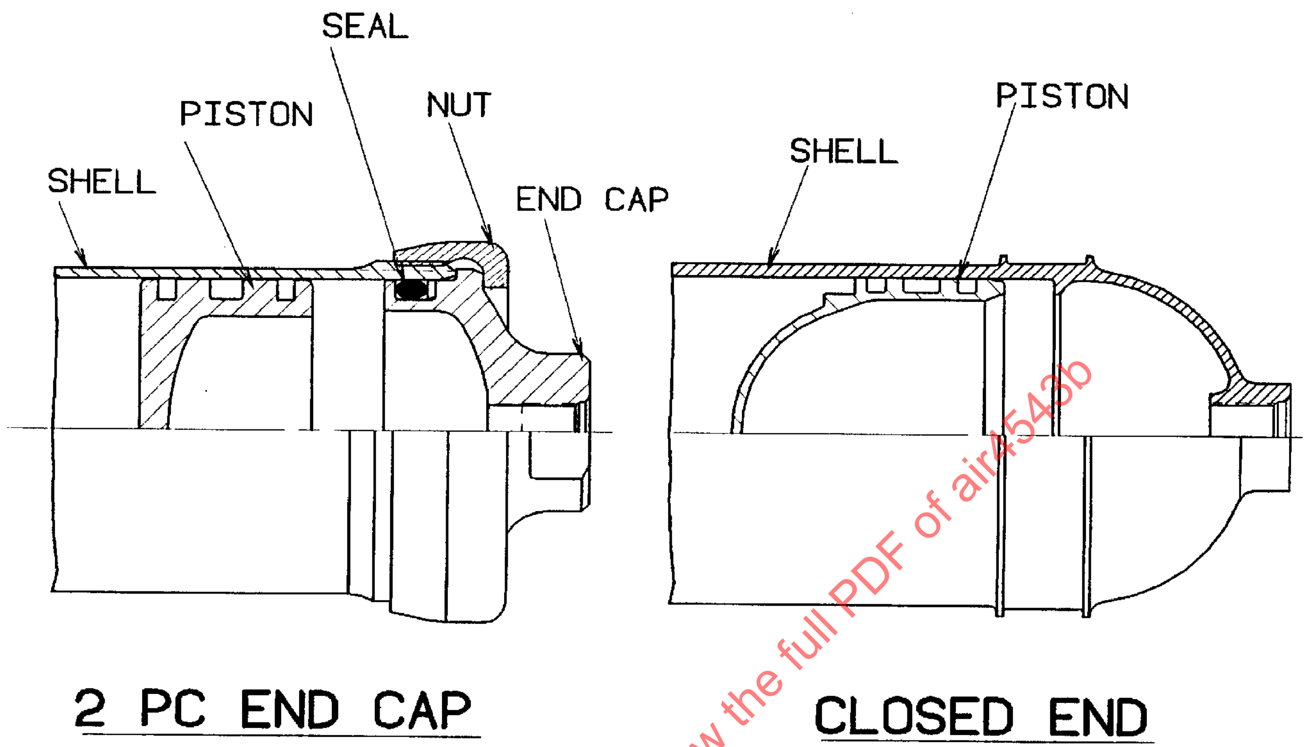
3.3.1 Accumulators

3.3.1.1 Excessive or Premature Leakage Across Accumulator Seals

PROBLEM: Excessive or premature leakage across accumulator seals.

ISSUE: Gas leaking into the system can cause a loss of aircraft or system.

ILLUSTRATION:



SOLUTION:

a. Ensure that the chrome plate is controlled to minimize cracking (cracks inherent in chrome plating process).

b. Ensure that the seal compound meets all temperature, pressure, and pressure discharge requirements. Minimize the compression set and explosive decompression.

c. Use the closed end cap on gas side (eliminate static seal).

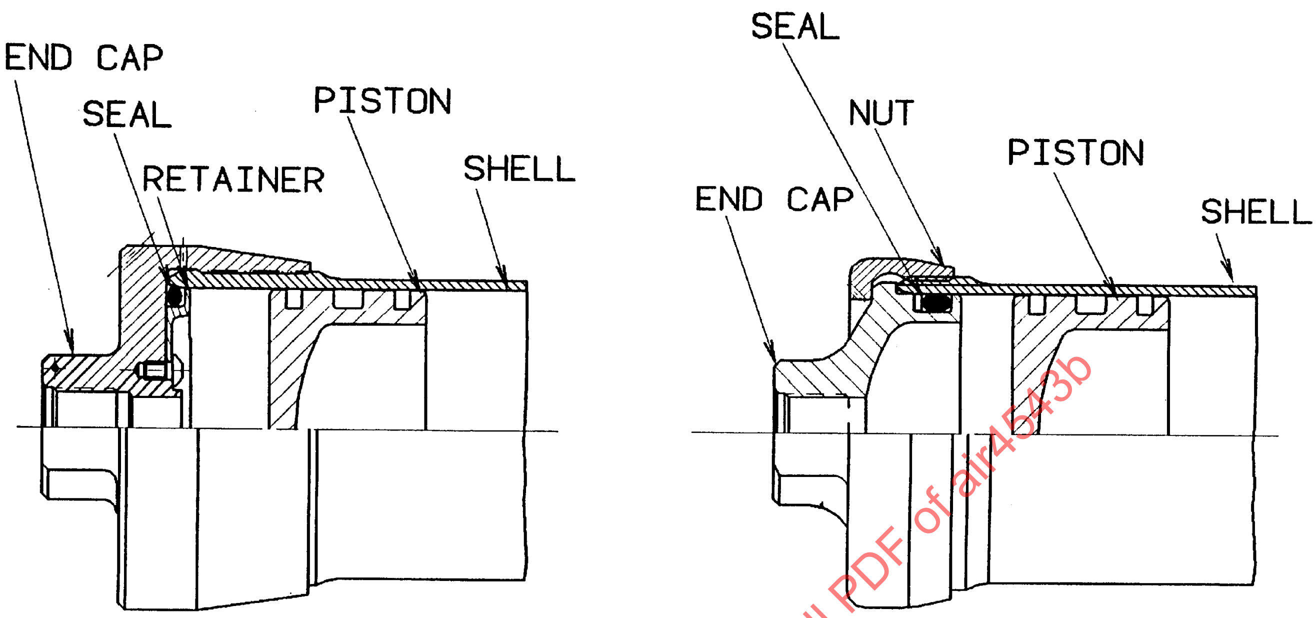
d. Improve the seal concept to improve sealing capability (multiple seals, metal bellows, etc.).

3.3.1.2 Premature Failure of Accumulator End Caps

PROBLEM: Premature failure of accumulator end caps.

ISSUE: Failed accumulators are an extreme hazard to aircraft.

ILLUSTRATION:



1 PC END CAP

2 PC END CAP

SOLUTION:

a. Replace aluminum with steel or equivalent.

b. Ensure that equipment is used at design pressure.

c. Use two piece end cap design to distribute stresses (steel nut and aluminum gland).

d. Use the closed end cap on gas side.

e. Use materials with good fracture toughness - slow crack propagation.

f. Provide adequate corrosion protection.

3.3.1.3 False Accumulator Gage Reading

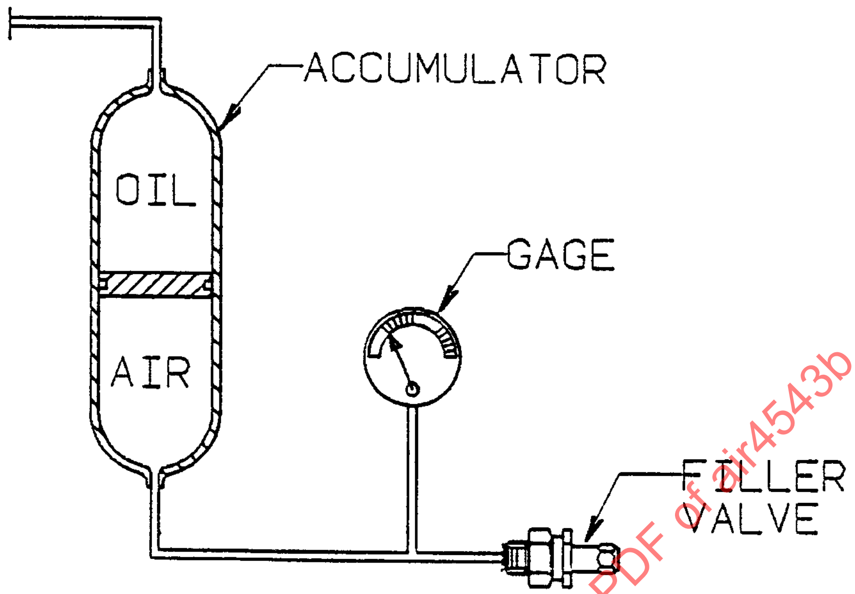
PROBLEM: False accumulator gage reading.

ISSUE:

a. The line and filler valve volume on the air side of the accumulator were not correctly included in the air side calculations.

b. The fill valve had been relocated after initial sizing. The accumulator piston bottomed before reaching 3000 lbf/in^2 (20 700 kPa).

ILLUSTRATION:



SOLUTION: The accumulator pre-charge was recalculated and adjusted to the correct value.

3.3.1.4 Hydraulic Accumulator Safety

PROBLEM: Failure of hydraulic accumulator end caps in service.

ISSUE: End cap failure causing aircraft structural and system damage.

BACKGROUND:

From the late 60's sporadic failures of accumulator end caps reported on both commercial and military aircraft. Up to this time only slight structural damage occurred.

a. In the mid 70's increase in failure rate as aircraft fleet aged. One incident causing major structural and system damage to large military transport.

b. Conducted Investigation and evaluation by combined commercial and military panel at A-6 to determine cause and corrective action. Evaluate all accumulator installations for service environment and safety.

c. Failed accumulators were standard MIL-DTL-5498 type.

REVIEW:

Accumulators are generally installed on aircraft and not removed for service or overhaul.

a. Accumulator life is affected by the installation environment. Under certain environmental conditions the surface protection treatments can deteriorate and allow corrosion and pitting which can lead to early stress corrosion type failures.

b. Since the accumulator is charged with gas the results of a failure can be extremely violent.

c. Inadvertent use of moist nitrogen or air to service accumulator can cause loss of internal corrosion protection system.

d. Location of accumulator in structure could cause damage and failure to adjacent systems and/or personnel.

SOLUTION:

a. A minimum in-service inspection program is recommended.

– AIR4150 (now ARP4150) procedure for in-service inspection written.

b. On existing aircraft the installation to be evaluated for:

– Effect of environment on accumulator

– Safety of surrounding structure, services, and personnel

c. Improve design for future procurement:

– MIL-A-5498 rewritten

– New ARP's written to cover different types of accumulator:
ARP4378 Cylindrical (Bellows)
ARP4379 Cylindrical (Piston)
ARP4553 Self-Displacing

3.3.2 Filtration, Filters, and Fluids

3.3.2.1 MIL-F-8815 Aircraft Filters Deficient in Marine Applications

PROBLEM: MIL-F-8815 aircraft filters perform unsatisfactorily in ship and submarine applications.

ISSUE:

a. MIL-F-8815 filter pop-up indicators actuate under cold start conditions and the elements often have an unsatisfactory life.

b. MIL-F-8815 differential pressure indicators are equipped with thermal lockouts to prevent actuation for temperatures below 100 °F ± 15 °F (38 °C ± 8.5 °C). The contractor had changed the required thermal lockout temperature to 30 °F ± 20 °F (-1 °C ± 11 °C) rendering the thermal lockout useless since ambient temperatures are almost always above 50 °F (10 °C).

c. The viscosity of MIL-PRF-5606 aircraft hydraulic fluid varies from about 17 cst (17 x 10^{-6} m^2/s) at 85 °F (29 ˚C) to 14 cst (14 x 10^{-6} m^2/s)at 115 °F (46 ˚C), a change of 20%. Submarine hydraulic fluid 2190-TEP (MIL-PRF-17331) varies from about 150 cst (150 x 10^{-6} m^2/s) at 85 °F (29 ˚C) to 60 cst (60 x 10^{-6} m^2/s) at 115 °F (46 ˚C), a change of 250%. At normal operating temperatures the viscosity of MIL-PRF-5606 (Inactive for new design) may increase 30% at 3000 lbf/in^2 (20 700 kPa) whereas the viscosity of the 2190-TEP under the same conditions will increase 65%. The large changes in fluid viscosity over normal operating temperatures often makes the use of pop-up indicators impractical when using 2190-TEP fluid.

SOLUTION:

a. The standard thermal lockouts for pop-up indicators should be retained when using MIL-F-8815 filter assemblies.

b. Depending upon the fluid and operating temperature, the flow rating of MIL-F-8815 filter assemblies and elements must be significantly down-rated when using more viscous fluids. It is not uncommon to down-rate flow capacity by a factor of 10 or more.

c. In many applications it is better to use the larger MIL-F-24402 filter housings and elements. These housings can be equipped with gage-type differential indicators for applications not satisfactory for pop-up indicators. The larger size elements have longer life, reduce the number of spare elements that must be carried, and have lower operating and maintenance costs.

3.3.2.2 Filter Bypass Contaminates Servovalves

PROBLEM: Contamination due to filter bypass reliefs lifting results in costly maintenance of servovalves.

ISSUE: On submarines the pilot stages of steering and diving system electro hydraulic servovalves were protected with MIL-F-8815 filter assemblies with bypass reliefs. On numerous occasions, filter element replacement was not accomplished in a timely manner and the reliefs lifted allowing contaminants to enter the servovalve. This resulted in a deterioration of servovalve performance and required disassembly and cleaning of the servovalve to restore the valve to service.

SOLUTION: Plug the bypass relief valves and use filter assemblies without bypass reliefs in new applications. If maintenance is neglected, the increased pressure drop across the filter increases and servovalve performance slowly deteriorates. The change in servovalve performance is no worse than when contaminated fluid enters the valve and corrective action is much easier as only the filter element needs to be replaced.

3.3.2.3 Failure of MIL-F-8815/2-8 Filter Housing

PROBLEM: Failure of MIL-F-8815/2-8 (inactive for new design) filter housing.

ISSUE: Failure due to compression ignition explosion.

Cause of Failure:

a. Air left in housing when installing replacement element.

b. Rapid compression of air when shifting to emergency mode.

Contributory Causes:

a. The filter element was an old style in which media was limited to approximately one-half the length of the element. Air could not flow through the upper portion of the element, which was a solid tube, and remained trapped in the housing.

b. The filter differential pressure indicator is equipped with thermal lockout, which may have prevented the indicator from actuating even though the element was loaded. This could have contributed to the rapid compression of the air.

SOLUTION:

a. Fill filter bowls with fluid when changing elements.

b. Do not use elements in which media does not extend the entire length of the element.

c. Provide vent fittings on components to facilitate venting of air after maintenance. Vent air before rapidly pressurizing.

d. Carefully review designs for air traps in portions of the system subject to rapid pressurization.

e. Try to avoid designs in which low pressure regions are subject to rapid pressurization.

3.3.2.4 Inadequate Component Life at Elevated Temperatures

PROBLEM: Inadequate pump life at elevated temperatures.

ISSUE:

a. When a hydraulic pump was run in preservative fluid, MIL-PRF-46170, at 230 °F (110 °C) severe corrosion of bronze and copper parts occurred. Pump would not operate after 200 h of endurance test time.

b. Fluid tests indicated that above 160 °F (71 °C), the barium dinonaphtalene sulfonate corrosion inhibitor breaks down to form strong sulfonic acids which attack bronze in the pump. Acid number of fluid increased.

c. Corrosion inhibited hydraulic fluids were developed for recoil mechanisms of army guns where temperatures are low, less than 160 °F (71 °C). Also they are used successfully in tank turrets at low operating temperatures, less than 160 °F (71 °C).

SOLUTION:

a. Do not use preservative fluids containing barium dinonaphtalene sulfonate as operational fluids above 160 °F (71 °C).

b. Consider banning the use of MIL-PRF-46170 as a shipping fluid for components as clogging of filter may result in aircraft systems.

3.3.2.5 Ice Formation in Hydraulic Fluid

PROBLEM: Brake grabbed after aircraft landed at Northern Tier base after flight in sub-zero temperatures.

ISSUE: T-38 landed at Cold Lake Canada in mid-winter after flight from Washington. While taxiing down runway, one brake grabbed causing aircraft to immediately turn into snow bank.

Was originally diagnosed as caused by conversion of aircraft to MIL-PRF-83282 and malfunction of brake attributed to high viscosity of MIL-PRF-83282 at low temperatures.

SOLUTION: Investigation of hydraulic system revealed that the hydraulic fluid was MIL-PRF-5606; aircraft had not been converted to MIL-PRF-83282 yet.

Further investigation found that grabbing brake was due to formation of an "ice block" in the brake lines preventing proper operation of the brake system. The water content of the MIL-PRF-5606 hydraulic fluid was in excess of the solubility limit (>400 ppm).

High water content in the hydraulic fluid (in excess of solubility limit) can result in formation of ice crystals or blocks in the hydraulic lines or components causing malfunctions.

3.3.2.6 Fluid Additives and Stuck Servovalves

PROBLEM: Servovalves were sticking in UH-1 helicopter hydraulic systems.

[From the text, the sticking subcomponents were poppet-type check valves rather than the electrohydraulic, or mechanically driven proportional-type valves normally called “servovalves” by the industry Ed.]

ISSUE: The Army was experiencing frequent sticking of servovalves in helicopters stationed at Fort Rucker.

a. Resulted in grounding the entire fleet of helicopters stationed at Fort Rucker - Training base for Army helicopter pilots.

b. Meeting of pilots, maintenance and program office personnel, Army and Air Force scientists and contractor personnel convened to discuss problem.

c. Training mission included hydraulics-off maneuvers.

d. Barium dinonylnaphthalene sulfonate (rust inhibitor) found in operational hydraulic fluid in helicopter.

CAUSE: During hydraulics-off operation of helicopter, the servovalve opens and closes rapidly in concert with the rotation of the helicopter blades which causes degradation of the less stable rust inhibitor contaminant in the hydraulic fluid resulting in the formation of a mild adhesive like compound which caused the poppet in the servovalve to stick to the seat.

SOLUTION:

a. Reduce barium content in helicopter fluid to <15 ppm.

b. Remove and clean stuck valves from helicopters and re-install.

c. Assure that rust inhibited hydraulic fluid (MIL-PRF-6083) is thoroughly drained from hydraulic components before they are installed on helicopters.

VALIDATION:

a. No stuck valves for 1st year after implementation of above steps.

b. Similar deposit found in laboratory tests involving dithering cycle of poppet on seat with contaminated hydraulic fluid.

3.3.2.7 Hidden Zero Pressure Volume in Filter

PROBLEM: Blow-out failure of hydraulic flter element base cap at 3000 lbf/in^2 (20 700 kPa).

ISSUE:

a. A slug of bottom cap was blown out of end cap and could have drifted downstream into system.

b. Failure caused by unknown volume trap of zero pressure which created unexpected differential pressure on end cap of 3000 lbf/in^2 (20 700 kPa).

c. Failure was on second source element and was not detected during qualification testing.

d. Element bottomed and aligned on dome in base of bowl.

SOLUTION:

a. Immediate fix for filters in the supply system was to machine relief groove into base cap to eliminate zero pressure trap.

b. Final fix was to require a filter element end cap that would withstand 4500 lbf/in^2 (31 000 kPa).

3.3.2.8 Case Drain Filter Replacement Interval

PROBLEM: Main hydraulic pumps on fighter aircraft were experiencing in-flight failures and fires at an unsatisfactory rate.

ISSUE: High back pressure on pumps caused by clogged case drain filters was suspected of initiating pump failure. Filter change was on-condition of red delta pressure button.

SOLUTION:

a. Evaluate filter clogging at 200 and 400 hours.

b. 200 hour replacement recommended from data.

c. C.D. filter replacement reduced pump failures and improved mean time between failures from 950 hours to 2000 hours.

3.3.2.9 Hydraulic Fluid Oxidation

PROBLEM: Oil oxidation is significantly accelerated by high temperature and the combined effects of water and metal contaminants.

ISSUE: A rough approximation is that the oxidation rate will double for every 10 °C rise in oil temperature. As the table below indicates, oil oxidation rate increases in the presence of water, and the combination of water and metal surfaces, such as is generated by fresh wear debris, significantly increase it. Small metal particles act as catalysts to rapidly increase the total acid number. Additionally, air entrapped within the oil with further accelerate oil oxidation.

OXIDATION RATES

Run	Catalyst	Water	Hours	Total[1] Acid Number Change
1	None	No	3500+	0.00
2	None	Yes	3500+	+0.73
3	Iron	No	3500+	+0.48
4	Iron	Yes	400	+7.93
5	Copper	No	3000	+0.72
6	Copper	Yes	100	+11.03

[1]Total acid number increases which exceed 0.5, indicate significant fluid deterioration.
Reference: Weinschelbaum, M., proceedings of the National Conference on Fluid Power, VXXIII-269.

SOLUTION:

a. Operate system at lowest temperatures possible and avoid areas of excessive heating.

b. Design system to reduce possibility of ingression of water and contaminants (use closed reservoir or use reservoir vent filter/dryer).

c. Use a fine "green run" filter and a high efficiency (MIL-F-81836) (canceled with no known replacement) GSE filter to remove built-in contaminants.

d. Use fine system filters (i.e., pressure, return, case drain, etc.) to remove fresh metal wear particles and preclude them from traveling to hot zones where oxidation occurs rapidly.

e. Use a fluid purifier to reduce water concentration and solid contaminants in the oil.

f. Use in-system air eliminator or automatic air bleed valve.

3.3.2.10 By-Pass Type Pressure Filter and “Delta P” Button Surveillance

PROBLEM: A trainer aircraft was experiencing uncommanded pitch. Review of filter design revealed a by-pass type 5 µm pressure line filter and no pre/post flight indicator button monitoring. A secondary filter (25 µm) is at the inlet of flight control actuators.

ISSUE: Without daily monitoring of Delta "P" buttons, a pressure filter could by-pass and allow contamination to flight control actuators. Contamination (25 µm) was suspected of contributing to un-commanded pitch problem.

SOLUTION:

a. Require pre/post flight monitoring of filter Delta "P" buttons.

b. Change pressure filter to non by-pass type as specified in AS5440 (previously MIL-H-5440).

c. Do not allow painting over buttons during rework.

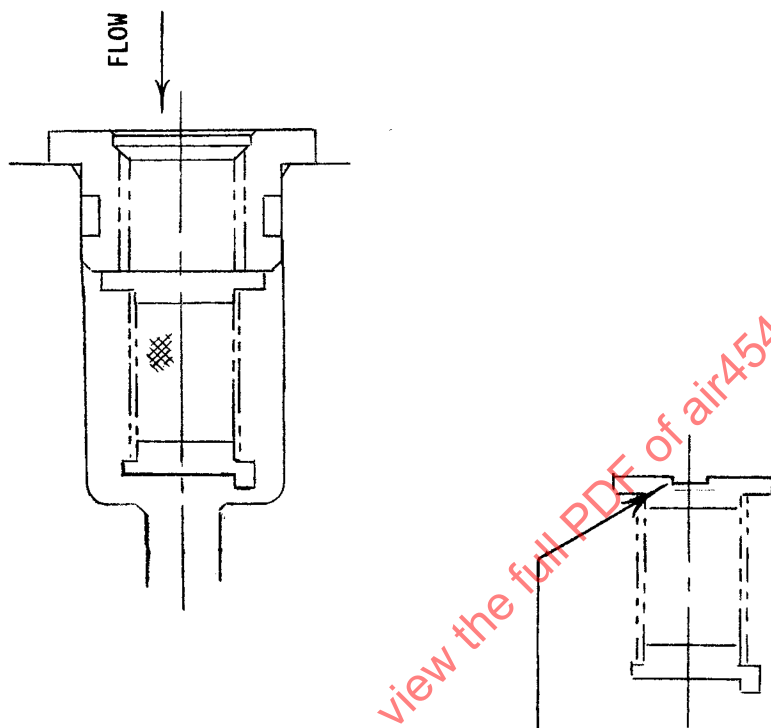
d. Periodically check operation of Delta "P" indicators.

3.3.2.11 Manifold Does Not Depressurize

PROBLEM: Manifold does not depressurize.

ISSUE: Inlet screen housing acts as a check valve preventing the module from depressurizing after the system's isolation valve is actuated.

ILLUSTRATION:



SOLUTION: Provide a relief slot on the face of the filter housing.

3.3.2.12 Valve Erosion by Contaminated Phosphate Ester Fluids

PROBLEM: Valve erosion caused by chlorine contamination of Phosphate Ester aircraft hydraulic fluids.

ISSUE: Contamination of pre-Type IV phosphate ester aircraft hydraulic fluids by chlorinated solvents was shown to significantly increase electrochemical erosion rates in servo control valves.

[“Type” definitions per AS1241 Ed.]

SOLUTION: Use Type IV phosphate ester hydraulic fluids, and limit the chlorine level in hydraulic fluid to 200 ppm; or avoid the use of chlorinated solvents.

NOTE: Type IV aviation hydraulic fluids for commercial aircraft have been available since 1974. There are no Type I, II, or III fluids now manufactured or used throughout the world.

There have been no reports of electrochemical erosion with Type IV fluids as long as fluid properties have remained within recommended limits. Some cases of electrochemical erosion have occurred when some fluid properties were allowed to fall outside recommended limits. There is no evidence of the effect of chlorine contamination on electrochemical erosion with Type IV fluids.

The mechanism of electrochemical erosion, and chlorine's role in such, has been studied extensively but is still not fully understood. If the fluid operating environment increases in severity, such as higher temperatures, the effect of chlorine contamination on electrochemical erosion could again become significant.

Other problems believed associated with chlorine contamination of commercial aviation hydraulic fluids, such as seal deterioration or metal corrosion, have not been sufficiently documented to warrant an SAE document.

ILLUSTRATION 1: Erosion of Servovalve with Types I, II, III and IV Fluid with 1000 ppm Chlorine Added (Stabilized Rate of Erosion Occurring During Test A)

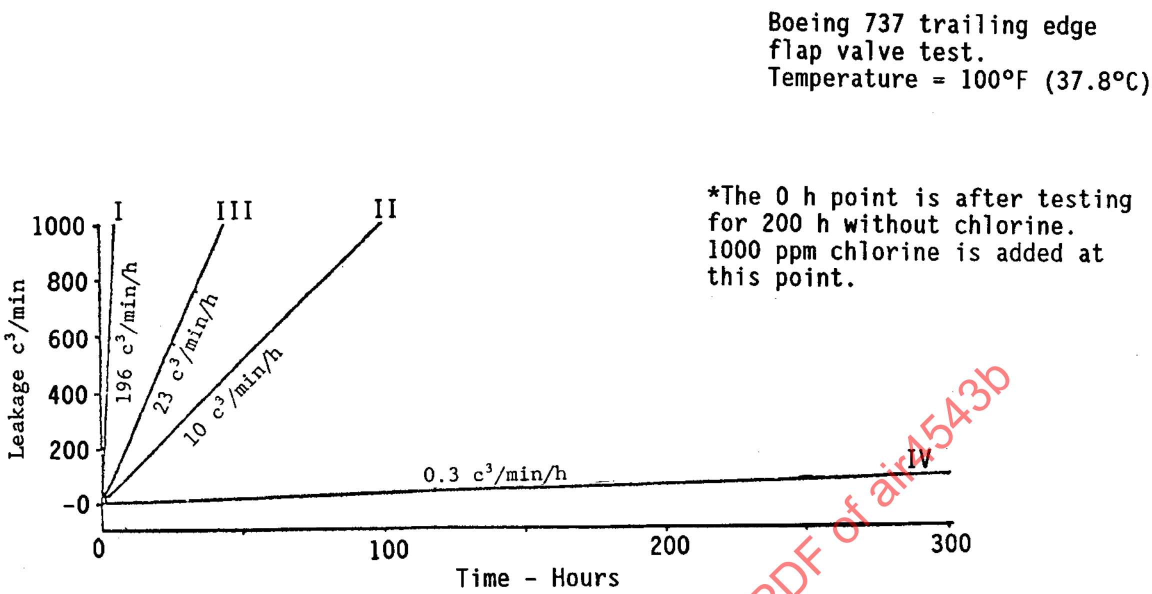
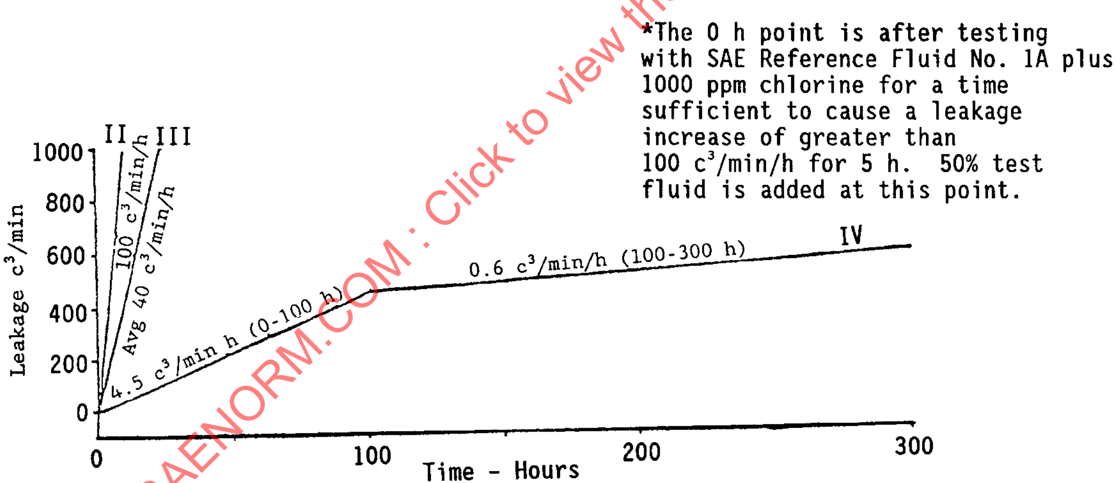


ILLUSTRATION 2: Erosion of Servovalve with Types I, II, III and IV Fluid with 1000 ppm Chlorine Added (Stabilized Rate Following Addition of Type IV Fluid During Test B)



3.3.2.13 Clogging of Component Built-in Filters

PROBLEM: Clogging of component built-in filters.

ISSUE: Components in the hydraulic system that use filters to protect small orifices or critical continuous fluid flow functions can get clogged and can cause control problems.

SOLUTION:

a. The built-in filter element should withstand full system pressure without failure.

b. On servovalves use central filtration to avoid differential flows as filter delta pressure increases.

c. Component filter micron rating to be larger than system filters. Recommended micron rating is 50 to 100 µm.

d. Filters should be of the finger-type (cone shape) which has higher dirt capacity. Avoid flat disc-type.

e. They should be adequately retained and trapped to avoid blowout. Do not rely on pressed fit interference.

f. All restrictors with 0.070 in (1.8 mm) dia or smaller in components should incorporate built-in filters.

3.3.3 Flight Critical Joint Retention

Subsection provided for future use. No Lessons Learned yet submitted.

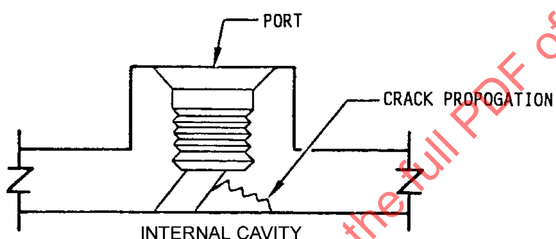
3.3.4 Manifolds

3.3.4.1 Aluminum Manifold Fatigue

PROBLEM: Aluminum manifold fatigue.

ISSUE: Early fatigue failures due to notch effect of hole intersections.

ILLUSTRATION:



SOLUTION:

a. Keep hole intersection angles as near 90° possible (Illustration 1).

b. Break edges at hole intersections (Illustration 1).

c. Add a localized compressive stress layer if feasible (various methods) (Illustration 2).

d. Design to avoid - remove discontinuities where possible (Illustration 3).

ILLUSTRATIONS:

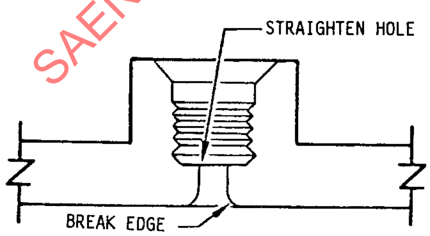


ILLUSTRATION 1

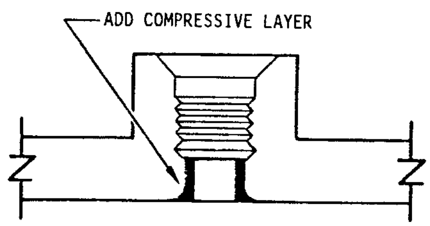


ILLUSTRATION 2

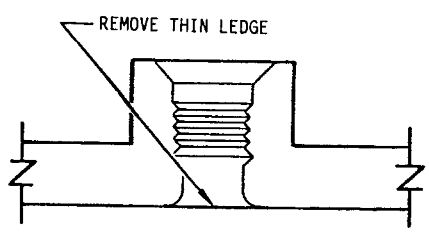


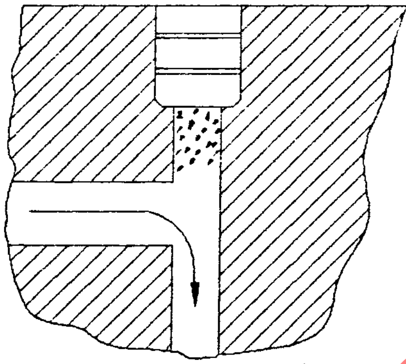
ILLUSTRATION 3

3.3.4.2 Entrapped Contamination

PROBLEM: Entrapped contamination

ISSUE: Dead end passages collect the contaminant.

ILLUSTRATION:

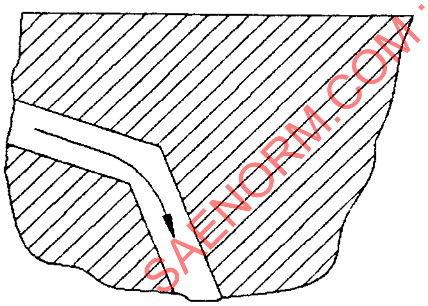


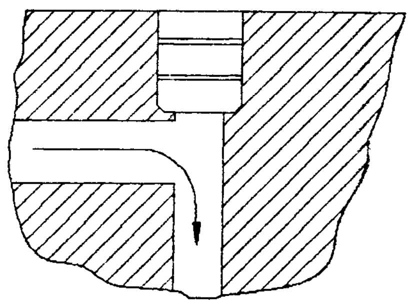
SOLUTION:

a. Reroute the flow passage to eliminate potential traps where flow will flush the contaminant through the valve.

b. Reduce the trap size.

ILLUSTRATION:



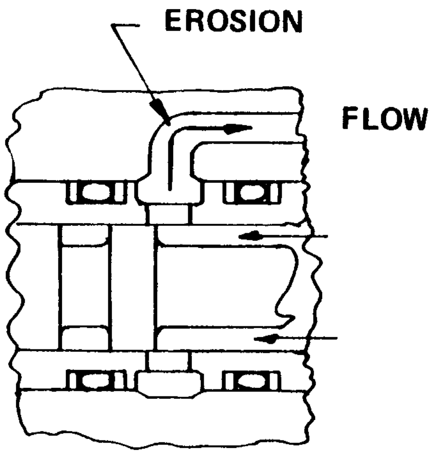


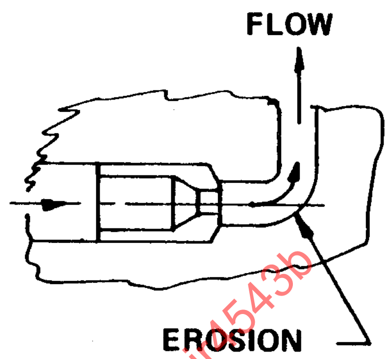
3.3.4.3 Erosion of Aluminum Valve Bodies Due to Jet Action of Restrictors

PROBLEM: Erosion of aluminum valve bodies due to jet action of restrictors.

ISSUE: High pressure impingement or deflected flow of fluid can erode through aluminum valve manifolds.

ILLUSTRATION:





SOLUTION:

a. Reduce restrictor exhaust velocity by using multiple series orifices.

b. Move restrictor exhaust jet away from the manifold wall so that fluid can expand and reduce velocity prior to impingement on the wall.

c. Reduce restrictor exhaust jet energy concentration by using multiple smaller parallel orifices.

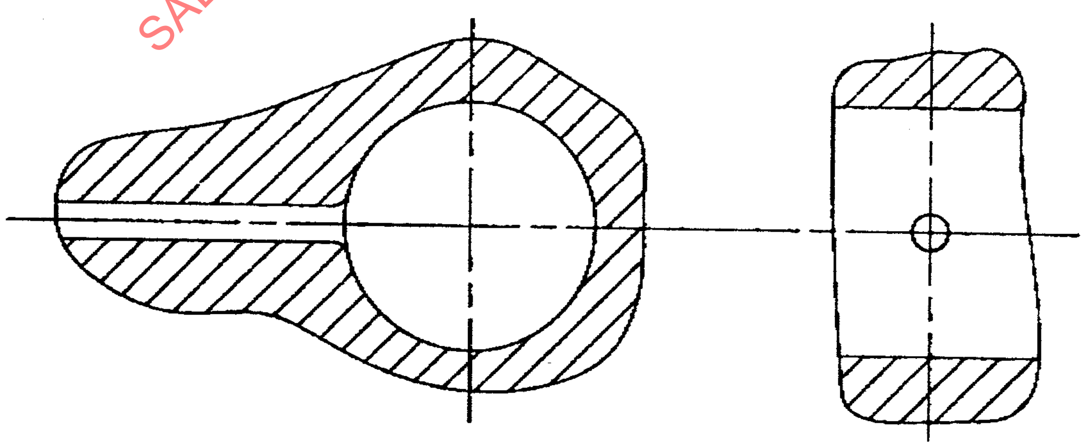
d. Incorporate the steel deflector or jet diffuser downstream of the restrictor.

3.3.4.4 Intersecting Bores Fatigue Failure

PROBLEM: Intersecting bores fatigue failure.

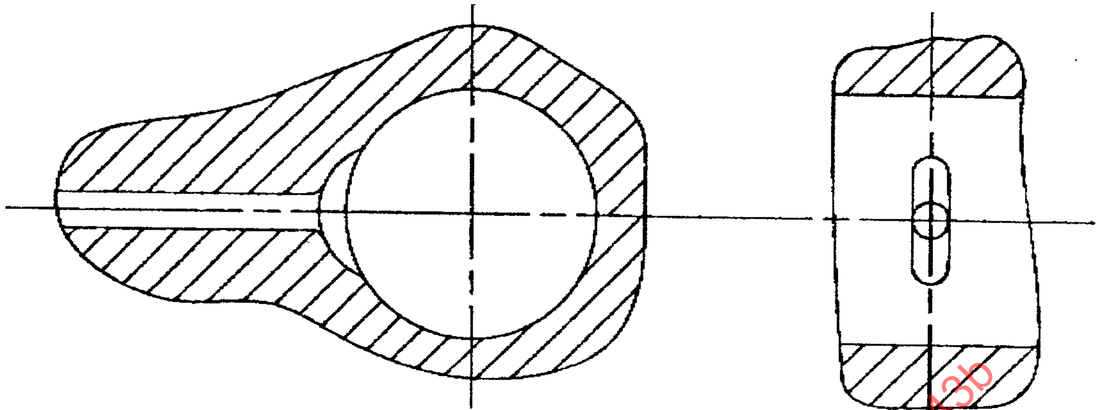
ISSUE: Small bores intersecting large bores increase stress, K = 2.5.

ILLUSTRATION:



SOLUTION: Simulate an elliptical intersection, K = 1.5. Put radius of .030 to .040 in (0.7 to 1.0 mm) on edges.

ILLUSTRATION:

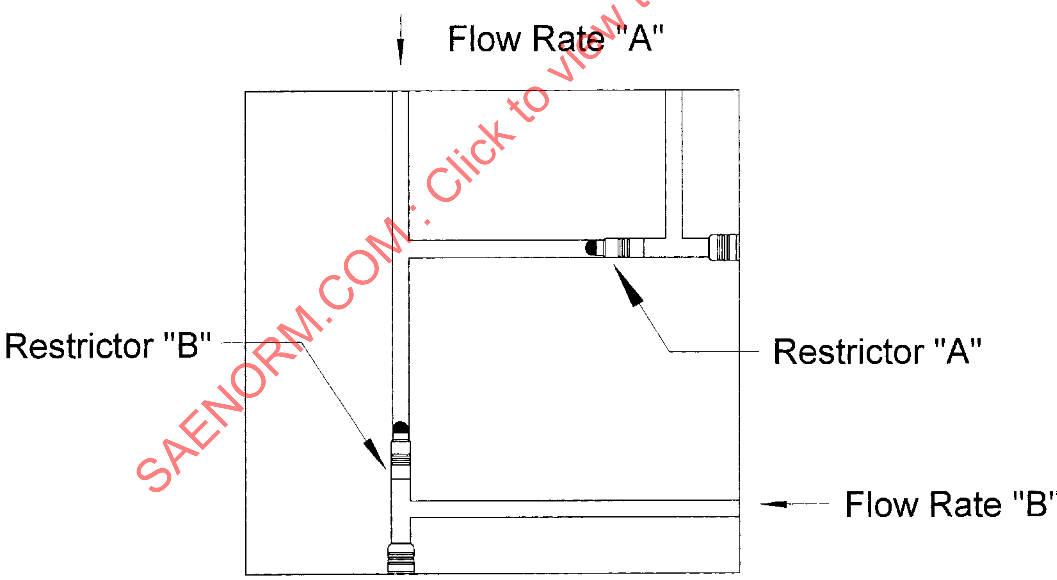


3.3.4.5 Flow Restrictors

PROBLEM: Various size flow orifices may be used in the same component and may be misinstalled.

ISSUE: When different size flow orifices are used in the same component to regulate various flows, provisions must be made to assure installation in the proper flow circuit.

ILLUSTRATION:



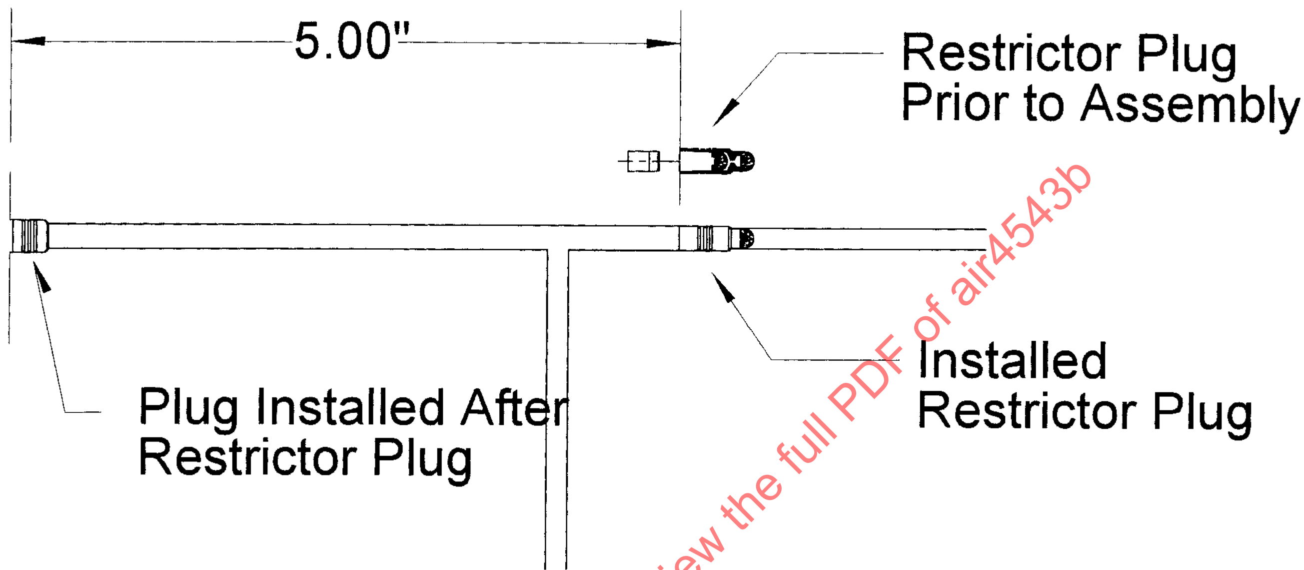
SOLUTION: When two or more different flow restrictors are used in a component, use different ODs to prevent installation of improper restrictor in flow circuits.

3.3.4.6 Small Diameter Orifices Location

PROBLEM: Small diameter orifices can be a challenge to remove if removal becomes necessary.

ISSUE: Small diameter orifices that are installed deep within components are extremely difficult to remove if removal is necessary.

ILLUSTRATION:



NOTE: Illustration dimension of 5.00 in is equivalent to 127 mm.

Small Diameter Orifices That Are Deeply Recessed Are Extremely Difficult To Remove

SOLUTION: When using small orifices consider the removal implications of such orifices and locate orifices where removal is feasible.

3.3.5 Pumps and Motors

3.3.5.1 Pump Cavitation/Erosion

PROBLEM: Pump cavitation/erosion.

ISSUE:

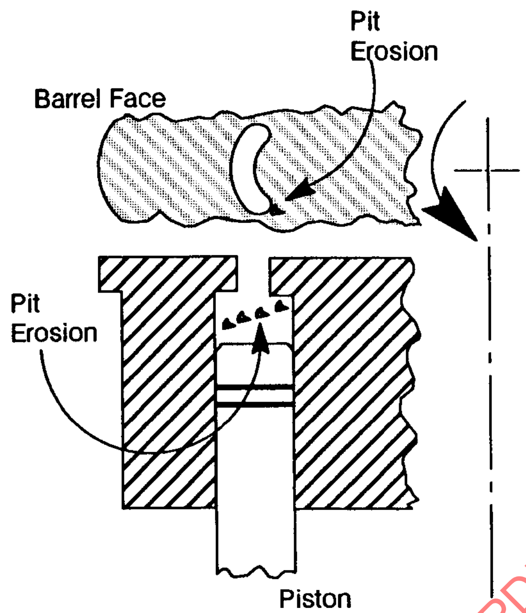
a. Implosion/erosion in pumps

b. Low output flow

c. Noise

d. Vibration/excessive wear

ILLUSTRATION:



SOLUTION:

a. Dynamic analysis of inlet filling component

b. Streamline inlet filling (lines)

c. Size inlet lines to match pump requirements (pressure)

d. Contour filling of pump inlet coverage

e. Increase inlet pressure

3.3.5.2 Specification Performance Requirements at Extended Conditions

PROBLEM: Specification performance requirements at extended (low and high speed/pressure) conditions.

ISSUE: Pump/motor performance at rated conditions is compromised to meet extended conditions' requirements.

SOLUTION: Constrain specification requirements to operationally realistic values.

3.3.5.3 Catastrophic Pump/Motor Failure at Start-up

PROBLEM: Catastrophic pump/motor failure at start-up.

ISSUE: Unlubricated parts at start-up due to fluid having drained out of the pump suction line.

SOLUTION: Fill check.

3.3.5.4 Short Pump Life and Inlet Pressure

PROBLEM: Short pump life due to low inlet pressure.

ISSUE: Low transient inlet pressure - fluid acceleration in long inlet line/fast pump response.

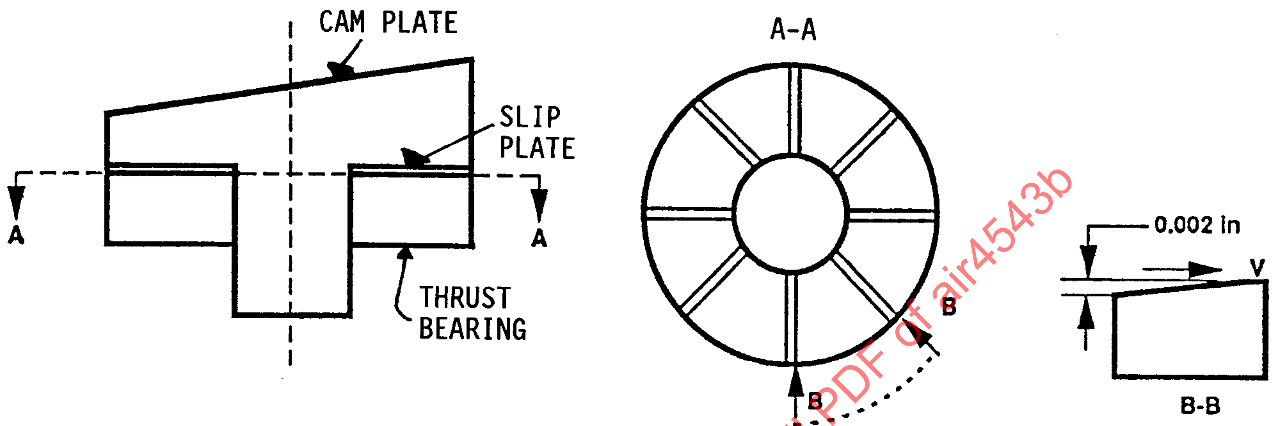
SOLUTION: Reduce pump response/increase reservoir pressure.

3.3.5.5 B-52 Pump Failures Early in Life

PROBLEM: Early pump failures due to rapid air turbine starts of the B-52 hydraulic packs (ten per shipset).

ISSUE: The turbines accelerated to full speed of 37 500 rpm (3750 at the pump shaft) in 1 s. The flat bronze thrust bearings galled due to failure of the oil film to carry the load.

ILLUSTRATION:



SOLUTION: The bearing pads were tapered to aid the development of a positive load-bearing oil film.

3.3.5.6 Force Fight Between Two Pumps in the Same System

PROBLEM: Variable delivery units pumping in the same hydraulic system may result in an unstable or oscillatory pressure.

ISSUE: Two variable delivery pumps in the same system have similar but slightly different response times and flow characteristics. Pumps will both attempt to dominate which will result in flow force fighting, pressure fluctuations, and pump crosstalk.

SOLUTION:

1. Make one pump a master and the other a slave by either:

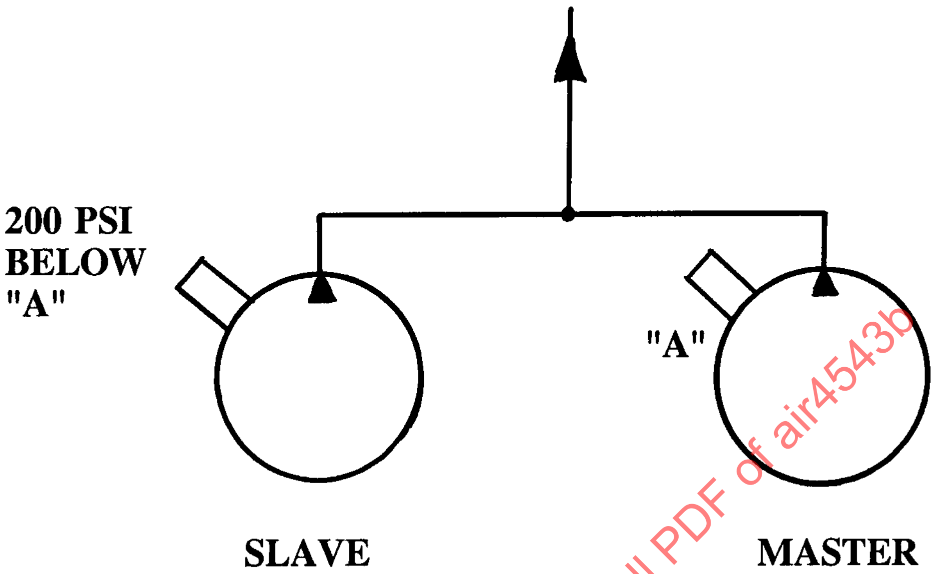
a. Setting one pump compensator (slave) 200 lbf/in^2 (1380 kPa) below the other (master) pump, or.

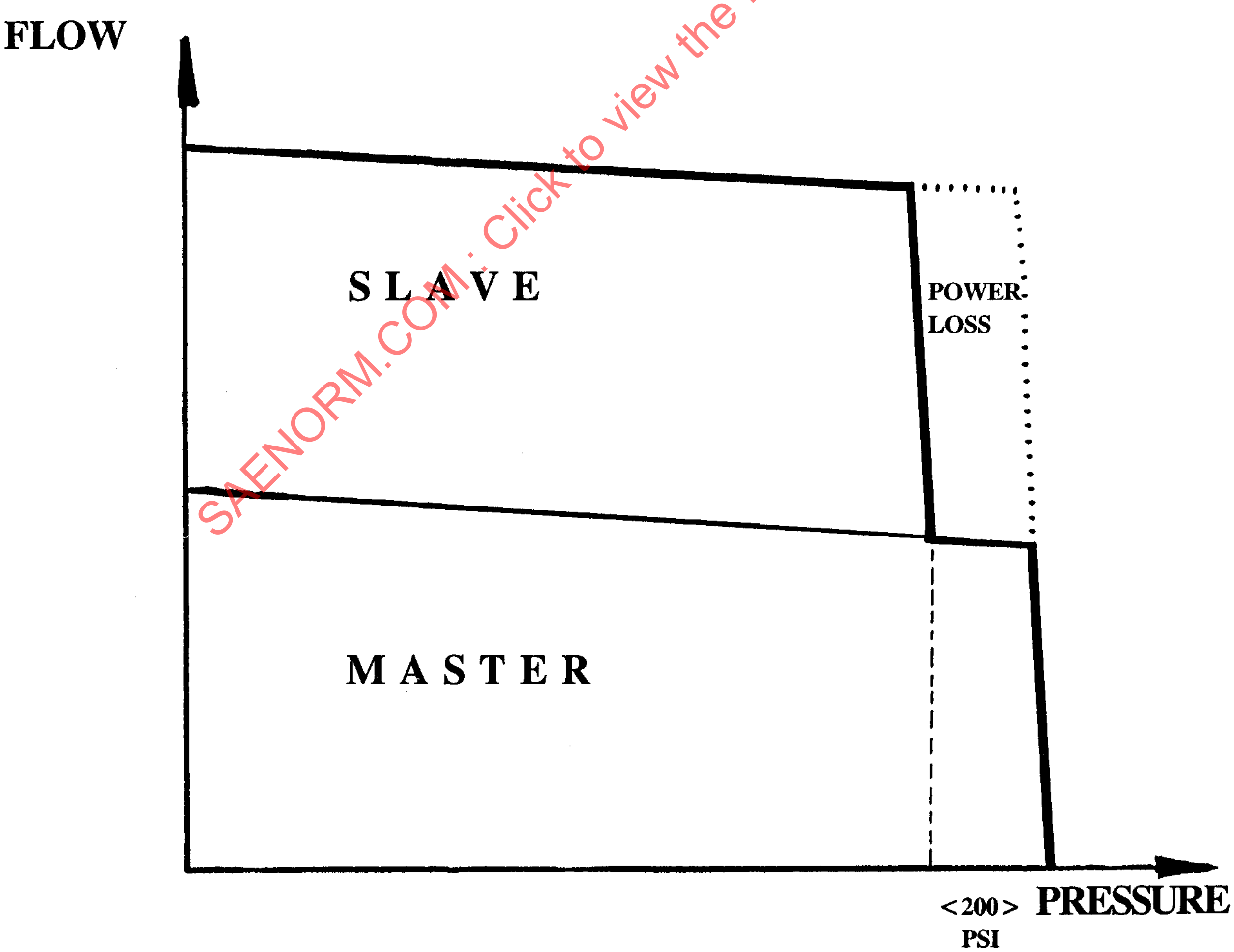
b. Incorporating a 50 to 200 lbf/in^2 (350 kPa to 1380 kPa) check valve on one pump discharge line (slave) to let the other dominate until higher flow by system is demanded of the slave pump.

2. Alternatively, analyze circuit / pump resonance. Set pumps with wide and opposite response.

ILLUSTRATION:

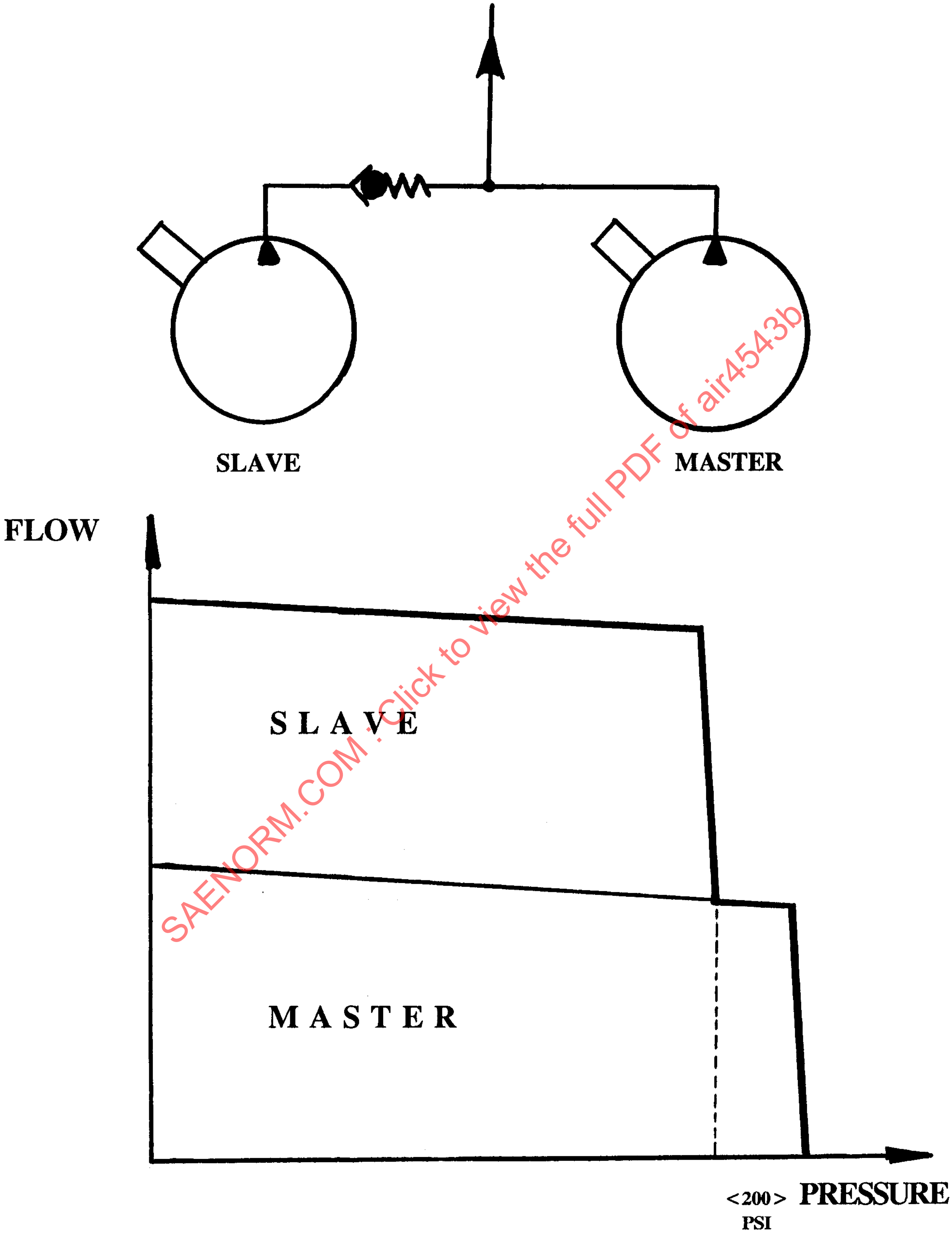
1. a. Compensator separation:





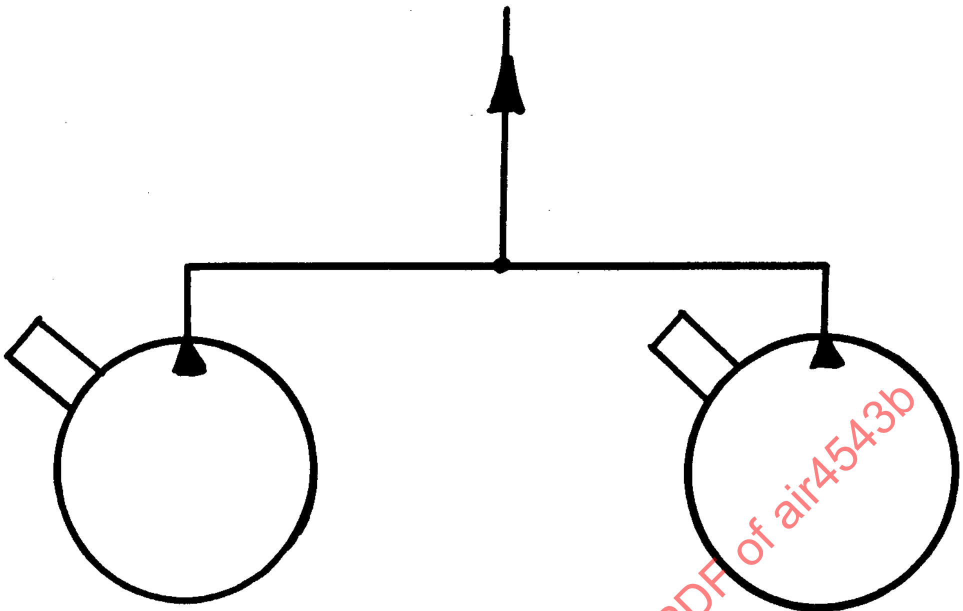
NOTE: Illustration pressure of 200 lbf/in2 (two places) is equivalent to 1380 kPa.

1. b. Check valve separation:



NOTE: Illustration pressure of 200 lbf/in2 is equivalent to 1380 kPa.

2. Performance separation:

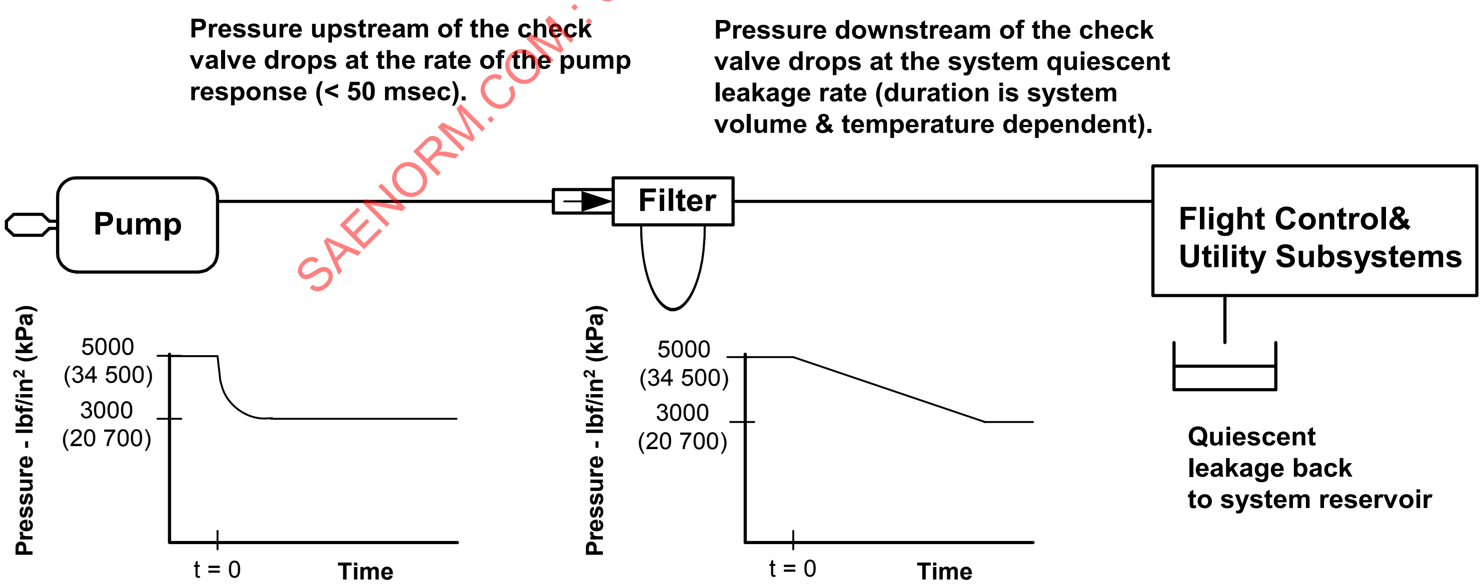


3.3.5.7 Dual Pressure Pump Instability

PROBLEM: Dual pressure pump unstable when switching pressure.

ISSUE: Switching from high pressure to low pressure causes the high pressure filter check valve to close resulting in inadequate system volume downstream of the dual pressure pump to maintain stability.

ILLUSTRATION:



SOLUTION: Specify a dual pressure pump response rate, from high pressure to low pressure, which is slower than the system quiescent leakage rate.

3.3.5.8 Timing of Dual Pressure Pumps

PROBLEM: Dual pressure pump timing optimization.

ISSUE: Pump timing is optimized at specified conditions to minimize pump generated pressure pulsations. Changes to any of these conditions can adversely affect the magnitude of the pressure pulsations.

a. Discharge Pressure

b. Shaft Speed

c. Displacement

d. Fluid Bulk Modulus (Temperature, Air Content)

SOLUTION:

a. Optimize the pump timing at the discharge pressure that is used in service most often (assuming the other discharge pressure setting doesn't generate unacceptable pressure pulsations).

b. Optimize the pump timing at a discharge pressure somewhere in between the two pump pressure settings (assuming neither discharge pressure setting generates unacceptable pressure pulsations).

NOTE: System designers should allow reasonable pulsation requirements for the non-optimized discharge pressure condition(s).

3.3.5.9 Pump Performance with Phosphate Ester Fluid

PROBLEM: Inconsistent performance characteristic of hydraulic pumps and motors.

ISSUE: If a phosphate ester (Skydrol) hydraulic circuit, whether a development, production, or aircraft, is brand new or refilled completely with new fluid, the hydraulic pump or motor can have different performance characteristic than fluid that has over 30 hours.

SOLUTION: If performance is critical, the hydraulic circuit needs a break-in run for 15 to 30 hours.

Phosphate ester fluid is made up of approximately 70 to 80% of an additive that does not have good initial lubricity. The additive needs to break down slightly to provide the lubricity. Hydraulic fluids MIL-PRF-5606 (inactive for new design), MIL-PRF-83282, and MIl-PRF-87257 do not have this problem as their additives are different.

3.3.5.10 Hydraulic Pump Fires

PROBLEM: A fighter aircraft experienced thirty in-flight failures a year, eight percent of total pump removals.

ISSUE:Fifty percent of in-flight failures shear the splined drive shaft.

Pumps removed often discolored or charred with occasional ruptured pump case. Four pump failures resulted in fires, one of which damaged the AMAD.

Suspect high pressure in pump case due to blockage of case drain circuit causes excessive pump wear or pump case rupture.

Ground test program on case drain circuit revealed debris clogged check valve in pump manifold.

SOLUTION:

Forced pump removal at 2300 hours. Case drain filter replaced every 200 hours.

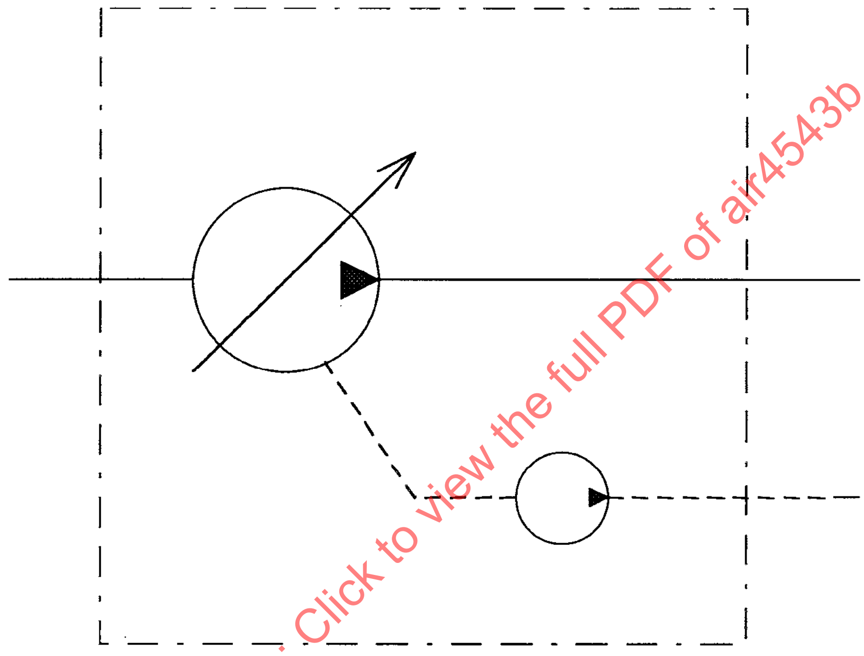
Check valves in case drain manifold removed to avoid jamming and back pressure on pump.

3.3.5.11 Pump Case Scavenge Device

PROBLEM: Low hydraulic pump reliability.

ISSUE: High pump case back pressures occur due to case drain line routing, contaminated filters, flow surges and restrictions. This results in pump overheating due to reduced case flow, deterioration of wear surfaces due to high internal pressure loading, fatigue of pump cases due to pressure spikes, high discharge pressures, and excessive heat rejection caused by higher discharge pressures, pump deterioration, pressure loading, and high fluid temperature.

ILLUSTRATION:



SCAVENGE PUMP SCHEMATIC

SOLUTION: Incorporate case scavenge device to provide positive case drain flow.

a. Improved heat rejection

b. Insulate pump case from back pressure
   1. Reduced pressure loading
   2. Improved fatigue life
   3. Steady outlet pressure

c. Reduced case temperature

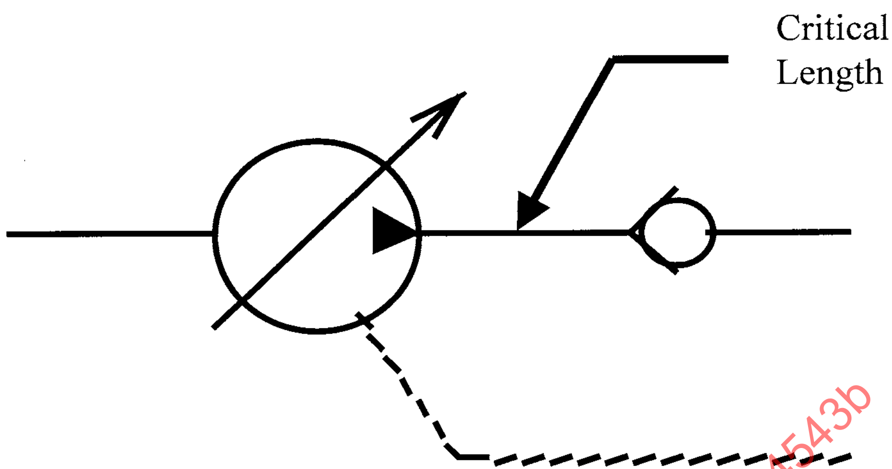
d. Improved reliability

3.3.5.12 Pump Instability Caused by Check Valve

PROBLEM: Pump instability due to pump outlet check valve location.

ISSUE: Pump instability due to close proximity of outlet check valve to pump. Location of the check valve in the pump outlet line can have a great impact on pump stability. Locating check valve too close to pump requires additional design considerations. Additional leakage may also be required to improve stability which results in reduced efficiency and increased power loss.

ILLUSTRATION:



SOLUTION:

a. Locate check valve as far away from pump outlet as possible.
   1. Improves stability
   2. Allows for lower internal leakage
   3. Improves performance
   4. Increases life

b. If check valve must be located close to pump, then special design considerations are required.
   1. Yoke over stroke angle
   2. Compensator spool clearances
   3. Compensator spool overlap
   4. Compensator spring rate

c. Check valve location and associated system volumes must be known early in pump design process to prevent problems and delays later in the program.

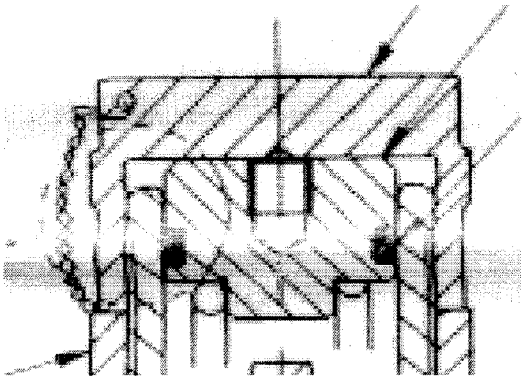
3.3.5.13 Pump Fire Test

PROBLEM: Requirement to pass a five minute, 2000 °F (1093 °C), flame test was added after the pump was designed.

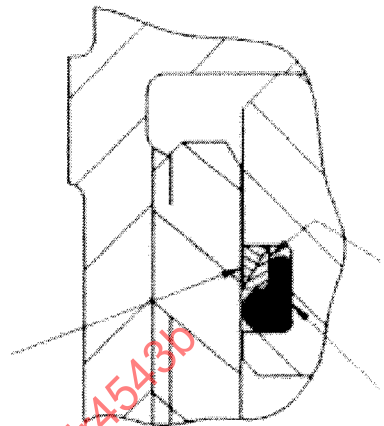
ISSUE: Because of the orientation of the pump in the aircraft, the compensator valve would be directly exposed to the flame. Seal damage and resulting external leakage would result.

Options considered were special fire resistant coatings, stainless steel shielding, thermal blanket insulating blanket and special seal design.

ILLUSTRATION:



End of Compensator



Detail of Seal Design

SOLUTION:

Incorporate special seal design with metal backups. This approach offers the advantages of:

a. Least cost approach

b. No additional envelope required

c. No added weight

d. Easy upgrade of fielded pumps

3.3.5.14 Bronze Pump and Motor Cylinder Blocks

[See AIR4543/1 for revised version. Ed.]

3.3.5.15 Pump Gearbox Installation – Shaft Seal Leakage

PROBLEM: Pump/motor gearbox installation – shaft seal leakage.

ISSUE: An engine pump installed on a gearbox eventually has hydraulic fluid shaft seal leakage. Due to the 1/4 in tube on the overboard drain system becoming clogged or restricted, pressure can build up and fail the gearbox seal. The result can cause gearbox lubrication oil to mix with the hydraulic oil. If the gearbox seal material is not compatible with the oils, seal failure will occur. Lubrication oil seals are not compatible with the phosphate ester commercial aircraft hydraulic fluids.

SOLUTION: Motors or pumps operating with different fluids or with different lubricants, such as intervening gearboxes, must have provisions for sealing incompatible fluids from one another. When this type of installation is used, adequate venting of the common chamber between the two seals is required. Small drain tubing has a tendency to clog up due to the on and off heat cycle operation. Use a 3/8 tube for engine installations in order to prevent clogging and pressure build up.

3.3.6 Seals

3.3.6.1 Low Friction Actuators and Dynamic Leakage

PROBLEM: Rejection of low friction actuators for dynamic leakage.

ISSUE: Actuators designed and qualified for low friction (manual reversion) applications sometimes fail leakage tests for no apparent reason. TFE capped rod seals, such as shown in Illustration 1, are usually employed to meet low friction.

ILLUSTRATIONS:

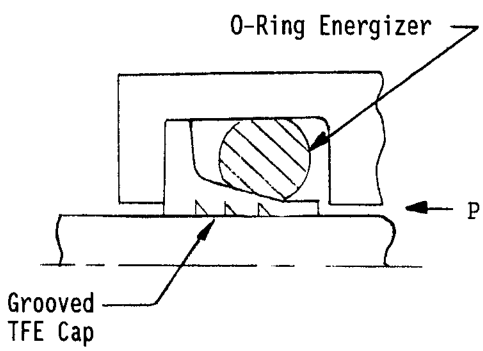


ILLUSTRATION 1

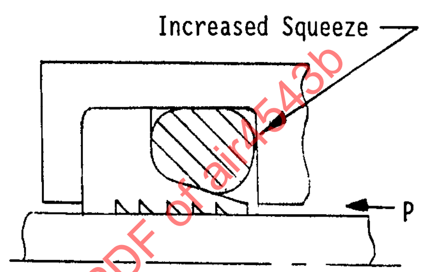


ILLUSTRATION 2

SOLUTION:

a. Low friction simultaneously with low leakage is a contradiction and a challenge. Analyses and development tests in concert with seal supplier(s) is required early to arrive at optimum seal configuration. (Reference Table 1).

b. Strive to qualify at least one alternate seal to preclude production and delivery problems later.

c. Roughening the surface of the seal with a mild abrasive prior to assembly can improve leakage. However, improvement is only temporary and uncontrolled process is undesirable.

d. Typical Example: An actuator with nine capped seals suddenly experienced a 50% rejection rate after several years of production. Rejections were eliminated by increasing squeeze (fill) from 51% to 91% and increasing the number of grooves (see Illustration 2). The friction penalty was doubled, but the total friction was still well within the 104 lbf (43 N) limit.

SEAL DESIGN CONSIDERATIONS:

a. Be wary of friction calculations. Eccentricities can double calculated values.

b. Provide smooth rod finishes – better than 8 µin (0.20 µm).

c. Design actuator to minimize piston cocking and side loads with resultant rod scratching.

d. Use largest possible seal groove cross section for maximum design flexibility.

e. Seal energizing forces should be sufficiently high and consistent over expected temperature and life. Take advantage of all the friction that is allowable.

f. Seal cap material should be homogenous – not grainy as is possible with some filled TFE.

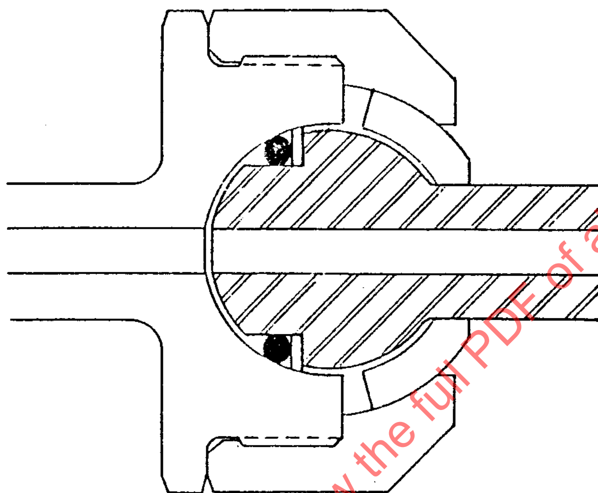
g. Annular grooves in the cap help reduce hydroplaning at higher piston velocities and reduce static leakage as well.

3.3.6.2 Spherical Ball Swivel Leakage

PROBLEM: Spherical ball swivel leakage.

ISSUE: Many spherical ball swivels tend to leak when initially pressurized. This is caused by the ball moving, within the clearance space, away from the seal.

ILLUSTRATION:



SOLUTION:

a. Use coiled tube rather than spherical swivels, if possible.

b. Increase seal squeeze, and reduce ball to housing clearance as much as possible in the design of the swivel.

3.3.6.3 Servoactuator Piston Seal Blowby

PROBLEM: Servoactuator piston seal blowby.

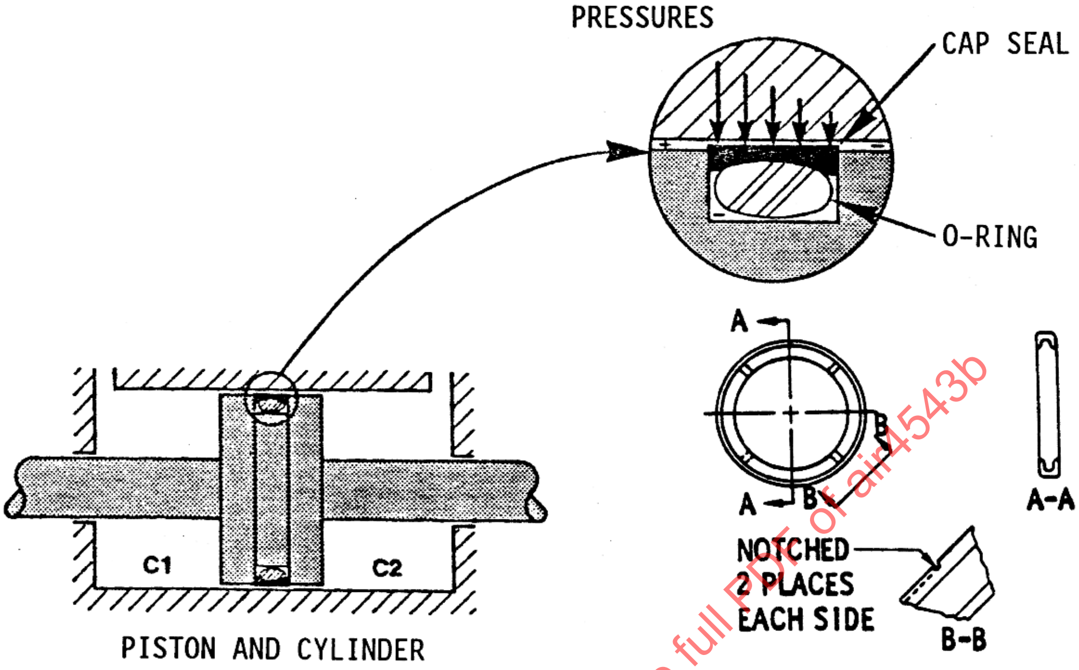
ISSUE:

a. Use of the TFE cap seal to improve wear life of piston head O-ring may result in piston stalling following a command to reverse the direction.

b. Edge of TFE cap may seal on the sidewall of the seal groove following the reversal of cylinder pressures. This can momentarily trap lower pressure under the cap seal and allow bypass flow over the top of the seal.

c. Without the sidewall notches, the piston cap may seal on the sidewall of the groove rather than on cylinder I.D. This can result in leakage flow from C-1 to C-2.

ILLUSTRATION:



SOLUTION: Specify the addition of notches on the sidewalls of TFE or similar piston cap seals. Refer to AIR1243 for more discussion.

3.3.6.4 Tight Rod Seal Leakage Requirements with Single Seals

PROBLEM: Tight rod seal leakage requirements are difficult to achieve using a single seal.

ISSUE: The oil film on the rod will pass through a single stage seal easily.

SOLUTION: Use dual stage seals. Test and field experience have shown dual stage seals outperform single stage seals by a wide margin. A vent between the seals is not necessary contrary to initial beliefs. Pressures trapped between the two seals do not exceed system pressure. Non-vented designs offer a smaller envelope size and reduced complexity. Primary seals in un-vented applications must have an anti-extrusion device on both sides.

3.3.6.5 Rapid Wear of a PTFE Piston Head Seal

PROBLEM: Rapid seal wear of a PTFE piston head seal.

ISSUE: Nodules left on surface after "thin dense chrome" plating abraded seal. Profilometer readings showed the proper surface finish. SEM[1] analysis was required to detect presence of nodules.

SOLUTION: Perform a honing operation following the plating process. Allow a sufficient plating thickness for honing.

NOTE: [1] SEM (scanning electron microscope)

3.3.6.6 Use of Small Section O-Rings

PROBLEM: Use of small section O-rings (Less than 0.103 in (2.6 mm) cross-section diameter).

ISSUE: Nitrile O-rings with a cross-section diameter that is less than 0.103 in (2.6 mm), and the equivalent proprietary seals to replace them, have more leakage than O-rings and other seals that have a cross-section diameter, or equivalent, of 0.103 in (2.6 mm) or larger.

Problem	Effect
Part and seal tolerances become a larger and larger percentage of the basic cross-section size as the basic cross-section size gets smaller and smaller.	Maintaining adequate seal squeeze becomes more difficult as the basic size gets smaller and smaller.
The ratio of surface area to surface volume increases as the basic cross-section decreases.	Seal life decreases as the basic cross-section decreases.
MIL-G-5514F (inactive for new design) has inadequate seal squeeze and inadequate groove width on some several glands.	Leaks, groove overfill

[No known superceding document for MIL-G-5514, relevant SAE document is AS4716 Ed.)

SOLUTION: Avoid the use of small cross-section seals. It may be possible to use a -102 through -109 O-ring (0.103 in [2.6 mm] cross-section) with an appropriate nonstandard backup ring. If a small cross-section must be used, the following changes are recommended:

a. Provide 0.005 to 0.007 in (0.13 to 0.18 mm) minimum squeeze in the worst possible combination of part and seal tolerance, part position, and I.D. installation stretch (this may require the use of a nonstandard backup ring).

b. Provide adequate groove width.

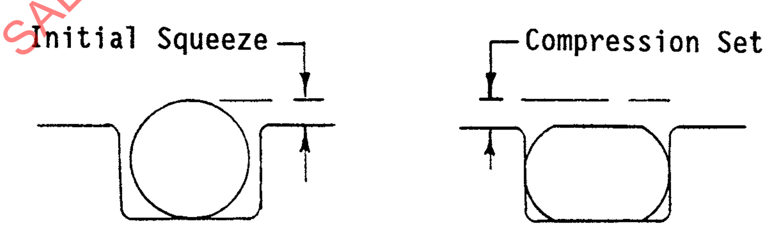
c. Avoid using more than 5% installed I.D. stretch.

3.3.6.7 Use of Small Cross-Section O-Rings

PROBLEM: Use of small cross-section O-rings.

ISSUE: Small cross-section static seals, typically 0.070 in (1.8 mm) diameter, per MS28775 (superceded by SAE AS28775) or MIL-P-83461 (Nitrile) (superceded by AMS-P-83461) are prone to leakage failures. A contributing factor may be local "necking down" caused by uneven installation stretch forces for large sizes. However, the primary problem is due to compression set of the elastomer, which is critical for small cross-sections. High temperature operation for a few hours results in a permanent loss of squeeze with leakage pronounced at normal as well as colder temperatures and low pressures.

ILLUSTRATION:



SOLUTION:

Where external seepage would result, avoid small cross-section seals especially for sizes above a 2 in (51 mm) diameter. Where they must be used, the elastomer compound should be changed:

a. Fluorocarbon per MIL-R-83485 (superceded by AMS-R-83485) has been successful in solving problems where the temporary loss of squeeze below -40 °F (-40 °C) due to the coefficient of shrinkage can be accepted, i.e., slight seepage at low temperature.

b. Phospho Nitrilic Fluoro elastomer (PNF) per MIL-P-87175 (now cancelled, no known superceding document) has been successful solving problems where squeeze must be maintained down to -65 °F (-54 °C).

[It is noted that SAE AMS CE Committee is preparing an updated document for Phosphonitrilic Fluoroelastomer (Fz) seals. Ed.]

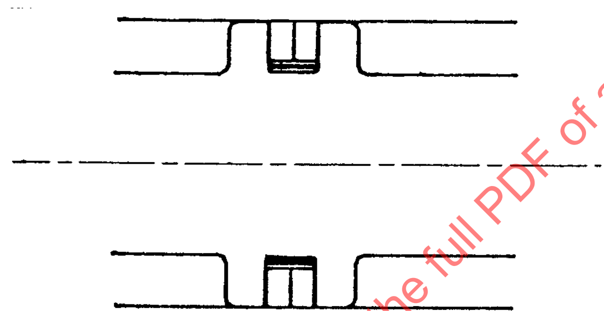
c. Fluoro Silicone per MIL-R-25988 may prove to be an acceptable lower cost alternate to PNF in certain applications such as face seals.

3.3.6.8 Small Actuator Piston Ring Leakage Blowby

PROBLEM: Small actuator piston ring leakage blowby.

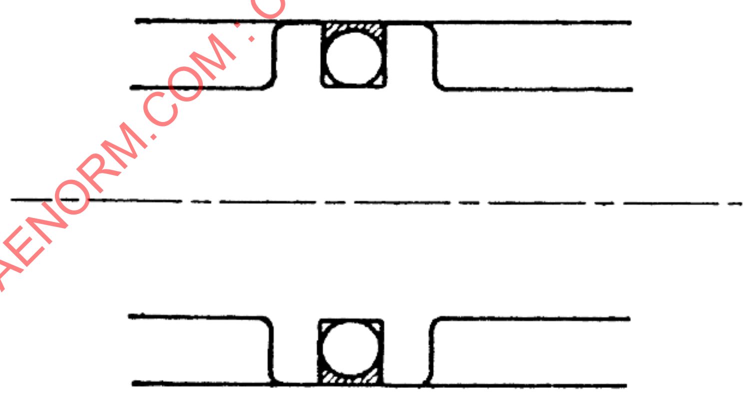
ISSUE: Small servoactuator has low flow rate, piston ring leakage prevents the cylinder to buildup ΔP on the small piston area. The actuator lost response.

ILLUSTRATION:



SOLUTION: Design with no-leakage-type piston ring with minimum shuttling of the seal during reversing of the flow direction.

ILLUSTRATION:



3.3.6.9 Deterioration of O-Rings Exposed to Different Fluids

PROBLEM: Deterioration of O-rings exposed to different fluids, e.g., pump coupling shaft in flooded gearbox.

ISSUE: O-ring compound not compatible with gearbox fluid and fluid from pump shaft seal leakage.

SOLUTION: Identify gearbox fluid in pump specification. Use AFLAS compound to -40 °F (-40 °C).

3.3.6.10 Fluid Leakage in Guided Missiles

PROBLEM: Fluid leakage in guided missiles.

ISSUE:

a. Small volumes - no leakage tolerable

b. -65 °F to +275 °F (-54 °C to 135 °C) storage and operation

c. Immediate full performance required

d. No warm-up

e. MIL-P-25732 (inactive for new design, superceded by AMS-P-83461) Nitrile seals found lacking due to permanent set at +275 °F (+135 °C)

SOLUTION:

a. Use fluorosilicone O-rings.

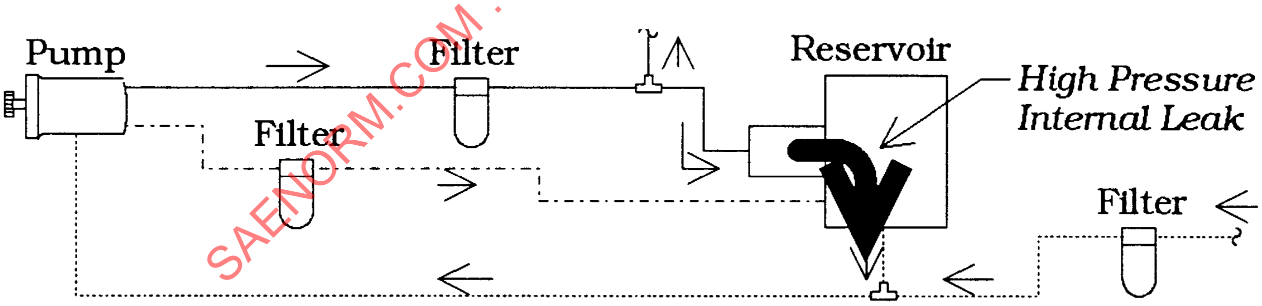
b. Shore 70 minimum for toughness and wear.

3.3.6.11 Reservoir High Pressure Seal Leakage

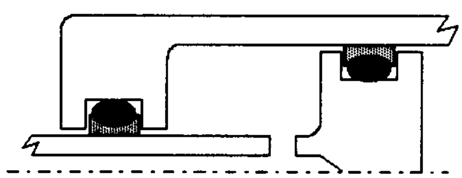
PROBLEM: High Pressure Internal Leakage in Bootstrap Reservoirs.

ISSUE: Extrusion/feathering of piston and rod seals results in high volume internal leakage within the reservoir and elevated system temperatures to 260 °F (127 °C).

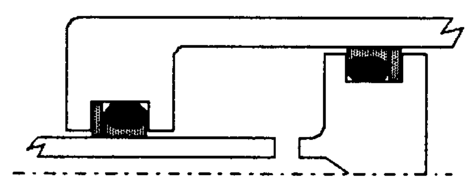
ILLUSTRATION:



SOLUTION: Utilize extrusion resistant materials and/or dual configurations on the high pressure piston and rod seal locations of a bootstrap reservoir and/or dual.



Original Seal Concept



Improved Seal Concept

3.3.6.12 Wear of Aluminum Bore by PTFE Seals

PROBLEM: Wear on soft anodized bores by unfilled modified PTFE seals.

ISSUE: Unfilled modified PTFE seals subjected to large numbers of cycles.

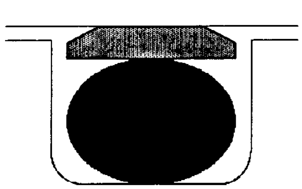
Loose sealing integrity due to:

a. Wear of soft anodized aluminum bores

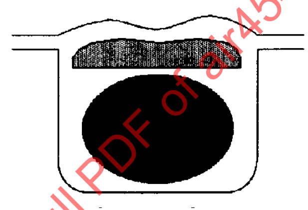
b. Wear on energizing O-rings

c. Wear of PTFE cap strips

ILLUSTRATION:



Newly Installed
Static Seal



Static Seal Worn by
Oscillating Pressures

SOLUTION: Utilize seal materials and surface treatments which are more resistant to high cycle wear

a. Hard anodized or nickel plated bores

b. Mineral filled PTFE seals, non-abrasive

c. Viton O-rings (if temperature hardening is evident)

3.3.6.13 Failure of Nylon Poppet Valve Seat

PROBLEM: Failure of Nylon, low-leakage, poppet shut-off valves.

ISSUE: Cracking and dislodgement of nylon poppet seats due to:

a. Embrittlement from overheating

b. Moisture degradation

c. Accumulated temperature cycles

d. Unexpectedly high frequency of operation

e. Combinations of all of the above

Results in:

a. Failure to stop flow when poppet is activated

b. Migration of seat material to downstream components

SOLUTION:

Step 1 - Assure that life requirements adequately cover service usage

a. Temperature range

b. Cycles of operation

c. Operational cycles vs. temperature

Step 2 - Assure that selected seat material can meet all service life requirements

- Design poppet for positive seat retention to minimize migration of particles if cracking occurs

3.3.6.14 Seal Degradation at High System Temperatures

PROBLEM: Hydraulic seal failure and leakage caused by high system temperatures.

ISSUE:

a. A major fighter aircraft was experiencing O-ring, cap strip seals, and poppet seal failures at about 200 flight-hrs into a 6000 flight-hrs useful aircraft life.

b. Higher than normal temperatures, up to 260 °F (127 °C), caused by degrading components was suspected as the major cause.

c. Qualification test procedure review indicated no endurance or impulse test cycling above 225 °F (107 °C). MIL-H-8775 (superceded by AS8775) specifies 275 °F (135 °C) or tests at expected temperatures.

SOLUTION:

a. Specify a portion (25% minimum) of the cycling endurance and impulse test for components to be conducted at 275 °F to insure seal performance.

b. Incorporate a fluid temperature indicator into system return lines for indication of system overheat/leaking components.

c. In known hot areas, specify high temperature seals, i.e. MIL-R-83485 (superceded by AMS-R-83485).

d. Tests at 275 °F (135 °C) have revealed two seal failures on a new aircraft program.

3.3.6.15 Pump cavitation on start-up

PROBLEM:Stiction of a Rubber Seal on a Reservoir Piston Caused Pump Cavitation.

ISSUE: During start-up, with a day or more of system idle time, an aircraft experienced pump cavitation because of the loss of reservoir pressurization.

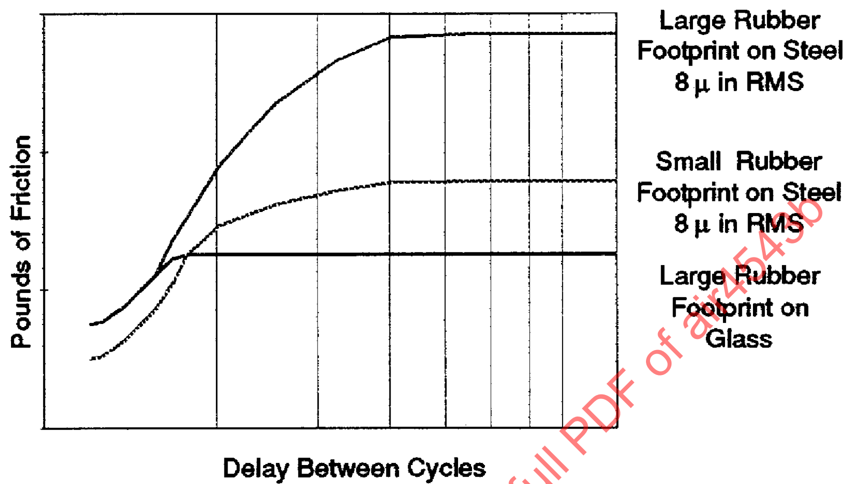
Loss of pressurization was traced to the sticking of the reservoir piston rubber O-ring seal on the reservoir housing.

The reservoir housing material is aluminum with hard coat (Type 3) anodize. The hard coat finish on the micro level looks like a dried lake bed. A crusty surface with voids and crevices.

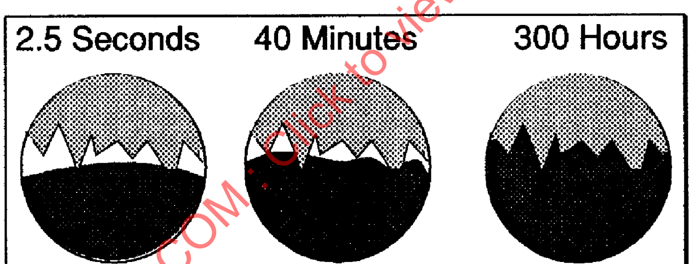
The dynamic seal used was a design with a large rubber footprint.

Hard coat anodize and a large rubber footprint is an unfavorable combination. The rubber extrudes into the voids and crevices of the hard coat. The degree of elastomer imbedding is time dependent but reaches a finite value ≈100%. The force required to shear is proportional to the amount of imbedded rubber and potentially can exceed the force available from the reservoir pressurization device.

ILLUSTRATION:



NOTE: Surface finish of 8 μin RMS (two places) is approximately equivalent to an ISO Grade Scale number of N4.



SOLUTION: Be cognizant of the micro-surface, its relationship with seals and select seals and finishes appropriately. For this particular application, a new low pressure seal with a small controlled rubber footprint was selected. This limited the amount of extruded rubber, therefore, reducing the shear force required to an acceptable level.

3.3.6.16 Use of Face Seals versus Transfer Tubes at Hydraulic Interfaces

PROBLEM: Leakage of face seals.

ISSUE: When to use face seals versus transfer tubes (quills) with diametrical seals.

SOLUTION: Face seals require that the surrounding structure and retention be stiff enough to prevent flexing which can cause gaps, nibbling of seals, and leakage.

Providing adequate stiffness on high pressure systems [above 3000 lbf/in^2 (20 700 kPa)] can result in a weight penalty, especially on multi-port installations.

F18 C/D reservoir used multi-port face seal interface with aircraft for easy maintenance.

F18 E/F had to use quills because of weight considerations and 5000 lbf/in^2 (34 500 kPa) pressure level.

3.3.6.17 Damage to Seals from Hydraulic Pump Ripple

PROBLEM: Dynamic seals leakage due to hydraulic pump ripple.

ISSUE: Dynamic seals can be damaged when exposed to hydraulic pump ripple. The elastomer is typically eaten away from under the cap.

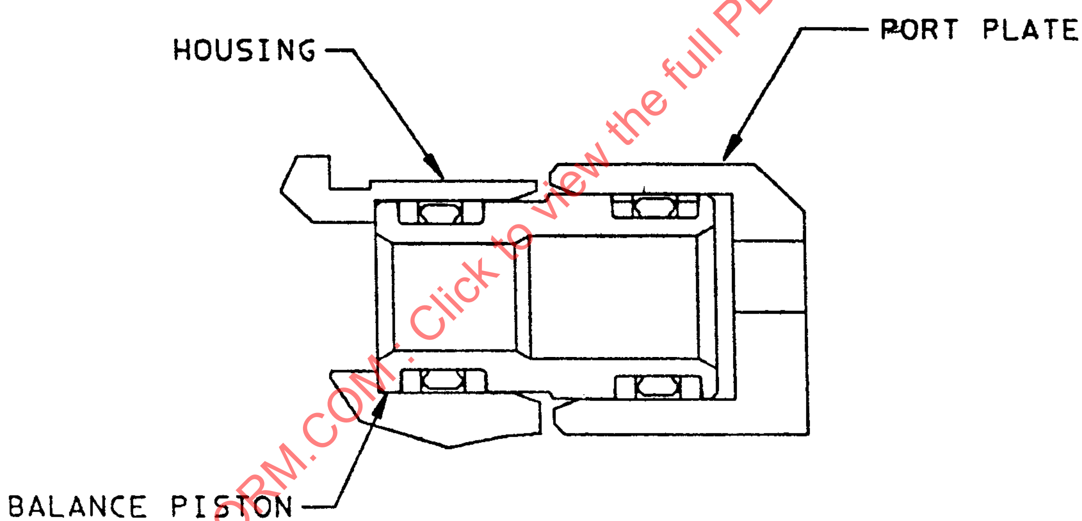
SOLUTION: Testing has proven that the Shamban plus Seal 11 with two backups or Greene, Tweed Ener-Cap seal with two backups perform well. The preferred elastomer when operating in MIL-PRF-5606 or MIL-PRF-83282 is Viton GLT per MIL-R-83485 (superceded by SAE AMS-R-83485).

3.3.6.18 Unbalanced Transfer Tubes

PROBLEM: Dithering wear of transfer tube seals in motors and pumps.

ISSUE: Cyclic/reversing pressures on floating port plate type units causes dithering wear on seals at high frequency. Can cause leakage and contamination debris

ILLUSTRATION:



SOLUTION: Unbalancing of transfer tube causes greater force in one direction and prevents dithering motion, thus reducing seal wear and leakage.

3.3.6.19 Reservoir Relief Valve Poppet Seal Failures

PROBLEM: Reservoir relief valve poppet face seal rings (Nitrile) were bulging out of retainer groove and causing complete dumping of reservoirs and loss of system.

ISSUE: Relief poppets were experiencing failure to reseat due to lifting of seal face which resulted in complete loss of system fluid. Sticking of Nitrile face seal to mating aluminum flat seat during relieving operation was suspected to cause bulging and deformation of sealing surface.

Test program utilizing Viton, MIL-R-83485 (superceded by AMS-R-83485) face seal and Teflon coated seat demonstrated greatly increased life at 275 °F (135 °C) compared with Nitrile face seal and anodized aluminum seat.

SOLUTION: Viton seal and Teflon coated valve seat have been incorporated into reservoir relief valves.

3.3.6.20 Uncut Back-Up Rings and Pressure Trapping

PROBLEM: Trapped pressure in a single seal caused excessive actuator friction.

ISSUE: The free fall requirements of emergency landing gear extension forced the landing gear actuator to need a lower than normal unpressurized dynamic friction. During initial gear tests, fluid pressure trapped between the uncut backup rings in a two backup width piston groove caused the landing gear actuator to have unacceptably high friction

SOLUTION: Replace the uncut backup rings on the piston with scarf cut backup rings.

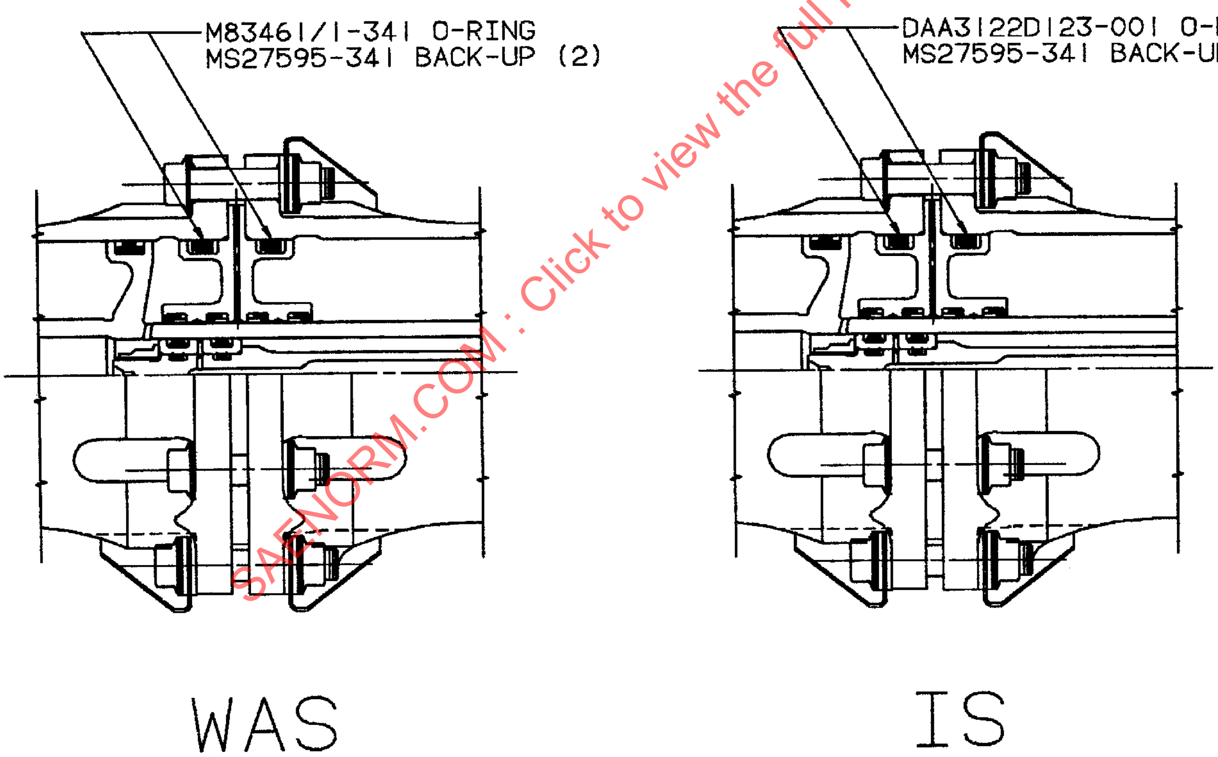
3.3.6.21 Nitrile O-Ring Leakage at -65 °F (-54 °C)

PROBLEM: Had static O-ring seal leakage at -65 °F (-54 °C) even though squeeze was greater than called for by MIL-G-5514 (inactive for new design,) [5.36%/0.011 in (5.36%/0.3 mm) minimum squeeze]

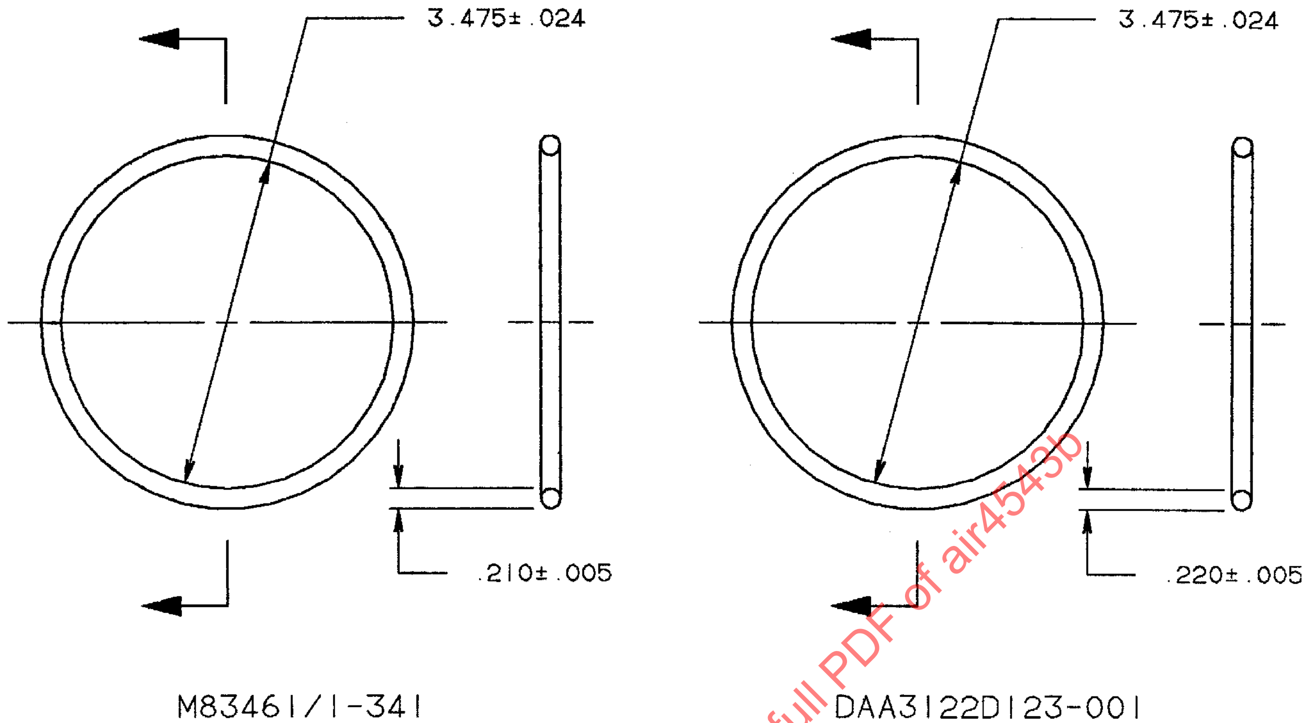
[No known superceding document for MIL-G-5514, *relevant SAE document is AS4716 Ed.)*

ISSUE: Leakage occurred at 4000 lbf/in^2 (27 600 kPa) when barrel would be expected to breathe by approximately 0.001 in (0.03 mm). Parker Handbook says high temperature Nitrile good to -20 °F (-29 °C).

ILLUSTRATION:



[M83461 packings inactive for new design, superceded by AMS83461 packings, MS27595 retainer inactive for new design, no superceding standard known Ed.]



NOTE: Illustration dimensions of 0.210 in ± 0.005 in, 0.220 in ± 0.005 in, and 3.475 in ± 0.024 in (two places) are equivalent to 5.3 mm ± 0,12 mm, 5.6 mm ± 0.12 mm and 88.3 mm ± 0.6 mm (two places), respectively.

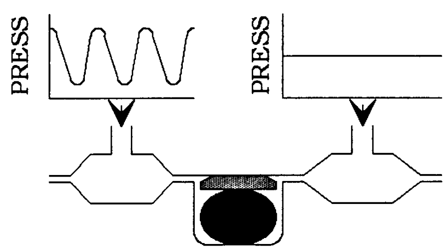
SOLUTION: Changed to DAA3122D123 seal wherein cross section was enlarged by 0.010 in (0.3 mm). Minimum squeeze now 9.71% or 0.021 in (0.5 mm) passed test.

3.3.6.22 Seal Wear and Intersystem Leakage

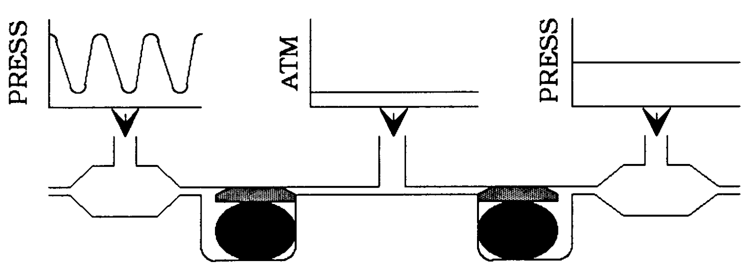
PROBLEM: Static seal degradation due to out-of-phase pressure cycling.

ISSUE: High cycle exercising of static seals within grooves can cause internal or intersystem leakage due to wear of sealing elements and surfaces.

ILLUSTRATION:



SOLUTION: Utilize dual seals (vented to atmosphere) where mis-phased oscillating pressures of similar magnitude can be expected.

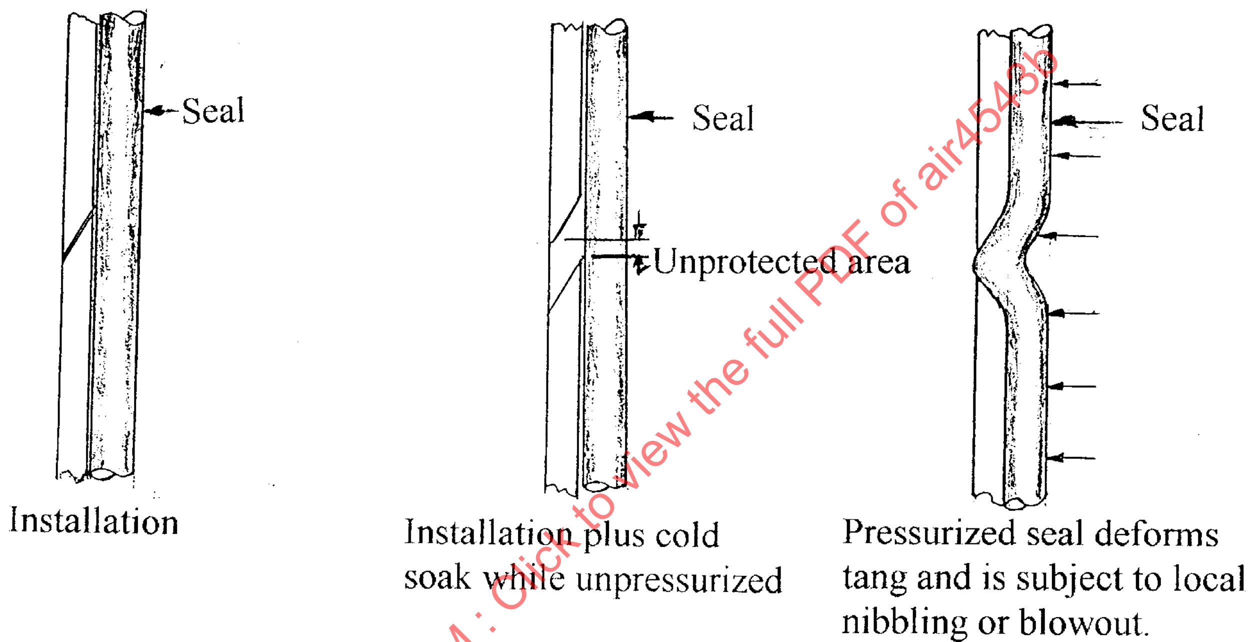


3.3.6.23 Seal Nibbling and Large Scarf-Cut Backup Ring

PROBLEM: Large diameter scarf-cut Teflon backup rings may allow seal nibbling or blowout.

ISSUE: When subjected to low temperatures with zero or very low pressure applied, Teflon backup rings will shrink. On large size static seals [approximately 4 in (102 mm) bores and larger sizes], 0.052/0.045 in (1.3/1.1 mm) thickness MS28774 (inactive for new design, no known superceding document, AS4716 is relevant SAE document) scarf cut backups will gap beyond the scarf cut length with conditions of an initial room temperature followed by a cold soak at approximately -40 °F (-40 °C) and with zero or very low pressure applied. Subsequent high pressure application, while in the shrunken condition, will drive the elastomer into the gap and subject it to possible blowout or nibbling failures.

ILLUSTRATION:



SOLUTION:

a. Use uncut Teflon backups for larger bore sizes. Do not use MS28774 (inactive for new design) which are not designed for current gland standards.

b. An applied axial seal pressure of approximately 100 lbf/in^2 (690 kPa) or greater will prevent shrinkage at low temperatures. (Considers elastic modulus of Teflon, coefficient of friction of Teflon on a smooth metal surface and the projected axial force on the backup due to pressure.) However, this is not a situation that one would live with.

3.3.6.24 PTFE Backup Rings in AS4716 Seal Grooves and Pressure Trapping

PROBLEM: Uncut PTFE backup rings per MIL-G-5514 (inactive for new design) act as a seal and trap pressure when AS4716 seal groove is used. Higher pressures, 5000 lbf/in^2 (34 500 kPa), may exacerbate this problem.

[No known superceding document for MIL-G-5514, relevant SAE document is AS4716 Ed]

ISSUE: Uncut backup rings for MIL-G-5514 seal grooves, due to interference with bottom of groove, can trap pressure resulting in jamming of moving parts. Fatigue failures also occur which leads to internal leakage. Five instances on spool/sleeve assemblies. One instance on aluminum threaded pressure plug, piston type seal.

SOLUTION:

a. On aluminum plug, redesign to MIL-G-5514 (inactive for new design) groove with scarf cut backup rings.

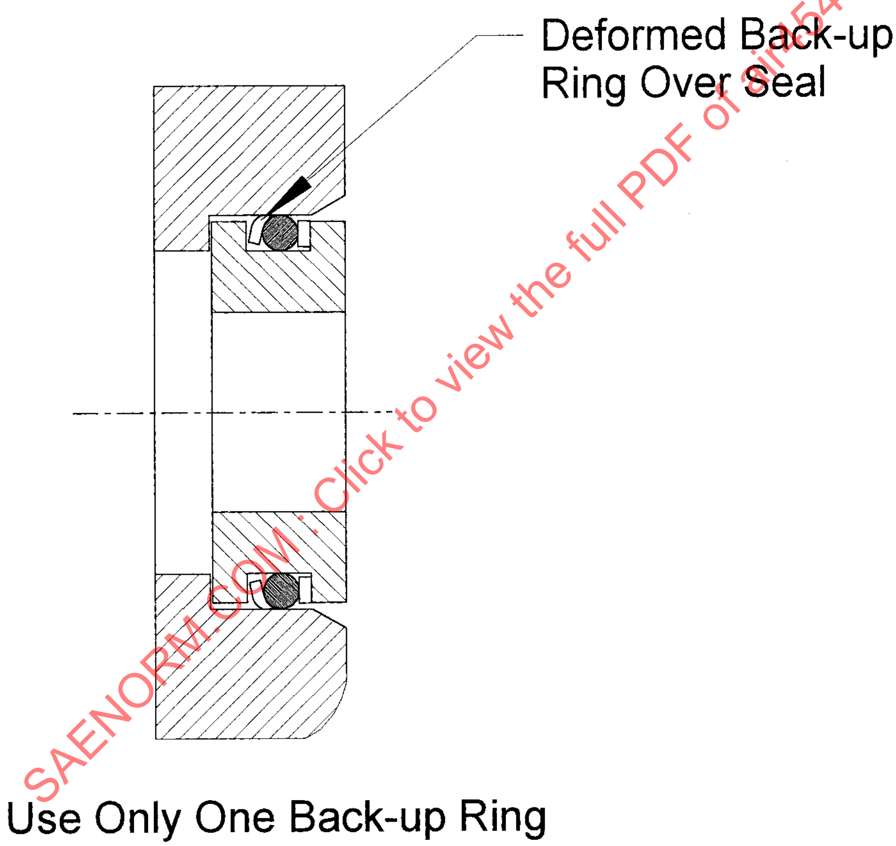
b. On static seals on spool/sleeve assembly, redesign to AS4716 solid delta backups with pressure relieving notches. Also notched sleeves on several designs.

3.3.6.25 Damaged Backup Ring

PROBLEM: Backup ring deformation during assembly.

ISSUE: Installation of O-ring seal with two backups resulted in first backup deforming over O-ring.

ILLUSTRATION:



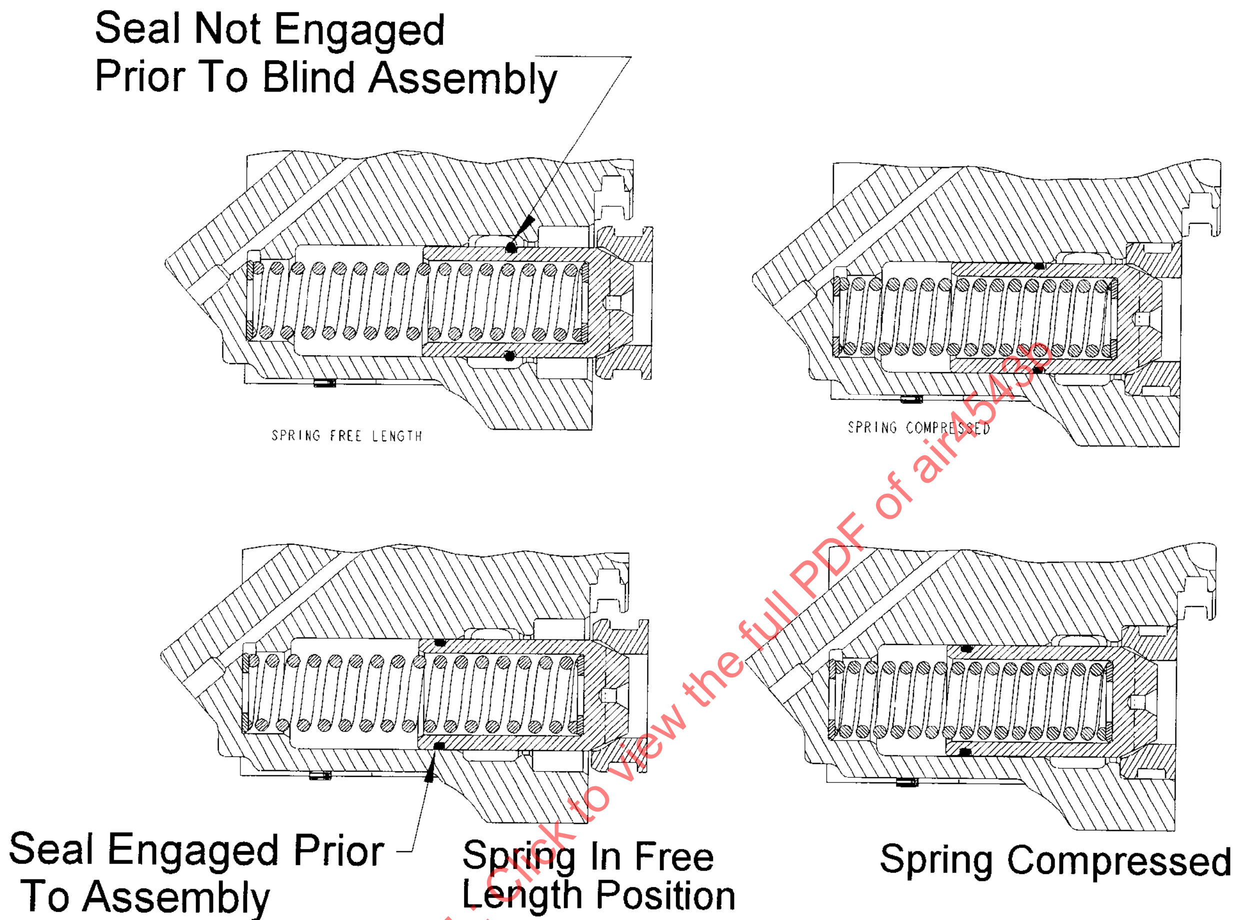
SOLUTION: Change design to eliminate upstream backup, i.e., single backup design which has short insertion of backup ring.

3.3.6.26 Seal Engagement

PROBLEM: Piston seal installation and alignment is difficult and results in damage.

ISSUE: Seals are damaged and leak on pistons which are used to preload springs in components. Seals do not engage bores before springs begin to compress.

ILLUSTRATION:



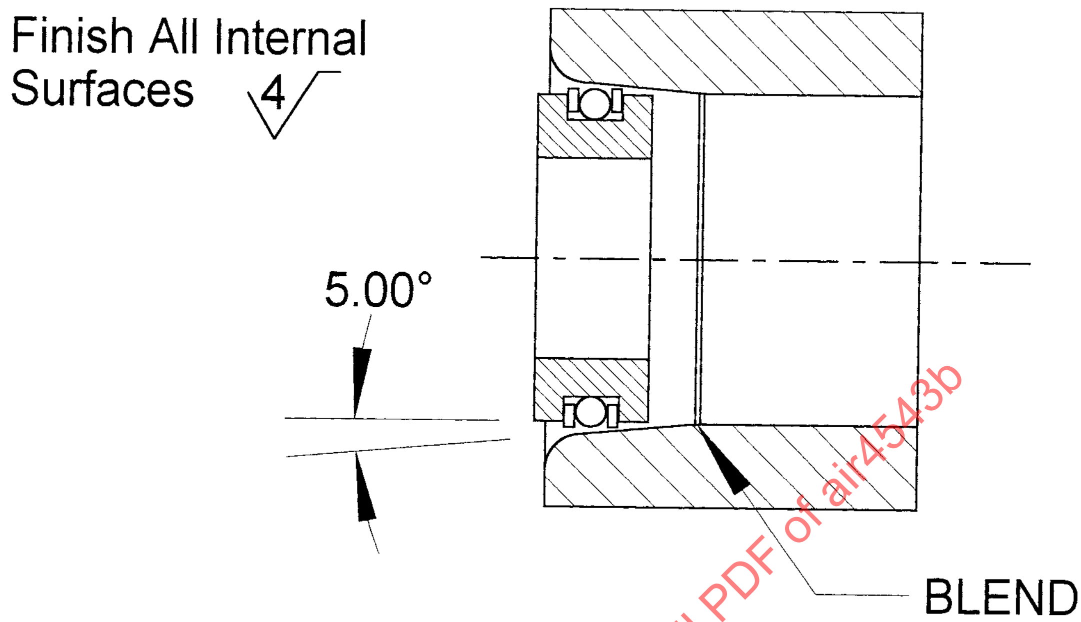
SOLUTION: Design components so that seals engage bores prior to assembly of mating components, i.e., compressing springs. Assembly is simplified and avoids damage to seals.

3.3.6.27 Compressing Solid Backup Rings

PROBLEM: Damage occurring to backup rings during installation.

ISSUE: Upon installation of solid backup rings and O-rings of a piston seal into a bore, damage to backup rings occurred.

ILLUSTRATION:



NOTE: Surface finish of 4 μin is approximately equivalent to an ISO Grade Scale number of N3.

SOLUTION: Use a compression type assembly tool to size backup rings prior to assembly into bores.

3.3.7 Tubing, Fittings, Bosses, and Hoses

3.3.7.1 Hydraulic Component and Tubing Installation Tolerances

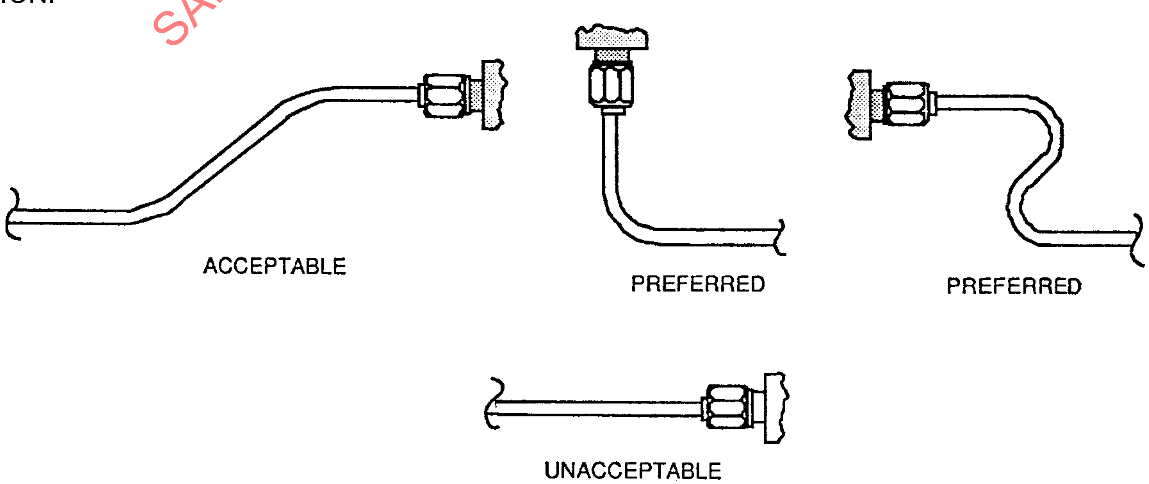
PROBLEM: Hydraulic component installation/tubing tolerances.

ISSUES:

a. During initial installations, tubes without flexibility cannot be sufficiently extended for fit.

b. Tubes without flexibility sustain damage from pressure surges and vibration.

ILLUSTRATION:



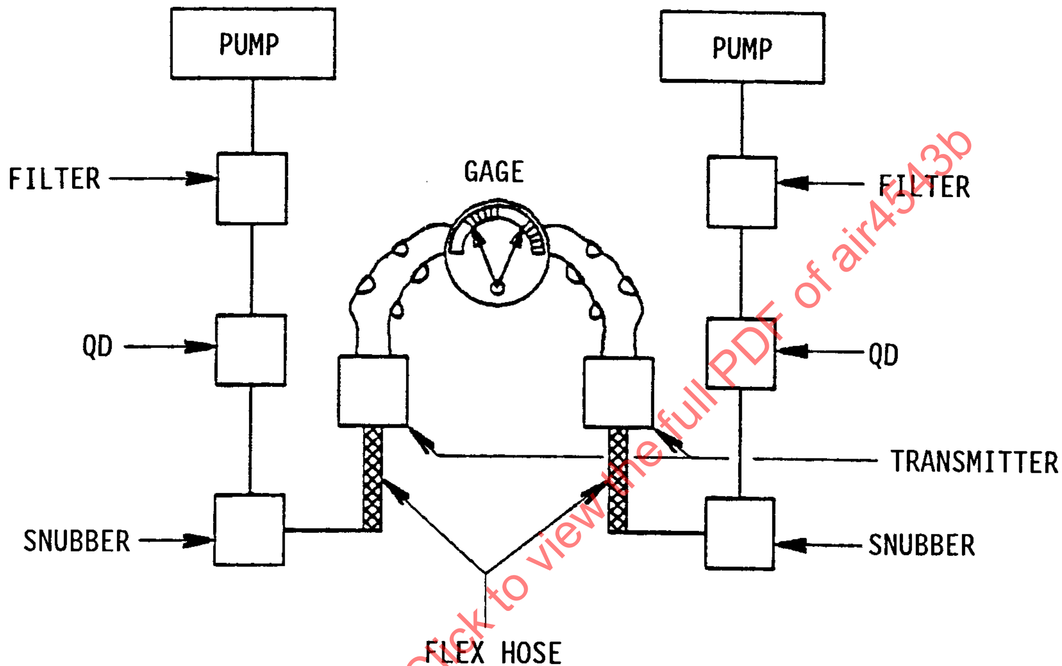
SOLUTION: Incorporate bend(s) in the hydraulic tubing to increase the tolerances and dissipate surges.

3.3.7.2 False Pressure Transmitter Indication

PROBLEM: False pressure transmitter indication.

ISSUE: While both pumps were off one gage indicated a pressure reading. It was discovered that the inner lining of the flex hose collapsed and had acted like a check valve.

ILLUSTRATION:



SOLUTION: Replace the flex hose.

3.3.7.3 Hydraulic Tubing Failures and Bend Ovality

PROBLEM: Hydraulic tubing failures due to excessive ovality in the bends.

ISSUE: Most tube assembly failures occur in a bend. A number of failures on an aircraft occurred due to excess ovality caused by worn shop tools. The specification limit for CRES tubing is 5%, for titanium, 3%.

SOLUTION: Better manufacturing and quality control.

3.3.7.4 Leaking Fittings

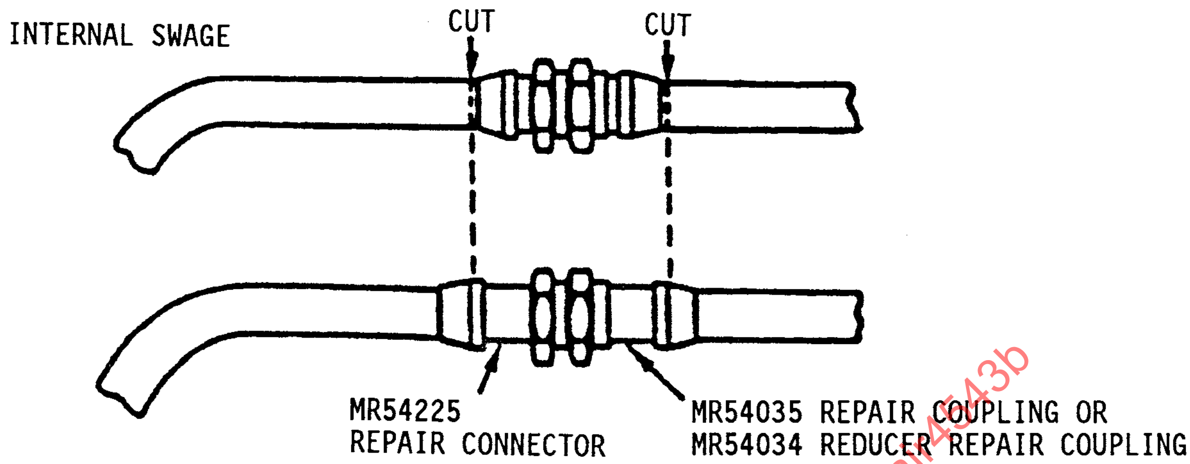
PROBLEM: Leaking fittings.

ISSUE:

a. A repair fitting is an option for leaky and/or damaged fittings if a sufficient clearance and straight tubing section is available.

b. Leaky or damaged fittings cannot be repaired.

ILLUSTRATION:



SOLUTION: Allow sufficient clearance and straight tubing sections at fittings so that repair fittings may be employed.

3.3.7.5 Male / Female Lip Seal Connection Damage

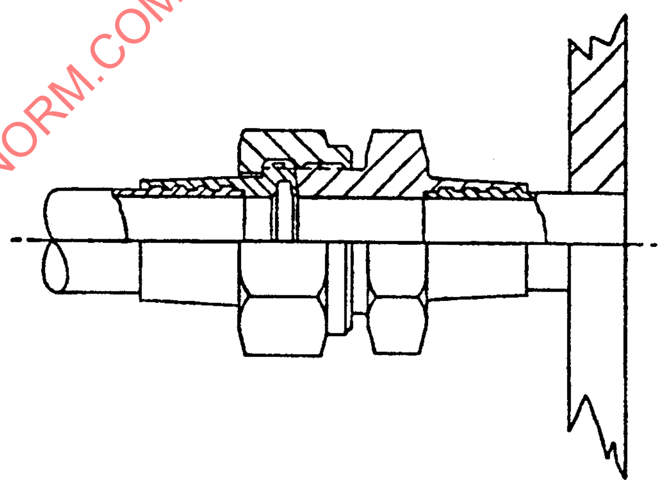
PROBLEM: Male / female lip seal connection damage.

ISSUE:

a. When a male/female connection of a permanent installation or tubing is damaged, the female connector usually sustains the damage.

b. The female fitting is more susceptible to damage since the male fitting is more durable and can be dressed.

ILLUSTRATION:



SOLUTION: Incorporate the female fitting on the removable component. The male end can be repaired in place.

3.3.7.6 Damaged Ports

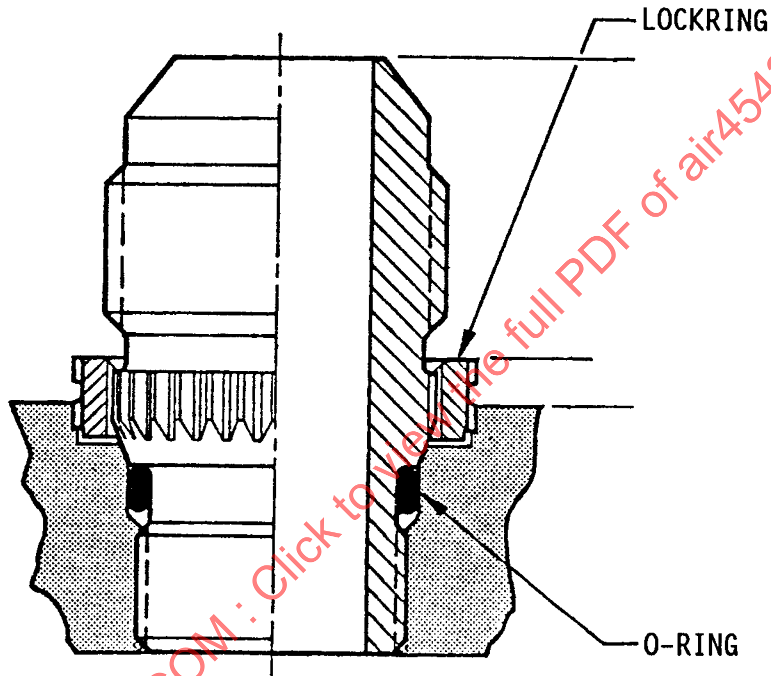
PROBLEM: Damaged ports.

ISSUE:

a. A component with a damaged port can be salvaged if removable boss fittings are employed.

b. When the port of a component is suddenly impacted (dropped, struck) and damaged, the component is discarded if the port is not replaceable.

ILLUSTRATION:



SOLUTION: Removable boss fittings should be employed whenever possible.

3.3.7.7 Teflon Covered Tube Clamps

PROBLEM: Teflon covered tube clamps.

ISSUE:

a. Subject to burnishing, chafing, and uneven wear leading to pin hole or large wear hole in tube.

b. Teflon cover is only 0.020 in (0.5 mm) thick. Tube preload, oil, and dirt between Teflon cover and tube coupled with aircraft vibration quickly wears Teflon allowing the clamp metal to abrade the tube causing a loss of hydraulic fluid.

SOLUTION:

a. Add thick additional Teflon or plastic grommet between clamp and tube to provide longer wear.

b. Replace the loop clamp with a two bolt plastic clamp block.

c. Change the clamp to one that has a cover of Nitrile butadiene rubber, 65-75 durometer with a minimum thickness of 0.060 in (1.5 mm).

3.3.7.8 Hose Assembly Tolerances

PROBLEM: Non-standard (tight) tolerances on flexible hose assembly lengths.

ISSUE: Hose assembly lengths are being specified with very tight overall length tolerances. The reason is reportedly to have a more consistent installation or to keep from exceeding the hose minimum bend radius criteria.

> EXAMPLE: Hose assembly length is specified as 48.000 in ± .125 in (1.22 m ± 0.003 m). Industry Standards would permit a length of 48.000 in ± .500 in (1.22 m ± 0.012 m).

Hose assembly specifications allow hose elongation or contraction while under operating pressure to typically vary ±2% [± 0.96 in (± 24 mm)for this example]. (Ref: AS1339, AS604 and AS4623)

Special fixturing and processing results in unnecessary cost resulting in higher prices.

SOLUTION: Design installations that allow for the use of standard industry hose assembly tolerances by incorporating sufficient hose length.

Industry Standards for hose assembly tolerances vary slightly between specifications (Ref: AS620, AS1339, AS604, AS4623), however, Committee SAE G-3D is currently considering using the following in the G-3D Policy/Guideline Manual:

± 0.125 in (± 3 mm) for lengths under 18 in (460 mm)

± 0.250 in (± 6 mm) for lengths from 18 to 36 in (460 to 910 mm) exclusive

± 0.500 in (± 13 mm) for lengths from 36 to 50 in (910 to 1270 mm) exclusive

±1% for lengths of 50 in (1270 mm) and over

3.3.7.9 Short Hose Assemblies

PROBLEM: Hose assemblies with short hose flex lengths may be inadequate to function properly having the desired flexibility traditionally expected of a hose. This is especially true with the higher pressure wire-reinforced hose styles which are made of a heavier wire construction.

ISSUE: Hose assembly designs with short flex lengths (free hose between sockets) may not perform as expected. Problems that can develop include:

a. Hose stiffness can make installations difficult due to the force required to bend the hose.

b. For lower pressure hose assemblies with less reinforcement, kinking is possible.

c. Loads are transmitted directly into the individual components which must be able to withstand the added stresses.

SOLUTION: Design installations that allow for the use of sufficient hose length and standard industry hose assembly tolerances.

Determine (or acquire) the force to bend characteristics of the specific hose to be used. Consider these forces in the component design.

3.3.7.10 Hose Drawing Tolerances

PROBLEM: Drawings are being supplied with nominal values describing the hose assembly design requirements. Page tolerances will then apply to the nominal values which are not always desirable.

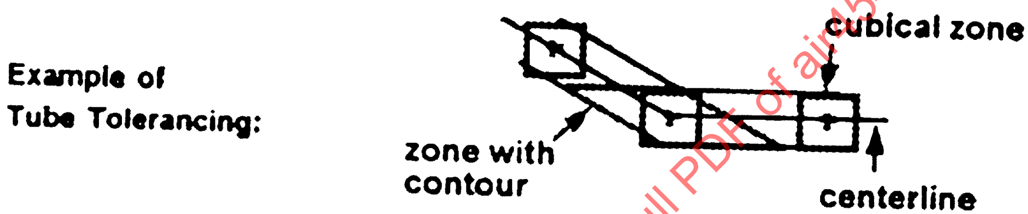
ISSUE: Engineering drawings are being supplied using nominal lengths for the hose assembly lengths and nominal coordinates for the multi-bend elbow descriptions. The default page tolerances defined near the title block will then apply to these nominal values. This is not always the intent of the designer and can result in un-manufacturable designs requiring exceptions to be taken or drawing revisions to be made.

EXAMPLE OF TYPICAL PAGE TOLERANCES

Decimals	0.X in (X mm)	± 0.1 in (± 2 mm)
	0.XX in (X.X mm)	± 0.02 in (± 0.5 mm)
	0.XXX in (X.XX mm)	± 0.010 in (± 0.2 mm)
Angles	± 0 deg 30 min	

Page tolerances on elbow coordinates will result in a cubical tolerance zone for the coordinate point at the bend radius apex. Controlling this theoretical intersection of two tubes is not necessary as is the desire to control the tube contour.

ILLUSTRATION:



SOLUTION: For hose assembly lengths, specify the length tolerance in accordance with the appropriate SAE specification (i.e., AS620, AS604, AS1339, etc.).

For dimensions of other design characteristics, ensure the page tolerance which will apply to all nontoleranced dimensions, is desirable. For single bend elbows, typical tolerances on the elbow drop dimension is ± 0.035 in (± 0.9 mm).

For multi-bend elbows (which often have charted coordinates), specify the coordinates as BASIC dimensions and call out a contour tolerance for the bent tube region. Typical tolerances for tube contours are ± 0.060 in (± 1.5 mm)

3.3.7.11 Hose Installation Practices

PROBLEM: Hose assembly installation practices.

ISSUE: Hose assemblies must be routed and installed properly to obtain the expected full service of the product.

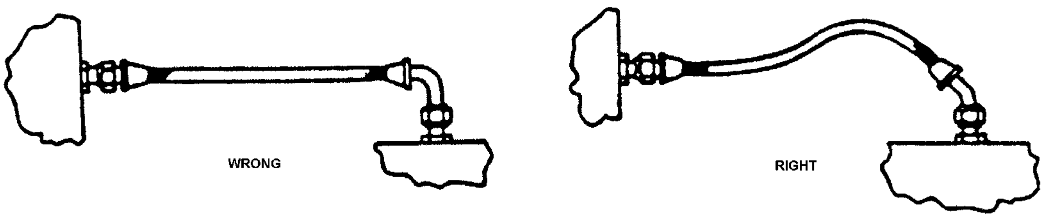
a. Proper length including at least one hose bend should be used to accommodate worst case of tolerance stack up.

b. Proper routing to withstand motions/deflections.

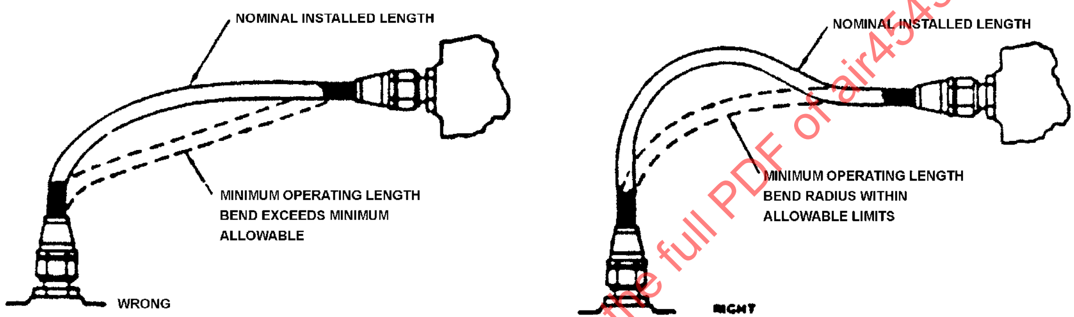
c. Proper installation to eliminate twisting.

d. Proper use of elbows to provide most direct routing and minimize opportunity for abrasion.

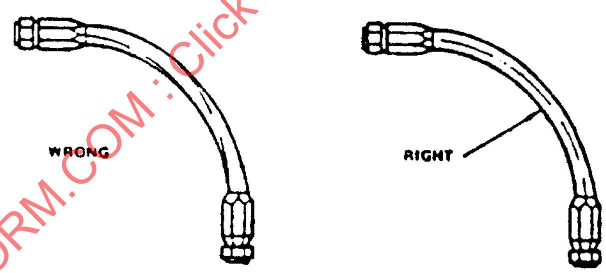
ILLUSTRATION:



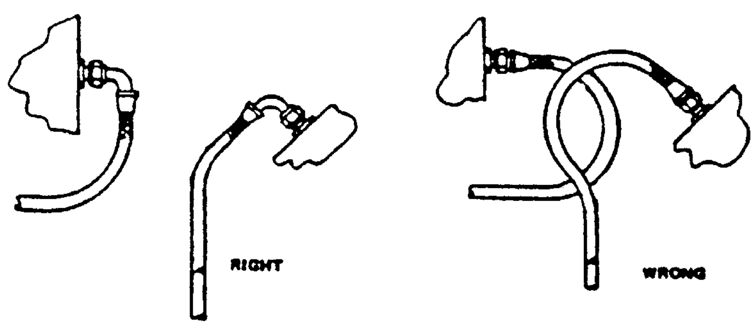
Proper length including at least one hose bend



Proper routing to withstand motions/deflections



Proper installation to eliminate twisting



Proper use of elbows to provide most direct routing

SOLUTION: Design installations and specify hose assemblies that have sufficient hose length with bends to allow for proper function. Refer to AIR1569, "Handling and Installation Practice for Aerospace Hose Assemblies" for additional information.

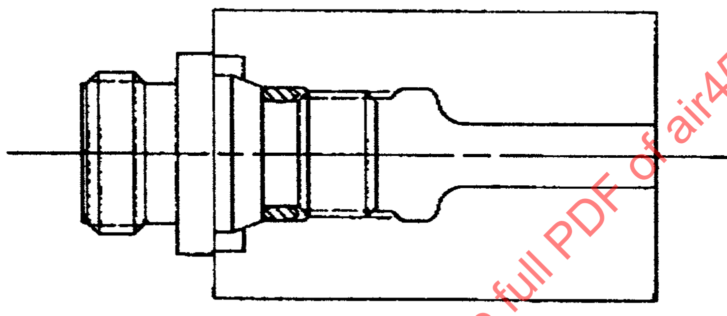
3.3.7.12 Dry Boss Port Threads, #1

PROBLEM: Boss fatigue failure.

ISSUE: "Wet" thread configuration caused short failure life 700,000 cycles.

ILLUSTRATION:

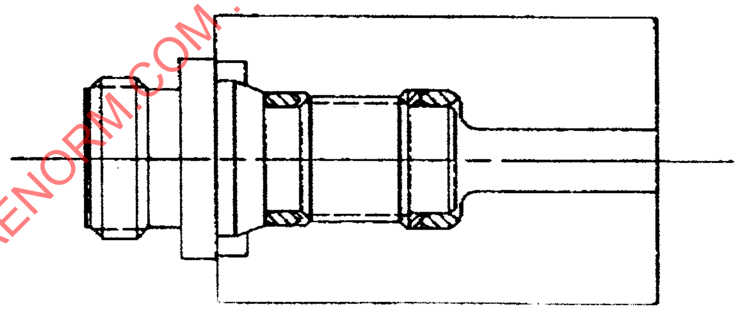
Standard -6 Rosan Port



SOLUTION: "Dry" thread increased fatigue life to 2.8 million cycles by incorporation of an O-ring and back-up at bottom of thread.

ILLUSTRATION:

Proposed Modified -6 Rosan Port

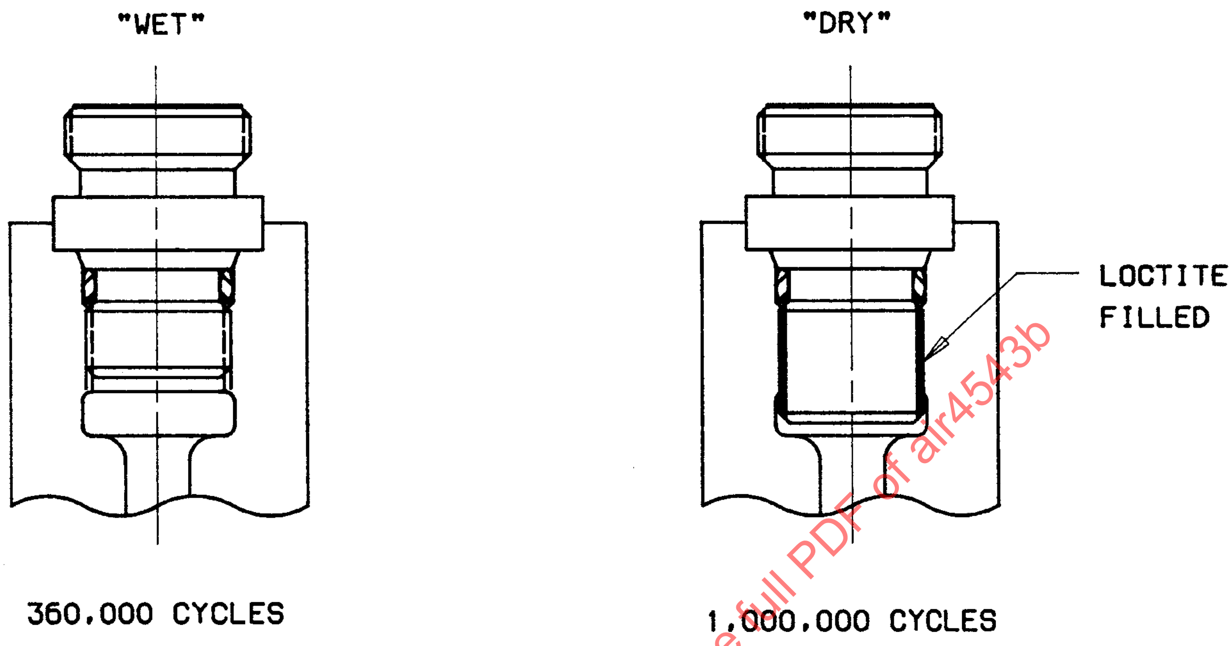


3.3.7.13 Dry Boss Port Threads, #2

PROBLEM: Low port fatigue life with aluminum 7075-T73.

ISSUE: "Wet" threads reduce life at 7500 lbf/in^2 (51 700 kPa), 200 °F (93 °C).

ILLUSTRATION:



NOTE: Sample Sizes of Six

SOLUTION: Use "dry" threads with plug threads extended beyond port threads.

3.3.7.14 Crossing Hydraulic Connections

PROBLEM: Crossing hydraulic connections.

ISSUE: Hydraulic pressure, return, or cylinder lines are sometimes crossed during component replacement, causing manifold failures, internal component binding or damage to line replacement units such as electrohydraulic servovalves. The problem of crossed lines can also exist with brake hoses where a locked brake can result.

SOLUTION: Critical flight control actuator or manifolds with combined functions should be designed to prevent reversing or crossing of the lines during installation. One way is to use different size line connections for pressure and return.

Where the same line size is required, it is recommended that the fitting connections not be in the same plane, in other words, 90 or 180° out of phase. However, sometimes due to installation space available the connections have to be in the same plane. If this is the case they should be at a different connection level and be tested for possible crossing.

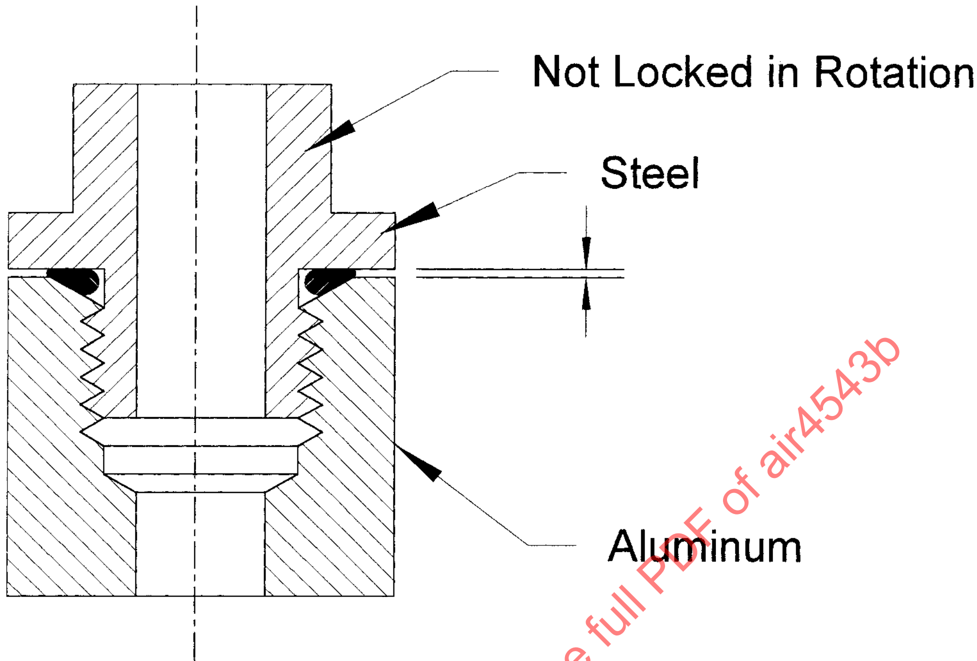
When hoses are used like in the brake systems, one should prevent cross connections by the way they are routed or preferably by using a different hose size for pressure.

3.3.7.15 Quick Disconnects Loosening

PROBLEM: Quick disconnects (steel) of the MS boss type loosen in service when installed in aluminum manifolds due to pressure impulse.

ISSUE: MS boss type quick disconnects were backing out of aluminum bosses. No positive lock for rotation other than torque was incorporated.

ILLUSTRATION:



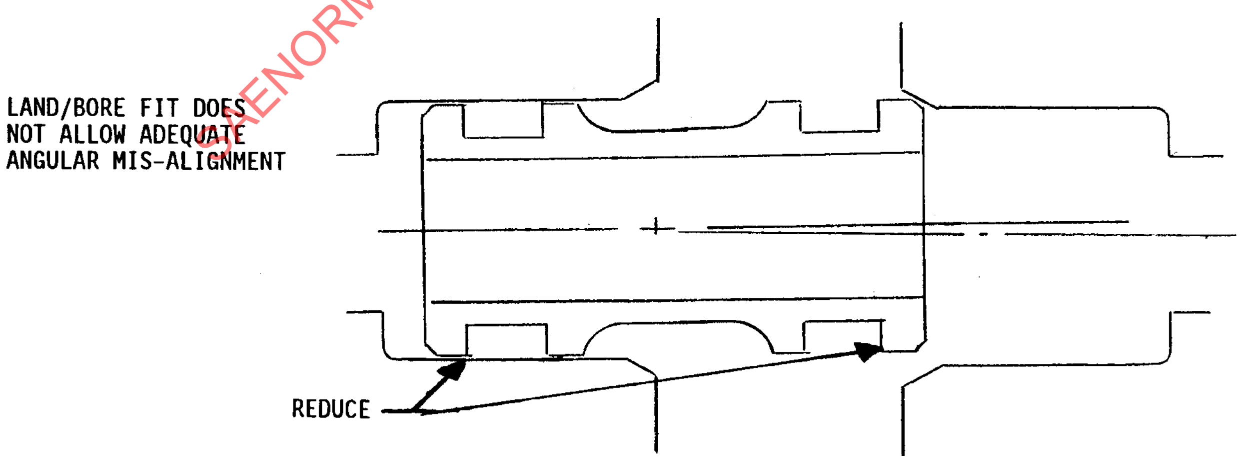
SOLUTION: The use of Loctite on the quick disconnect to boss threads has eliminated this loosening and backing out.

3.3.7.16 Binding of Fluid Transfer Quills

PROBLEM: Binding of fluid transfer quills.

ISSUE: Quills do not accept housing misalignment.

ILLUSTRATION:



SOLUTION: Reduce the diameter of the seal lands furthest from the interface to provide angular play.

3.3.7.17 Component Malfunction and Vent Blockage

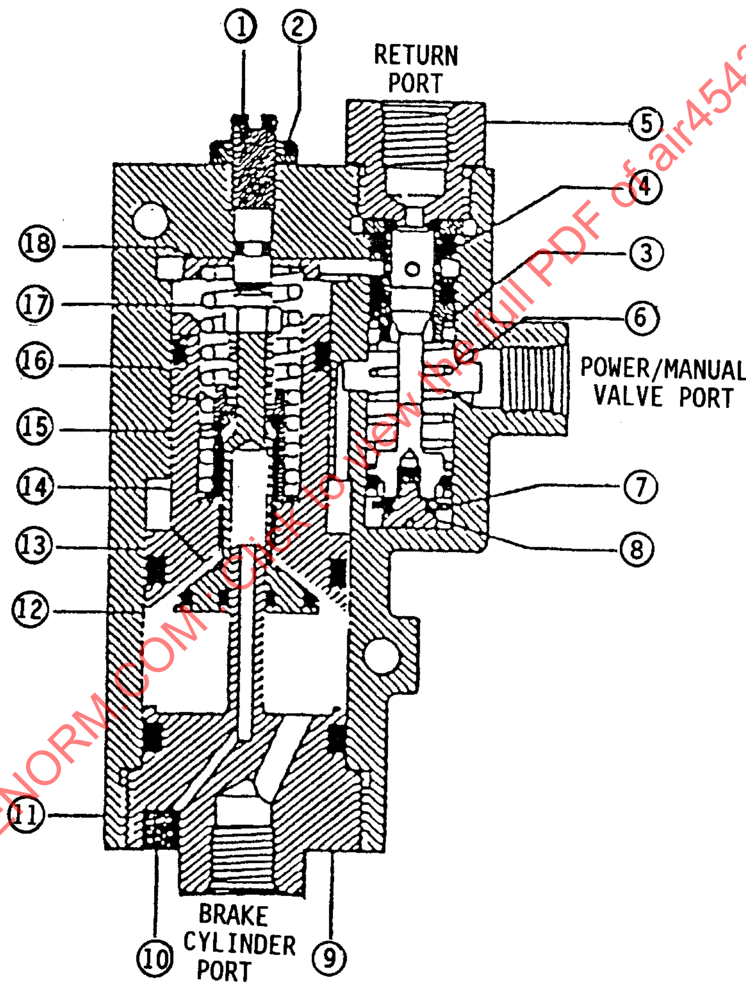
PROBLEM: Component malfunction due to vent blockage.

ISSUE:

a. Many hydraulic components have areas requiring atmospheric vents protected by filter screens.

b. Some screens have been totally clogged by improper painting.

ILLUSTRATION:



SOLUTION:

a. Do not paint vent holes.

b. Insure that the design aids vent hole recognition.

3.3.8 Valves

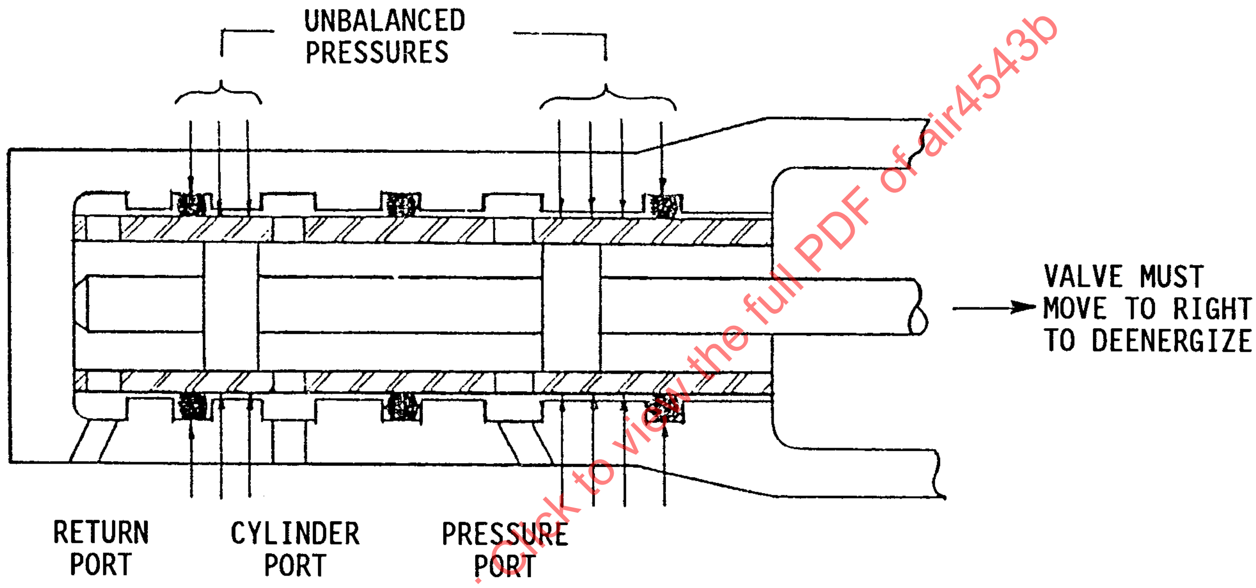
This section applies to hydraulic valves, but excludes electrically operated valves which are covered in section 3.4.

3.3.8.1 Unbalanced Sleeve Pressures and Valve Jams

PROBLEM: Unbalanced pressures cause sleeve to pinch valve slider.

ISSUE: Many valves have hung up or failed to operate because a high pressure on the outside of a sleeve was not balanced by an equal pressure inside the sleeve. In the valve shown, the controller would not return to the de-energized position.

ILLUSTRATION:



SOLUTION:

a. Attempt to balance the pressures inside and outside of the sleeve so that there will be no reduction in sleeve diameter.

b. When complete balancing is not possible, increase the clearance between the sleeve and slider. Include an acceptance test to demonstrate valve operation at a pressure higher than that to be experienced in service.

c. Incorporate at least three pressure balance grooves in each controller land.

3.3.8.2 High Spool Valve Friction at Elevated Temperature

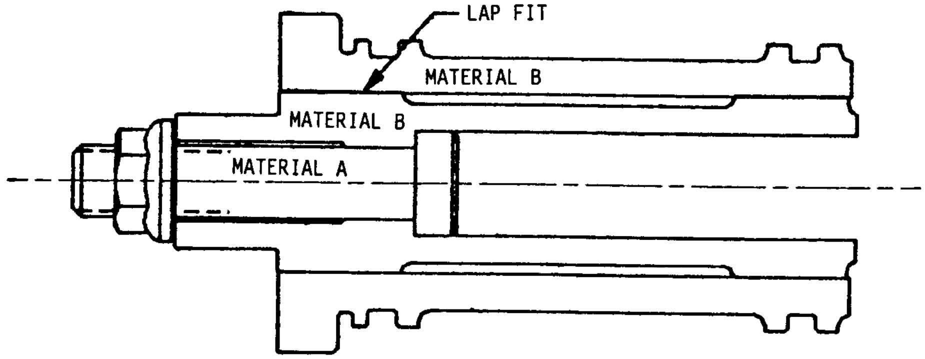
PROBLEM: Lap-fitted assembly high friction at elevated temperature.

ISSUE:

a. Spool is clamped/loaded with different materials of A and B.

b. Lap fitted diametral clearance changes due to axial load increasing from differential thermal expansion axially.

ILLUSTRATION:



SOLUTION: Increase lap clearance during manufacturing.

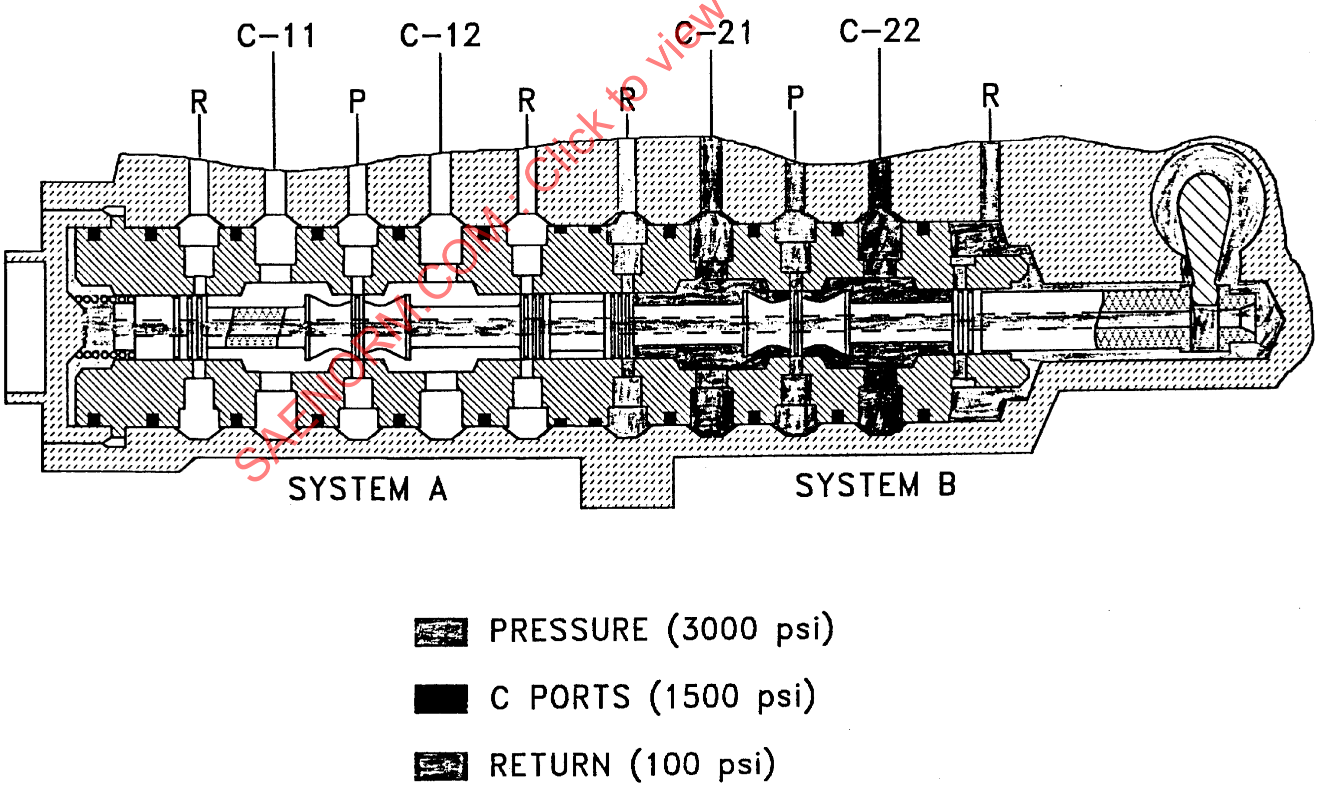
3.3.8.3 Rotary Input Servovalve Oscillations

PROBLEM: Rotary input servovalves.

ISSUE: Pressure impulses in valve return act on valve spool and cause valve oscillations.

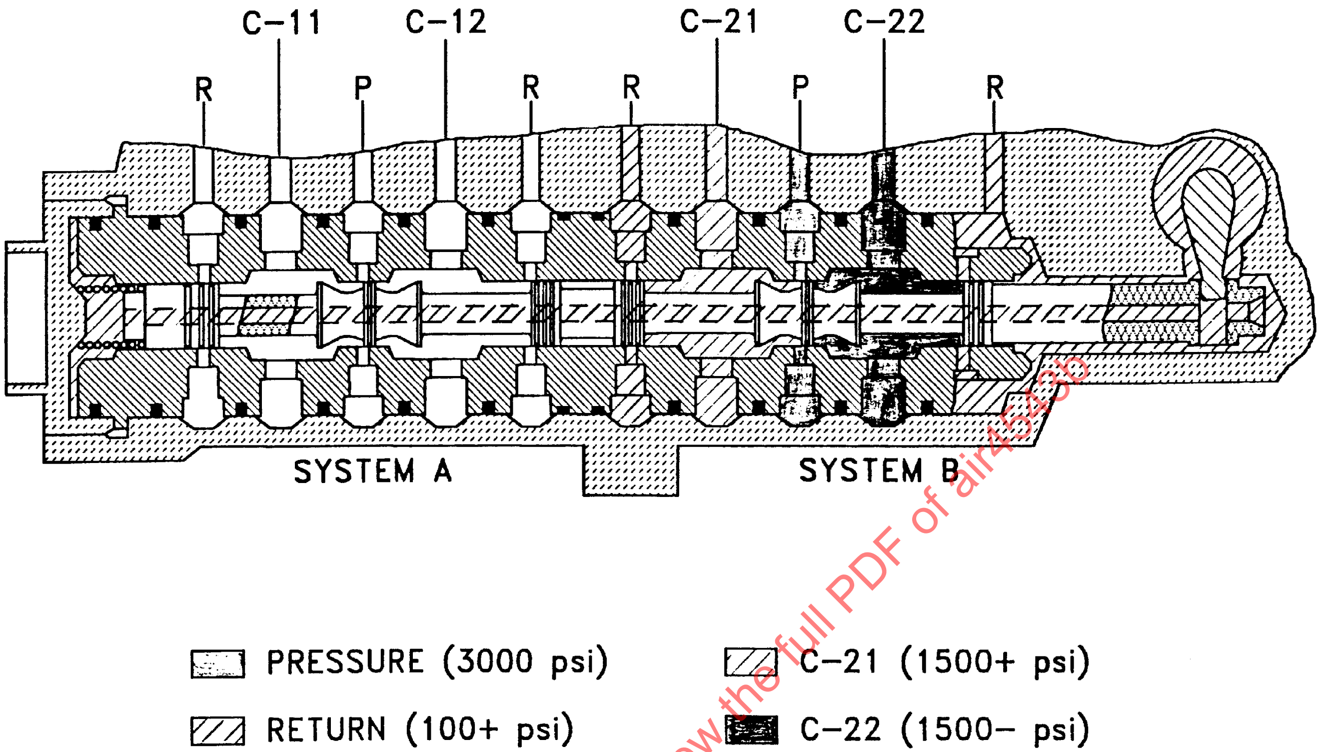
SOLUTION: Mask ends of spool from pressure impulses.

ILLUSTRATION 1 – CURRENT VALVE DESIGN



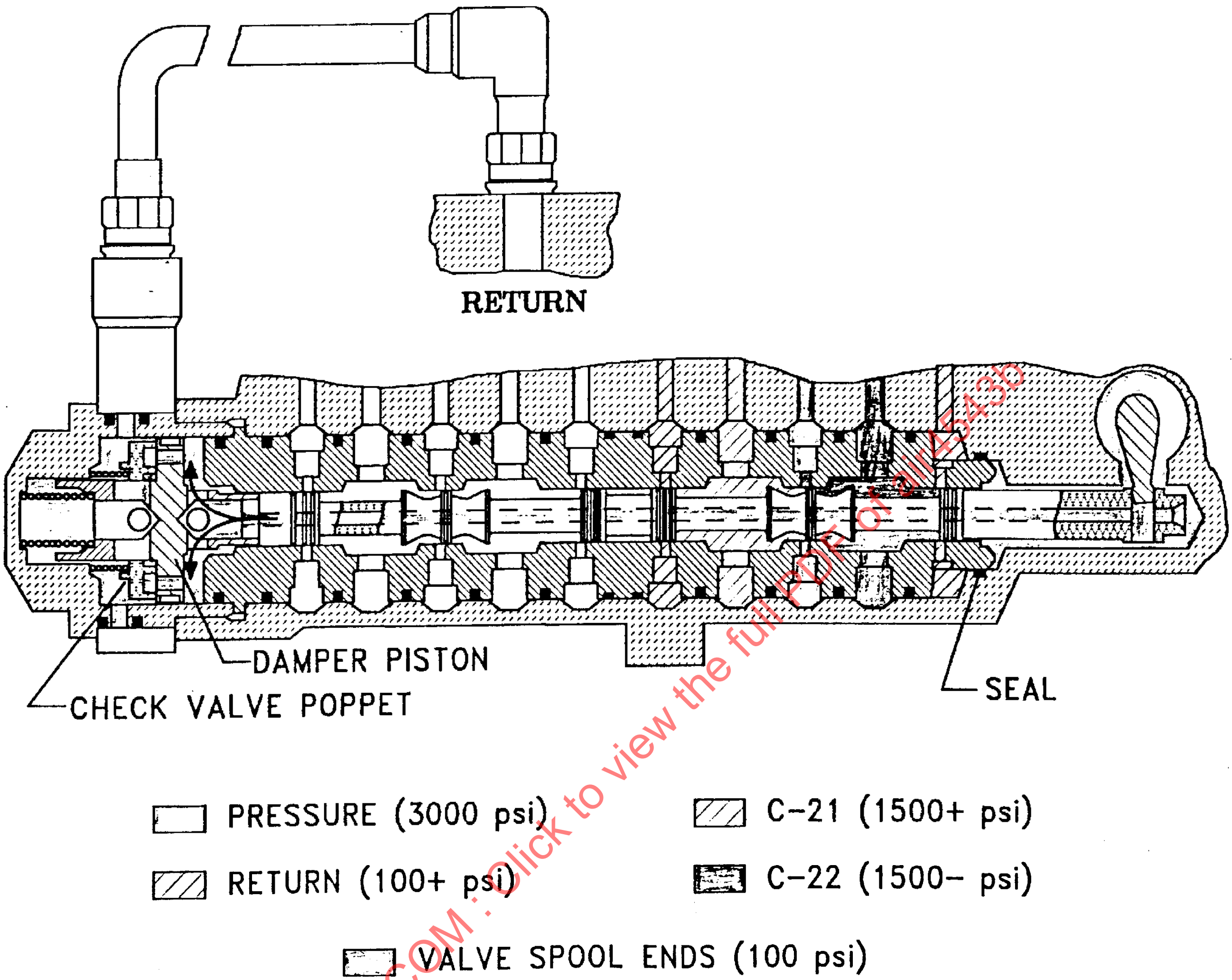
NOTE: The pressures of 3000, 1500 and 100 lbf/in^2 referenced in Illustration 1 are equivalent to pressures of 20 700, 10 300 and 690 kPa, respectively.

ILLUSTRATION 2 - RETURN IMPULSE SOURCE



NOTE: The pressures of 3000, 1500 and 100 lbf/in^2 referenced in Illustration 2 are equivalent to pressures of 20 700, 10 300 and 690 kPa, respectively.

ILLUSTRATION 3 - PROPOSED VALVE REDESIGN



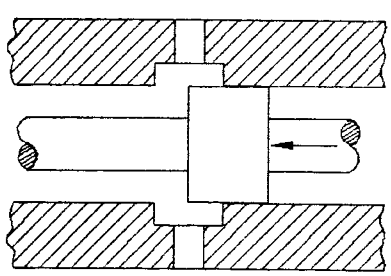
NOTE: The pressures of 3000, 1500 and 100 lbf/in^2 referenced in Illustration 3 are equivalent to pressures of 20 700, 10 300 and 690 kPa, respectively.

3.3.8.4 Valve Spool Jamming

PROBLEM: Valve spool jamming.

ISSUE: Spool land having to cross by an undercut in the valve sleeve.

ILLUSTRATION:

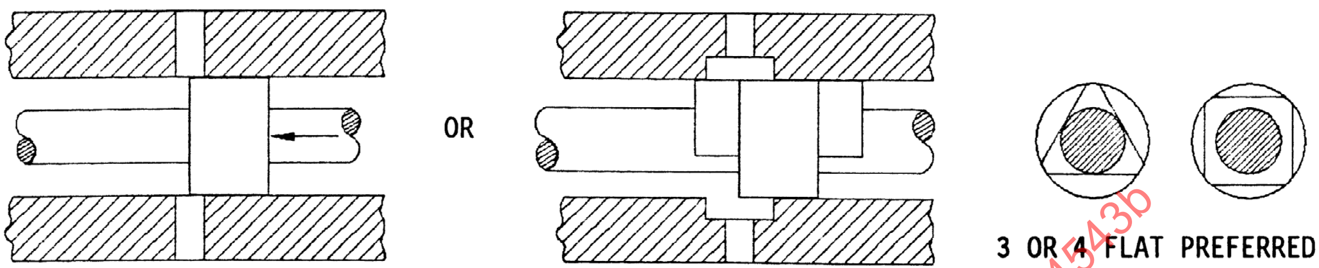


SOLUTION:

a. Eliminate undercut - use holes only.

b. Provide a guide on the spool such as 2, 3, or 4 flats.

ILLUSTRATION:

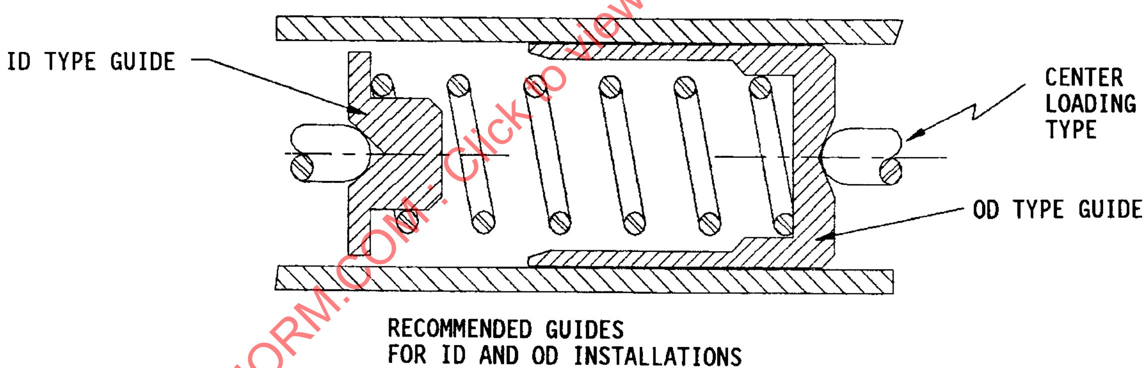


3.3.8.5 Spring Friction in Valves

PROBLEM: Spring friction in valves.

ISSUE: Inadequate or improper guide.

ILLUSTRATION:



SOLUTION:

a. Provide guide, with chamfers, at both ends of the spring (ID or OD) for 1-1/2 to 2 coils.

b. Increase the clearance between the guides and bore diameter.

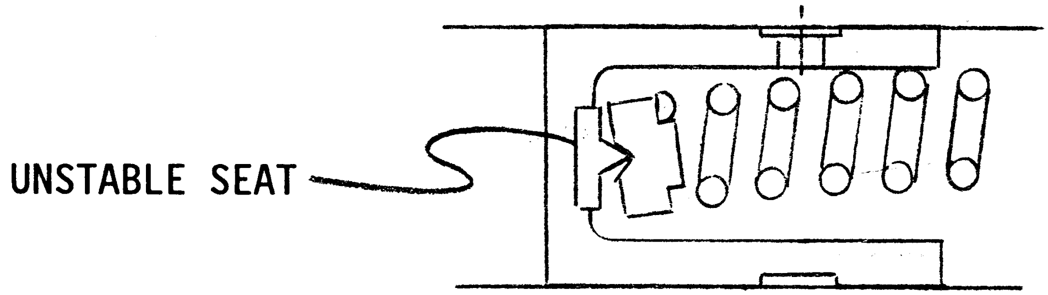
c. Where possible, eliminate the side loading by centering the spring load.

3.3.8.6 Hysteresis in the Valve

PROBLEM: Hysteresis in the valve.

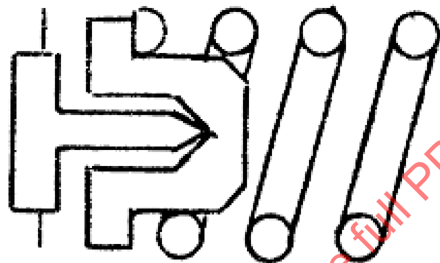
ISSUE: Spring pivots, located outside of the spring, cause the spring to buckle and bear against the valve spool.

ILLUSTRATION:



SOLUTION: Locate pivot within the spring.

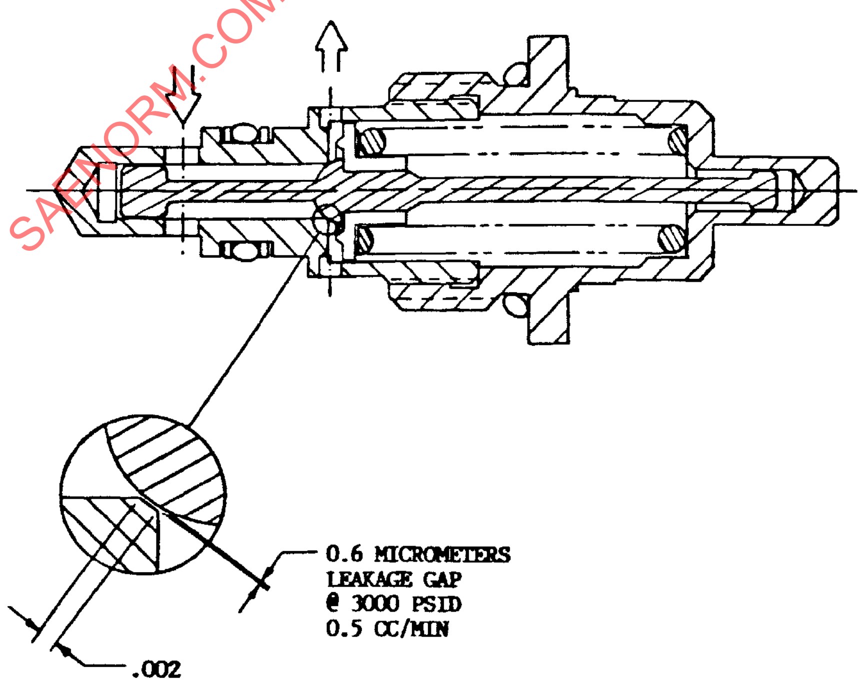
ILLUSTRATION:



3.3.8.7 Leakage of High Pressure Relief Valves

PROBLEM: Poppet style metal-to-metal relief valves for high pressure hydraulic systems (3000 lbf/in^2 (20 700 kPa) and up) demonstrate erratic shutoff perfromance for leakages less than 1 cc/min. High reject levels (10 to 20%) occur between the valve manufacturer and the system contractor, resulting in costly documentation, retest and often, Not-As-Stated reports.

ILLUSTRATION:



NOTE: The pressure of 3000 lbf/in^2 referenced in the Illustration is equivalent to a pressure of 20 700 kPa.

ISSUE: Relief valve leakage is usually measured on decreasing pressure tests at slightly below the cracking pressure. Net poppet force, hence seat stress, is low since the upstream pressure force is only slightly less than the spring force. Low seat stress means microscopic voids from seat imperfections and fluid particles.

COMMON CAUSES OF INTERMITTENT LEAKAGE:

- Poor Seat Preparation
- Random Particle Trapping
- Silt Damming
- High Particle Counts
- Low Particle Counts
- Low Fluid Viscosity
- Pump Ripple
- First Run Contamination
- Test Sequence and Method

SOLUTION: Assure that the valve leakage specification is realistic. Consistent shutoff to 1 or 2 drops per minute can be achieved with ideal conditions but not in real hydraulic environments. Elastomeric sealing may be required for repeatable near zero leakage.

Specify an upper and lower limit to particle counts. The lower limit assures adequate silt for shutoff of microscopic imperfections of the seat and poppet. Typical limits for aircraft component testing are AS4059 (or NAS 1638) (now inactive for new design) Class 2 to 6.

Assure test conditions are the same at the valve manufacturer and system contractor, and that they accurately simulate the valve's end use.

Run blow-down repeat cycles to test for seat imperfections versus fluid contamination. Start-up contamination often dissipates after a few cycles.

3.3.8.8 Balance Grooves Became Unbalance Grooves

PROBLEM: Stiction of spool caused erratic motion.

ISSUE: A 0.125 in (3.2 mm) diameter spool in a pilot valve with balancing grooves was binding and exhibiting erratic motion. Balancing grooves geometry revealed variation in groove width around spool circumference. When pressurized, force imbalance would push spool to one side of bore and cause jamming or erratic movement of spool.

SOLUTION: Use flat sided ground grooves versus V-grooves. Do not chamfer grooves after grinding to remove burrs.

3.3.8.9 Binding of Slider in Shrink Fit Valve Sleeve

PROBLEM: Temperature changes and thermal gradient effects on slide valves with shrink fit sleeves.

ISSUE: In order to minimize valve length, shrink fit sleeves are sometimes employed in hydraulic components. If the valves are built with minimal diametrical clearance between sleeve and slider there is a danger that hot oil being ported into a cold valve/housing assembly may cause the slider to bind in the sleeve due to the radial thermal gradient from the slider, through the sleeve, and the housing.

SOLUTION: Ensure that the clearance between sleeve and slider is sufficient to accommodate any reduction in that clearance resulting from the anticipated thermal gradient.

3.3.8.10 Isolation Valves and Pressure Transients

PROBLEM: Excessive pressure transient can result when opening an isolation valve to an unpressurized downstream volume.

ISSUE: When fast-acting isolation valves admit high pressure to an unpressurized downstream volume, an excessive pressure transient can result. The pressure transient is reflected, and sometimes amplified, throughout the system. In order to save cost and increase reliability (of the isolation valve), isolation valve opening rates are not normally damped unless there is a specific requirement to do so. When the isolation valve is initially specified, high-fidelity system dynamic analyses are generally not available to validate and quantify the need for isolation valve damping.