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AEROSPACE INFORMATION REPORT

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Aerospace Hydraulics and Actuation Lessons Learned

TABLE OF CONTENTS

1. SCOPE:

This SAE Aerospace Information Report (AIR) contains technical information on aerospace hydraulic, actuation and support equipment technologies lessons learned. The lessons learned were prepared by system designers and hydraulic 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 at eight Lessons Learned Symposia from 1989 through 1999 held during A-6 Committee meetings. The technical topics represent many years of design experience in hydraulics and actuation, which it is felt is a resource for learning that should be documented and made available to current and future aerospace hydraulic engineers and designers. The document is organized into two sections: systems and components with further categories within the components section. The information topics are presented in a concise format of problem, issue, and solution, with accompanying descriptive diagrams and illustrations for clarity and understanding.

2. REFERENCES:

There are no referenced publications specified herein.

3. TECHNICAL INFORMATION:

3.1 Systems Lessons Learned Topics:

This section contains topics that relate to the overall hydraulic power generation and distribution system, as they affect the total system function and performance. Figures 3.1.1 through 3.1.23.

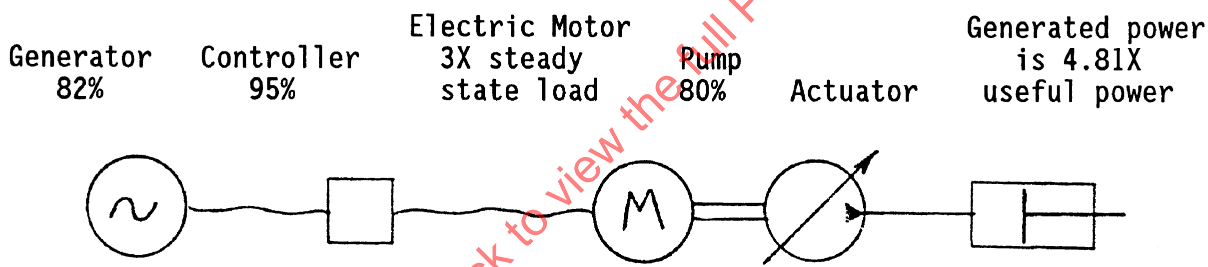
PROBLEM: POWER CONVERSION EFFICIENCY

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

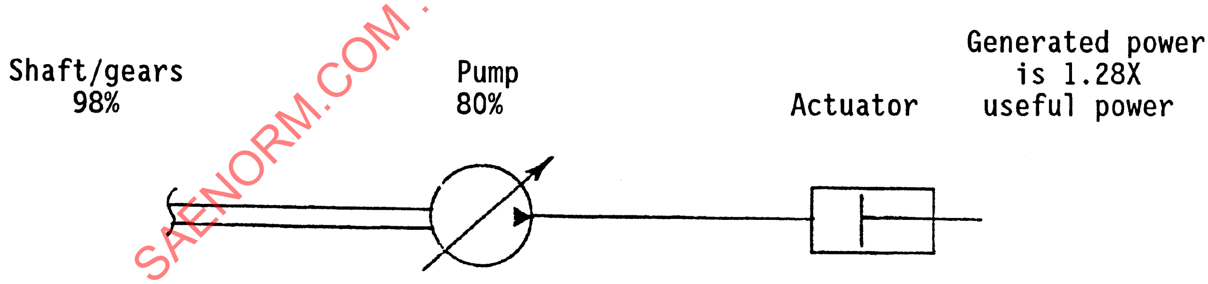
a. mechanical
b. electrical
c. hydraulic
d. pneumatic

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.

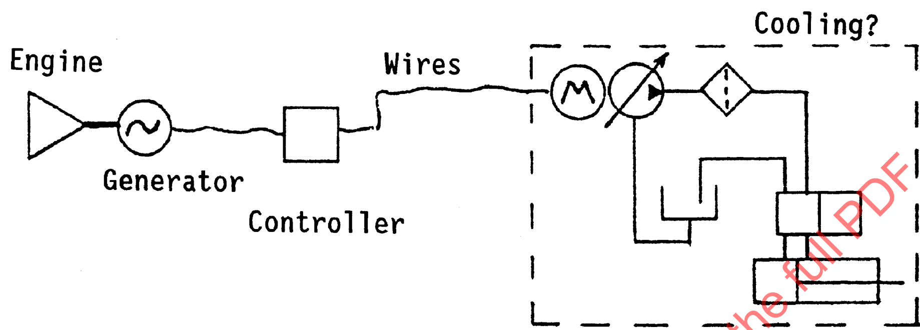
FIGURE 3.1.1 - Power Conversion Efficiency

PROBLEM: BIASED TRADE STUDIES

ISSUE: Trade studies often fail to assess all influences on the air vehicle.

ILLUSTRATION:

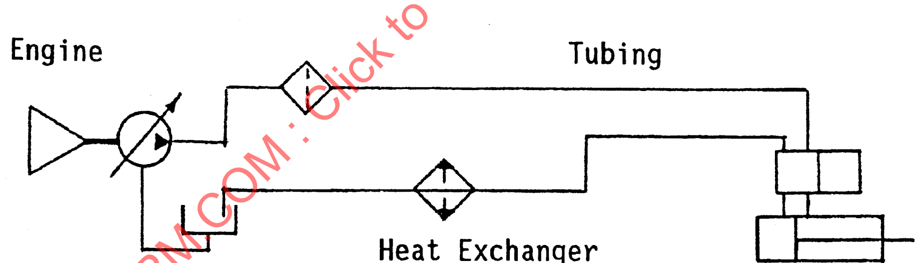
Distributed hydraulic packages



Frequently in such trade studies the following issues are not compared:

Cooling
Power transmission
Power conversion

Centralized hydraulic system



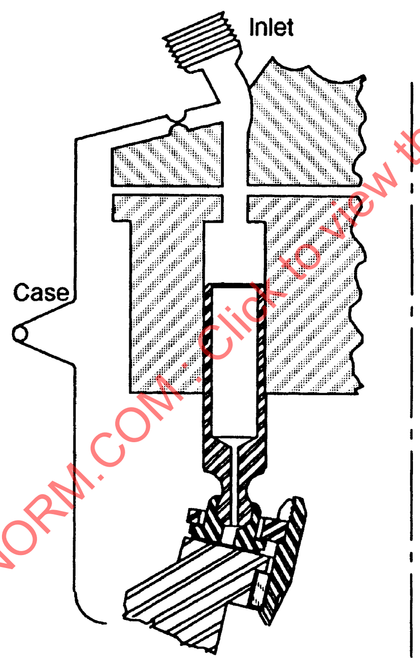
SOLUTION: Conduct the trade study from the point of original power generation (usually the engine) to the point of final utilization.

FIGURE 3.1.2 - Biased Trade Studies

PROBLEM: PUMP OVERHEAT AND RAPID WEAR

ISSUE:
1. Pump case drain line too small
2. Creates high back pressure (case)
3. Creates high discharge pressure (flow)
4. Overheat (pump)
5. Excessive pump shoe wear

ILLUSTRATION:

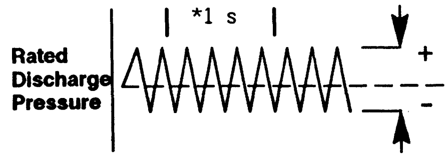


SOLUTION:
1. Increase line size (case drain)
2. Use bypass type filters
3. Reroute line to reduce pressure drop
4. Internally drain (to inlet) if minimum system demand available

FIGURE 3.1.3 - Pump Overheat and Rapid Wear

PROBLEM: EXCESSIVE PUMP PRESSURE RIPPLE IN SYSTEM

ISSUE:
1. Line fatigue
2. Excessive wear (moving parts)
3. Vibration/noise



SOLUTION:
1. Generate dynamic analysis of total system
2. "Tune" pump to match system compliance
3. Provide matched attenuator (pump to system)

Acceptable
±10%

Desirable
±5%

Achievable
±2%

CPS - Cycles Per Second

$$^{*}\left(\frac{\#\ \text{Pistons X rpm}}{60}\right) = \text{CPS}$$

FIGURE 3.1.4 - Excessive Pump Pressure Ripple in System

PROBLEM: PUMP OR MOTOR 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 installation. 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 psi proof pressure or less.

SOLUTION:

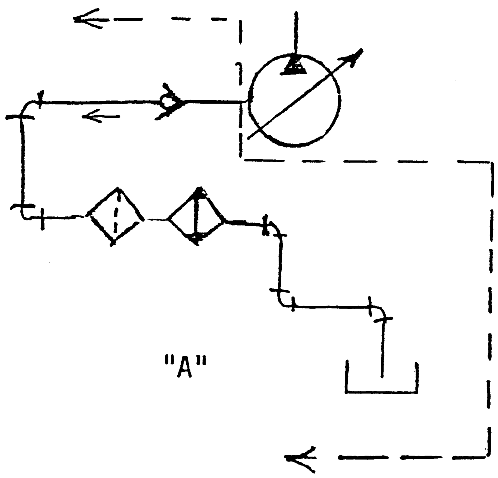
1. Analyze the installation and determine the pressure drop of the pump or motor case drain plumbing from the case to the reservoir. The analysis should include evaluations considering delta pressure of all possible case drain flow and fluid temperature combinations, case drain filter variables, transient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

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

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

FIGURE 3.1.5 - Pump or Motor Case Drain Line Design

Design pressure (pump/motor case) must be higher than any combination of A:



Typical "Case Circuit"

Line Length
Line Size
Bends
Fittings
Check Valve
Filter
Reservoir
Heat Exchanger

A = Pressure Drop Consideration

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

2. 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.

FIGURE 3.1.5 (Continued)

PROBLEM: PUMP CAVITATION

ISSUE:

1. High levels of pressure oscillation have resulted from:
   a. Poor maintenance (excessive air in system)
   b. Pump cavitation due to system malfunction
2. Bracket and hydraulic line failures between the engine driven pumps and the filter module have occurred.

SOLUTION:

1. Install an attenuator in the pump outlet line to reduce oscillations during cavitation.
2. Tune the attenuator to the system configuration.
3. Remove all free air from system.

FIGURE 3.1.6 - Pump Cavitation

PROBLEM: PUMP SUCTION LINE BURSTING

ISSUE:

1. On the initial engine run of a twin engine aircraft the R/H pump suction line burst.
2. The line was replaced and the R/H engine started - no problem.
3. The L/H engine was started and the R/H suction line burst.
4. The line was replaced and the L/H engine started - no problem.
5. The R/H engine was then started and the R/H suction line burst.

SOLUTION:

1. The R/H pump compensator was found to have a slightly lower pressure setting than required.
2. A failed pump outlet check valve created a reverse pumping situation on the R/H side.
3. Pump and check valve were replaced.

FIGURE 3.1.7 - Pump Suction Line Bursting

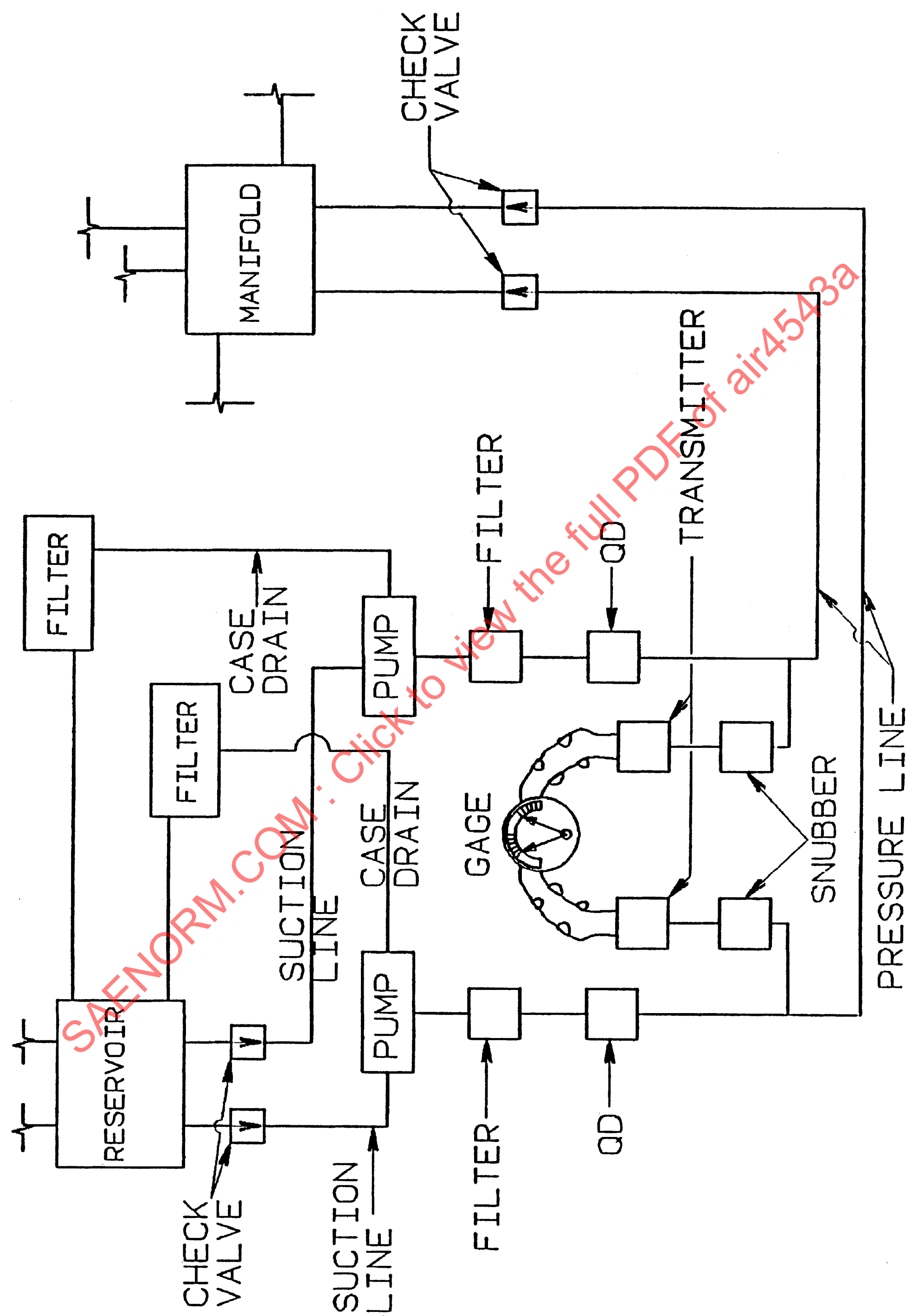


FIGURE 3.1.7 (Continued)

PROBLEM: HYDRAULIC SYSTEM NOISE

ISSUE:

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

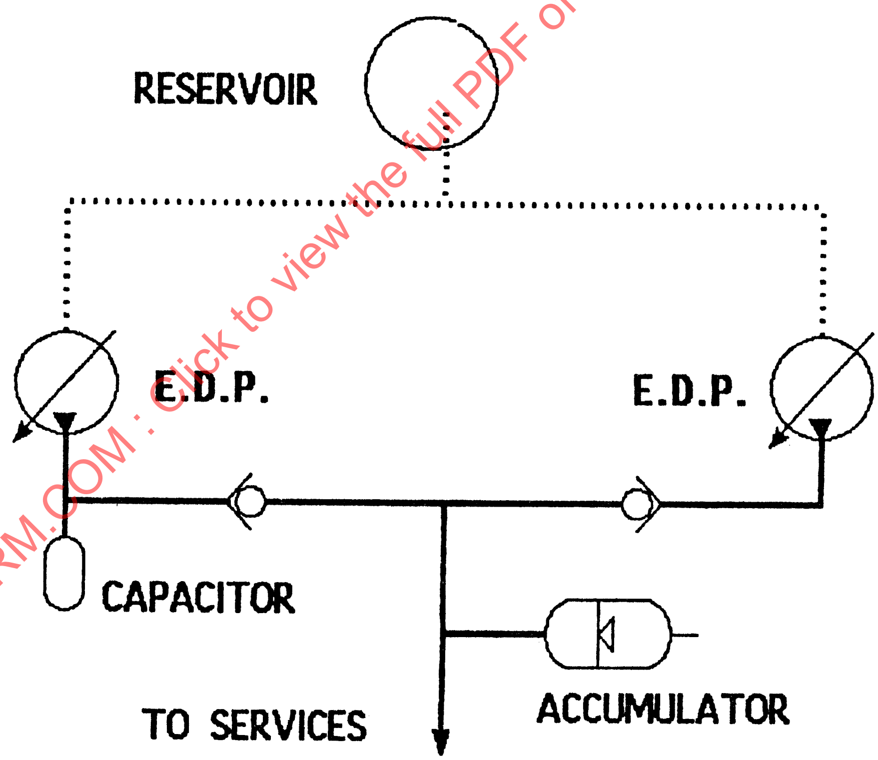
   a. Earlier production aircraft;

   b. When system is pressurized by ground rigs.

2. 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



SOLUTION:

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

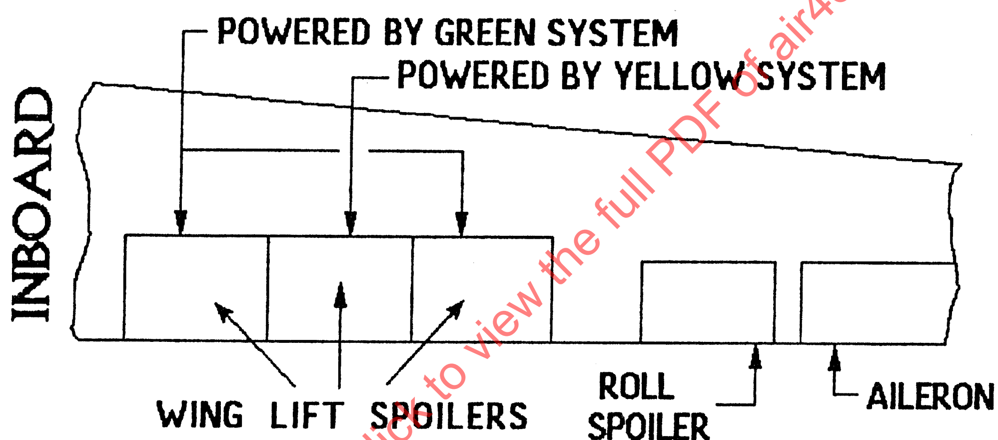
2. Finally, revert to a smaller accumulator and add a "capacitor" (a small pressure vessel (22 in^3) to the system).

FIGURE 3.1.8 - Hydraulic System Noise

PROBLEM: EFFECT OF JAR/FAR REQUIREMENT ON FLIGHT CONTROL SURFACE OPERATION

ISSUE:
1. Noncompliance with JAR/FAR 25.671:
2. A single failure of a flight control surface attachment point could cause a catastrophic event.
3. 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: 1. Reroute hydraulic supplies to the lift spoilers and link the center and outboard panels.

ILLUSTRATION:

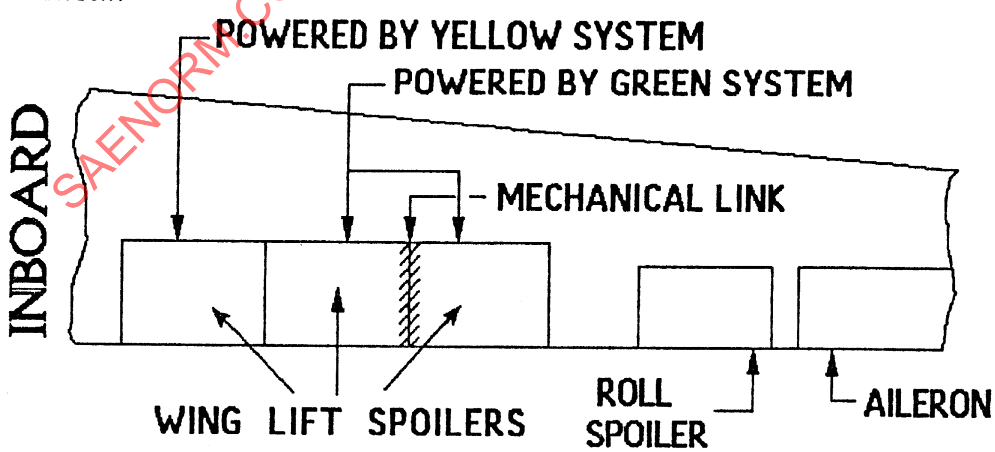


FIGURE 3.1.9 - Effect of JAR/FAR Requirement on Flight Control Surface Operation

PROBLEM: SYSTEM INSTABILITY DUE TO AIR IN PUMP SUCTION LINE

ISSUE: Undissolved air outgassing

SOLUTION:
1. A system design/bleeding procedure
2. A system design for adequate pump inlet pressure

FIGURE 3.1.10 - System Instability Due to Air in Pump Suction Line

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 gpm. The valve cap eroded through in about 35 000 cycles. The area of the valve which eroded was exposed only to 60 psi maximum and a negative 10 psi.

SOLUTION:
1. A relief valve set at 150 psi was put into the return line of the circuit to maintain a positive pressure in the area. This solved the problem.
2. A later metallurgical analysis confirmed the low pressure cavitation.

FIGURE 3.1.11 - Low Pressure Cavitation During Valve Endurance Test

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.)

ISSUE:

1. The bulkhead was replaced and mistakenly fastened with a single row of rivets rather than the required double row.
2. The released cabin air overpressurized the fin, causing failure of the hydraulic lines.

SOLUTION:

1. The access port from the tail section (aft of the bulkhead) was closed.
2. A hydraulic fuse was installed in one of the four hydraulic pressure lines leading to the fin.

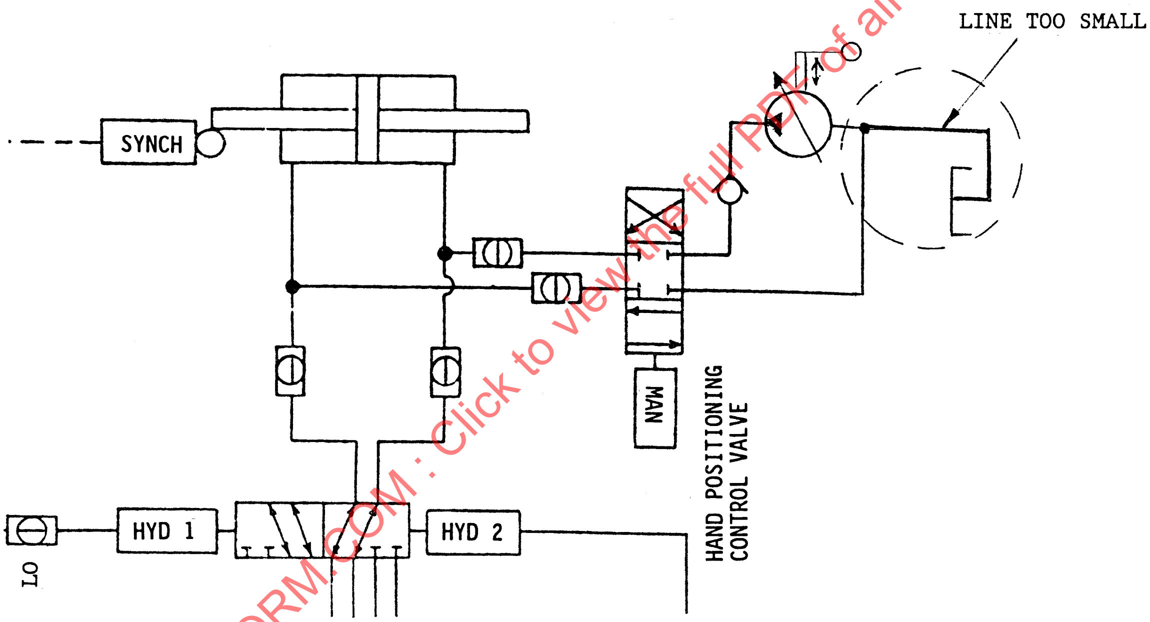
FIGURE 3.1.12 - Failure of All Four Hydraulic Systems Due to Blowout of the Aft Bulkhead. (Normal flight control was lost and the airplane crashed into a mountain.)

PROBLEM: INADEQUATELY SIZED HAND PUMP SUCTION LINE

ISSUE:

1. The hand pump doesn't operate the submarine steering cylinder at the required rate.
2. 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.

FIGURE 3.1.13 - Inadequately Sized Hand Pump Suction Line

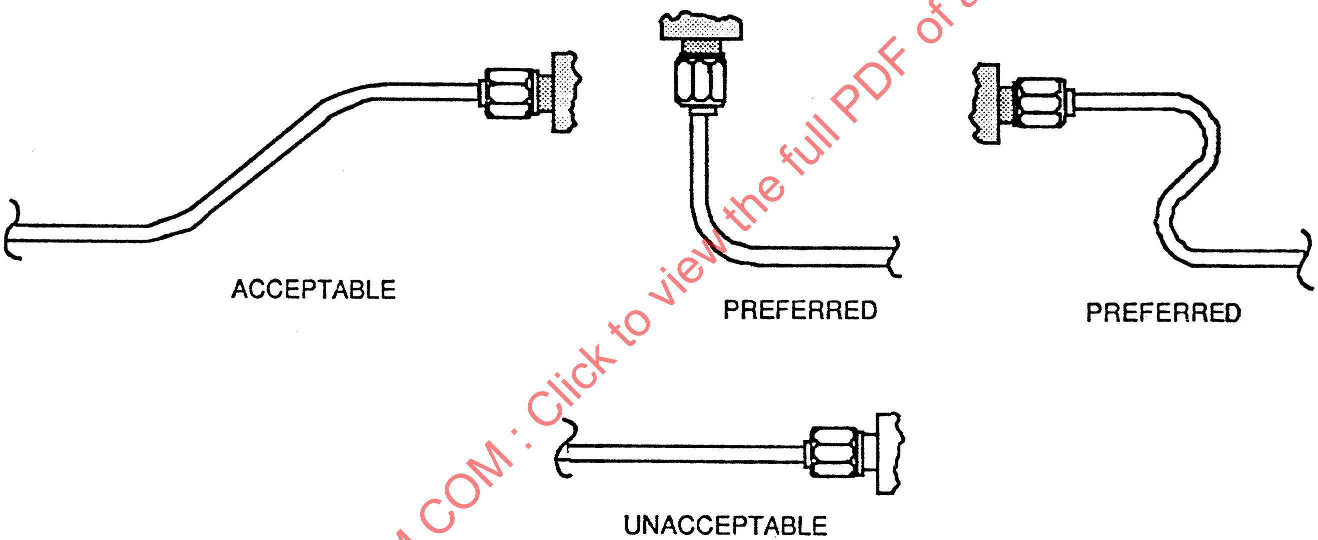
PROBLEM: HYDRAULIC COMPONENT INSTALLATION/TUBING TOLERANCES

ISSUE:

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

2. 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.

FIGURE 3.1.14 - Hydraulic Component Installation/Tubing Tolerances

PROBLEM: LOSS OF HYDRAULIC PRESSURE DUE TO AIR IN THE SYSTEM

ISSUE: Failure to properly bleed the hydraulic systems after servicing has/can lead to the following problems:

a. Loss of systems after negative G maneuvers

b. Failure of pumps to develop pressure on engine start

c. Damage due to pressure spikes/cavitation

d. Excessive drop in reservoir level on engine start

SOLUTION:

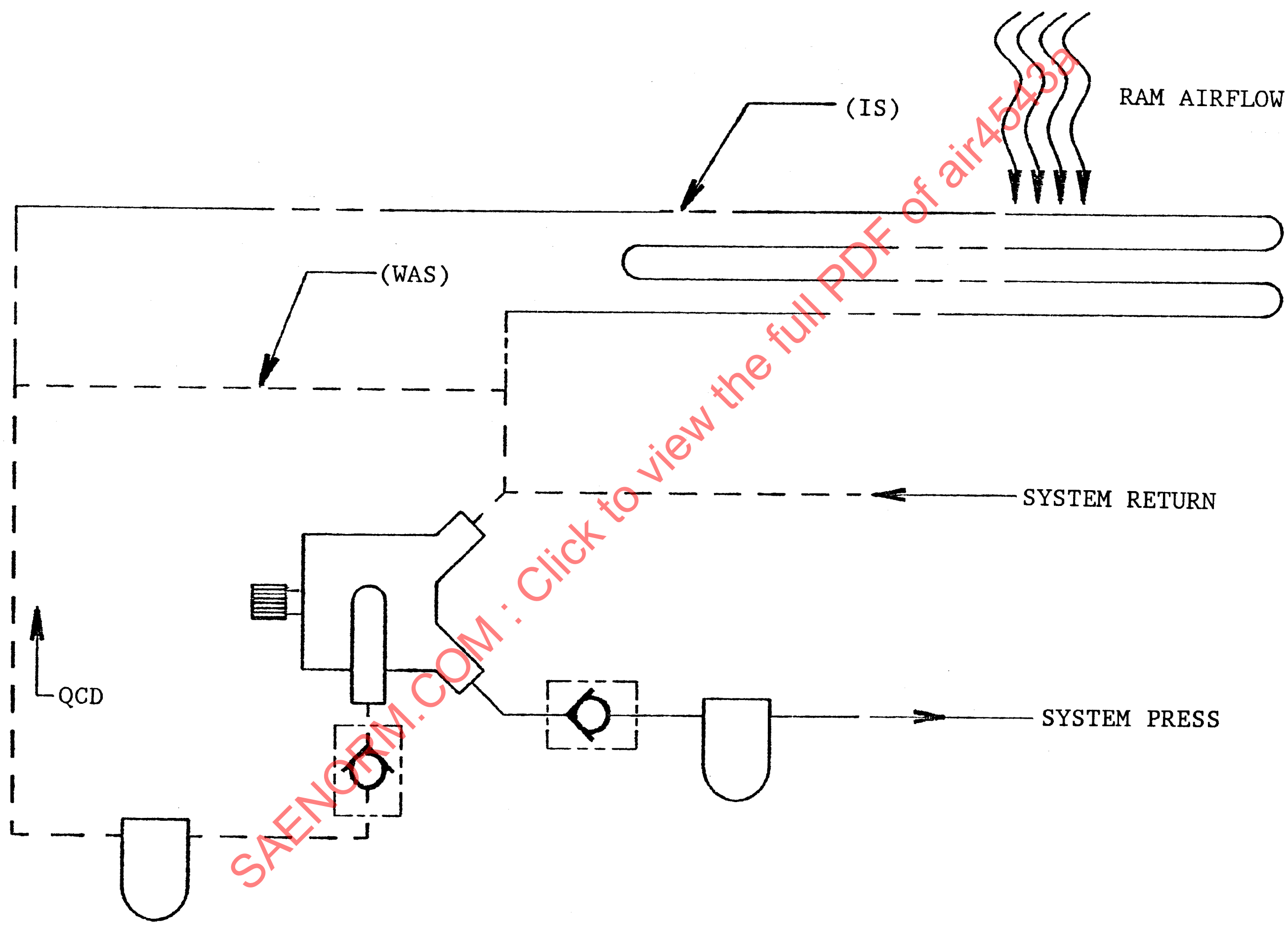
1. Bleed systems with ground cart in open loop mode
2. Schedule regular air sampling checks (20% by volume, maximum)
3. Establish regular reservoir level sink checks
4. Before operating closed loop, bleed cart system/hoses
5. Bleed aircraft that have been down for extended periods
6. Always verify proper ground cart fluid level before connecting to aircraft systems.
7. Consider automatic reservoir bleed valve to avoid reoccurrence.

FIGURE 3.1.15 - Loss of Hydraulic Pressue Due to Air in the System

PROBLEM: SYSTEM OVERHEAT CAUSED BY INEFFICIENT PUMP AND LACK OF HEAT EXCHANGER

ISSUE: System can exceed 275°F

ILLUSTRATION:



SOLUTION:

1. Revise the specifications to buy the pump with guaranteed efficiency and case drain flow over the life of the pump.
2. Increase the case drain line diameter and the filter housing size.
3. Reroute the case drain line to increase surface area and also expose to area with ram air.

FIGURE 3.1.16 - System Overheat Caused by Inefficient Pump and Lack of Heat Exchanger

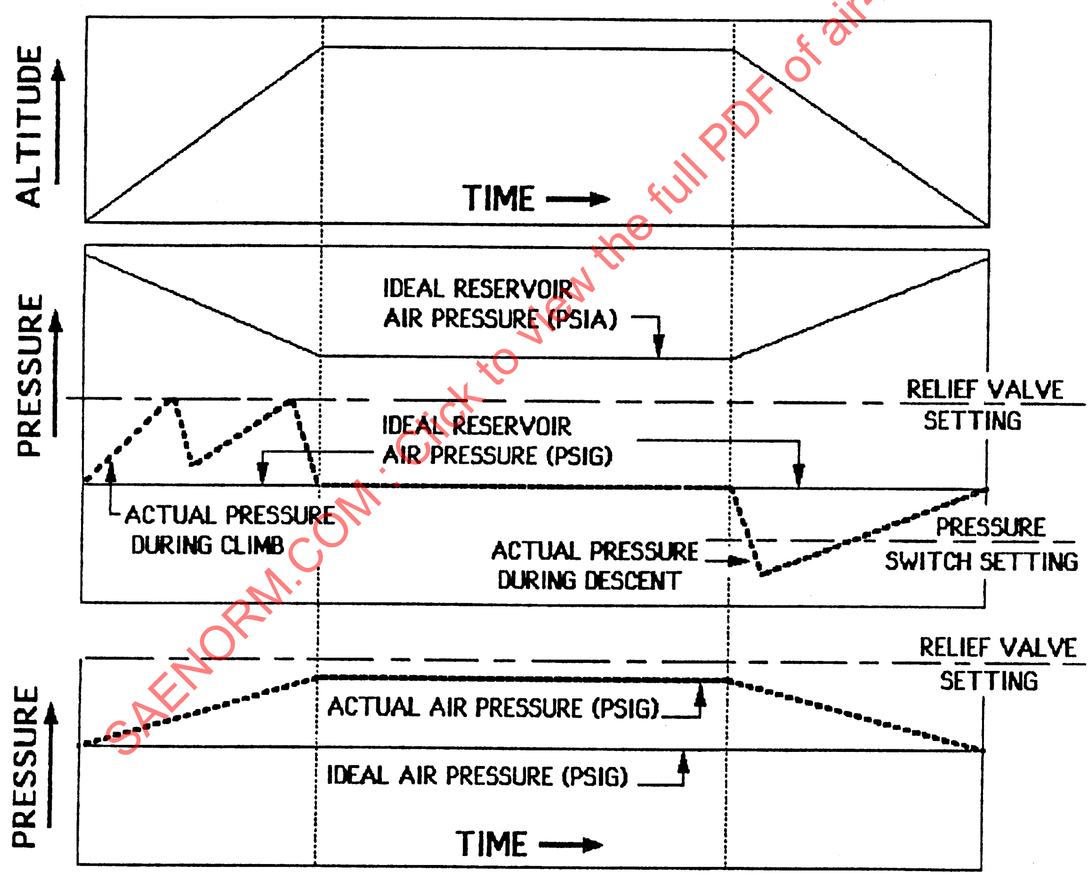
PROBLEM: SPURIOUS HYDRAULIC RESERVOIR LOW AIR PRESSURE WARNING

ISSUE:

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

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

ILLUSTRATION:



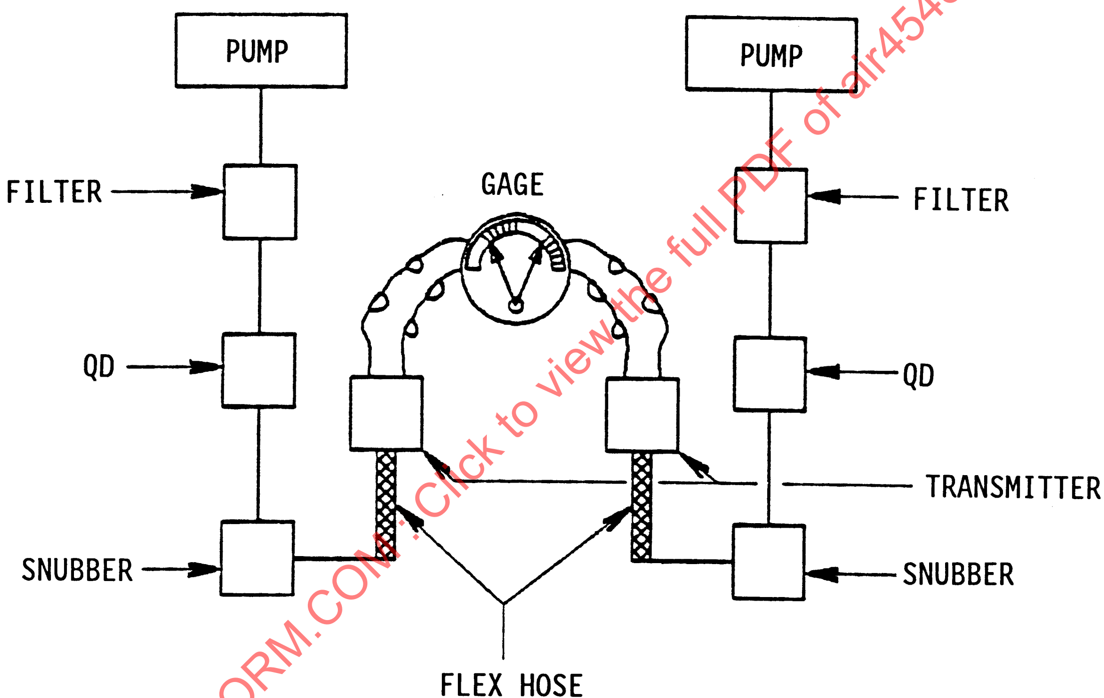
SOLUTION: Either reset the pressure switch to be less than the minimum bleed pressure or set the relief valve so that it does not operate during aircraft climb.

FIGURE 3.1.17 - Spurious Hydraulic Reservoir Low Air Pressure Warning

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.

FIGURE 3.1.18 - False Pressure Transmitter Indication

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 overheat 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.

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.

FIGURE 3.1.19 - Hydraulic System Failures Due to Tire Bursts in the Wheel Well

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.

FIGURE 3.1.20 - Failure of Both Hydraulic Systems Due to Failure of a Landing Gear Link

PROBLEM: 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

FIGURE 3.1.21 - Simultaneous or Out-of-Order Sequencing of System Functions

PROBLEM:

- NEW PARTS BEING MADE TO OLD DESIGNS WILL EITHER NOT ASSEMBLE OR WILL NOT MEET SPECIFICATION REQUIREMENTS DURING TEST.
- DESIGNS MAY HAVE COME FROM THE SAME COMPANY NOW BUILDING THE PARTS OR MAY HAVE BEEN ACQUIRED FROM OTHER COMPANIES OVER THE YEARS.

ISSUE:

- Parts made to drawings that have been procured during an acquisition would not assemble due to interference.
- Parts that have been made assemblable, by correcting the interference conditions, do not provide test results that will meet the specification requirements.
- In order to assemble units, parts must be permanently deformed.

INVESTIGATION RESULTS:

- A detailed tolerance stack up of all the parts indicated many cases of potential interference even in nominal tolerance conditions.
- Metal parts had to be deformed in order to be assembled. Stresses above yield were being encountered causing permanent deformation, especially in light spring members.
- No inventory of Engineering documentation was made at time of purchase.

FIGURE 3.1.22 - Tolerance Stack-Up on Old Drawings

SOLUTION:

- When dealing with old designs, do not assume that a proper tolerance stack up was done on the initial design. If difficulty is found in assembling parts, look to tolerance stack ups in the effected area. The best result comes from doing a complete tolerance stack up of the entire design.

- Prior to building the first units of a design from another company, the Engineering Organization should do a thorough review of the design, including verifying analysis, and tolerance stack ups. If analysis and tolerance stack ups cannot be found, Engineering should selectively reanalyze and retolerance stack parts to assure that the design is one that can be made in the shop of the company that is going to produce it.

- When obtaining a product or design that has been in production at another company, the following items need to be considered:

  1. Was the design artistically or scientifically done?
  2. Is all of the original and follow-up documentation available?
  3. Is all of the required information documented or is some contained in people skills?
  4. Do you need to obtain key people with the product?
  5. A complete inventory of Engineering and Manufacturing documentation is more important than a hardware inventory.

FIGURE 3.1.22 (Continued)

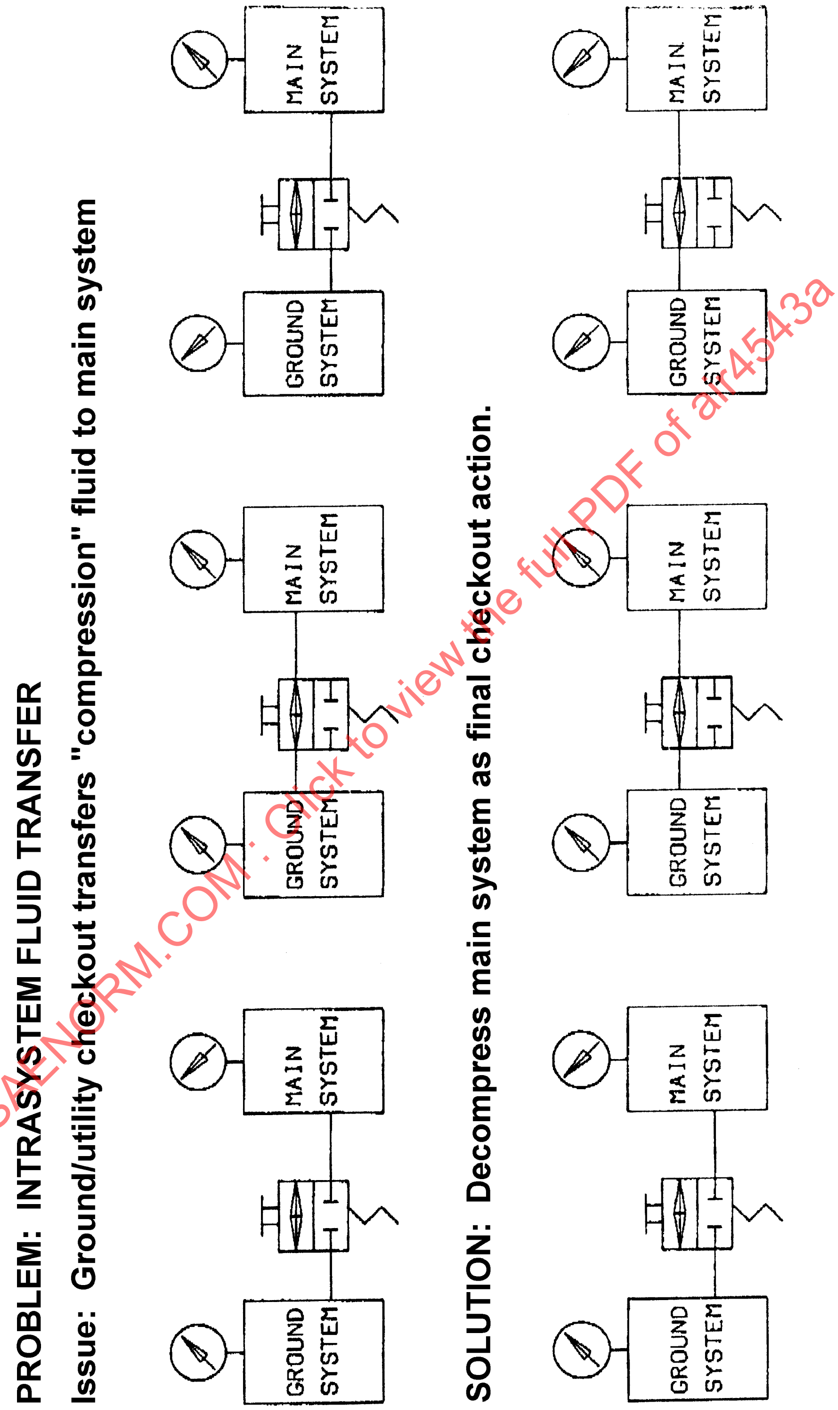


FIGURE 3.1.23 - Intrasystem Fluid Transfer

3.2 Component Lessons Learned Topics:

This section contains hydraulic component and actuation lessons learned topics and is grouped into major component categories including accumulators, servoactuators and cylinders, filters and fluids, corrosion, materials and processes, pumps, seals, fittings and tubing, and valves.

3.2.1 Accumulators: This section contains lessons learned topics relating to hydraulic accumulators. The lessons learned are presented in Figures 3.2.1.1 through 3.2.1.4.

PROBLEM: EXCESSIVE OR PREMATURE LEAKAGE ACROSS ACCUMULATOR SEALS

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

SOLUTION:

1. Ensure that the chrome plate is controlled to minimize cracking (cracks inherent in chrome plating process).
2. Ensure that the seal compound meets all temperature, pressure, and pressure discharge requirements. Minimize the compression set and explosive decompression.
3. Use the closed end cap on gas side (eliminate static seal).
4. Improve the seal concept to improve sealing capability (multiple seals, metal bellows, etc.).

FIGURE 3.2.1.1 - Excessive or Premature Leakage Across Accumulator Seals

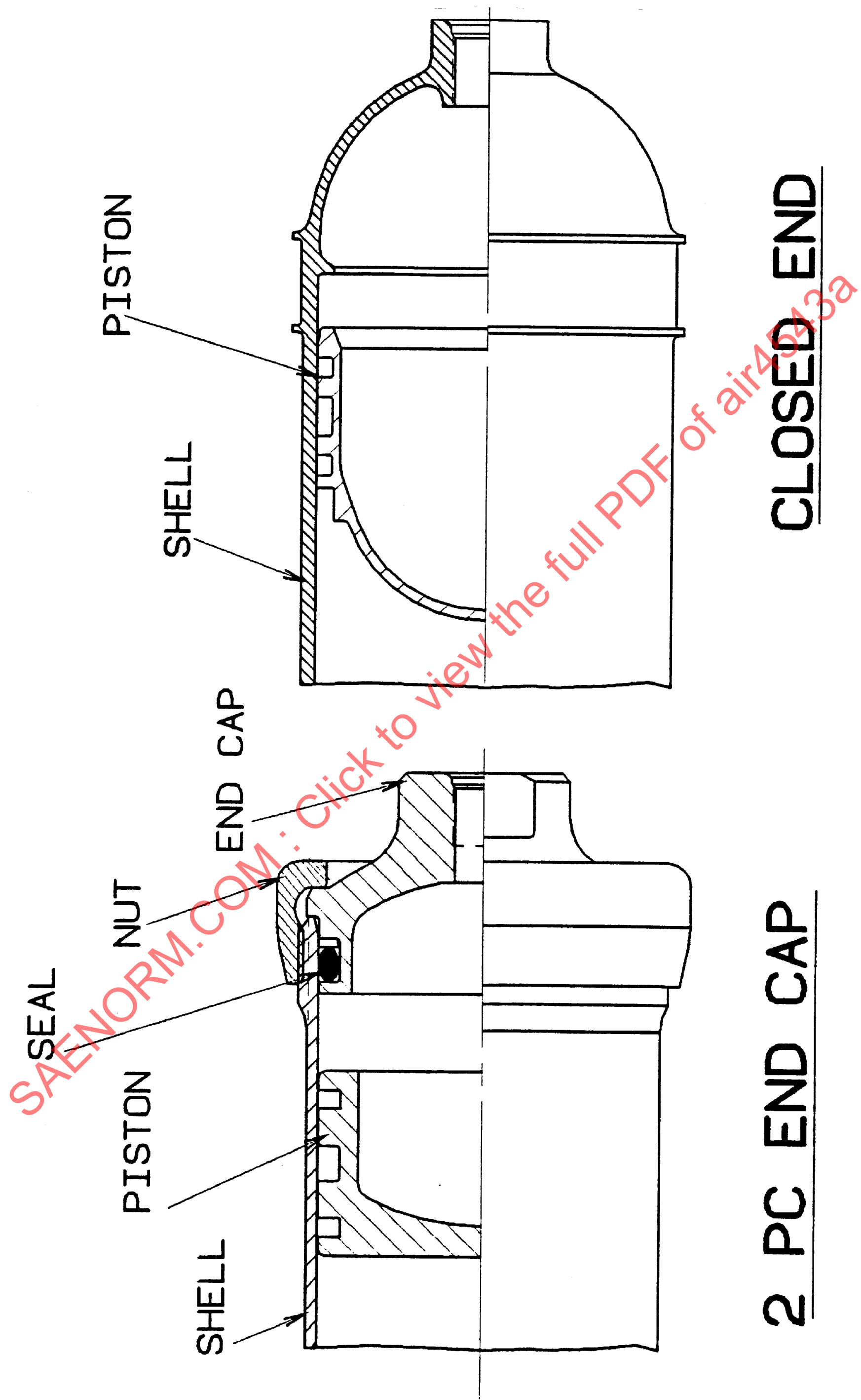


FIGURE 3.2.1.1 (Continued)

PROBLEM: PREMATURE FAILURE OF ACCUMULATOR END CAPS

ISSUE: Failed accumulators are an extreme hazard to aircraft.

SOLUTION:
1. Replace aluminum with steel or equivalent.
2. Ensure that equipment is used at design pressure.
3. Use two piece end cap design to distribute stresses (steel nut and aluminum gland).
4. Use the closed end cap on gas side.
5. Use materials with good fracture toughness - slow crack propogation.
6. Provide adequate corrosion protection.

FIGURE 3.2.1.2 - Premature Failure of Accumulator End Caps

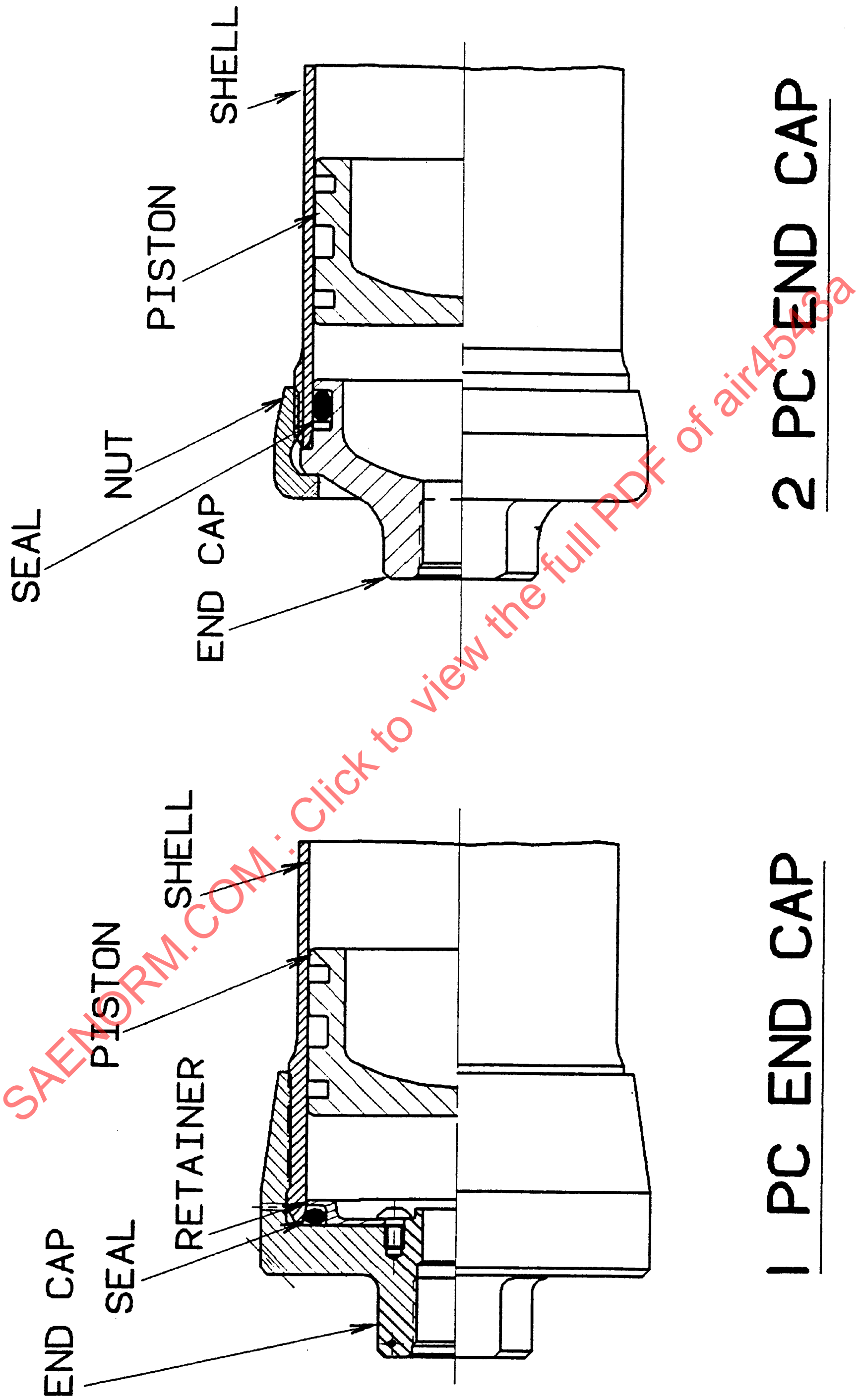


FIGURE 3.2.1.2 (Continued)

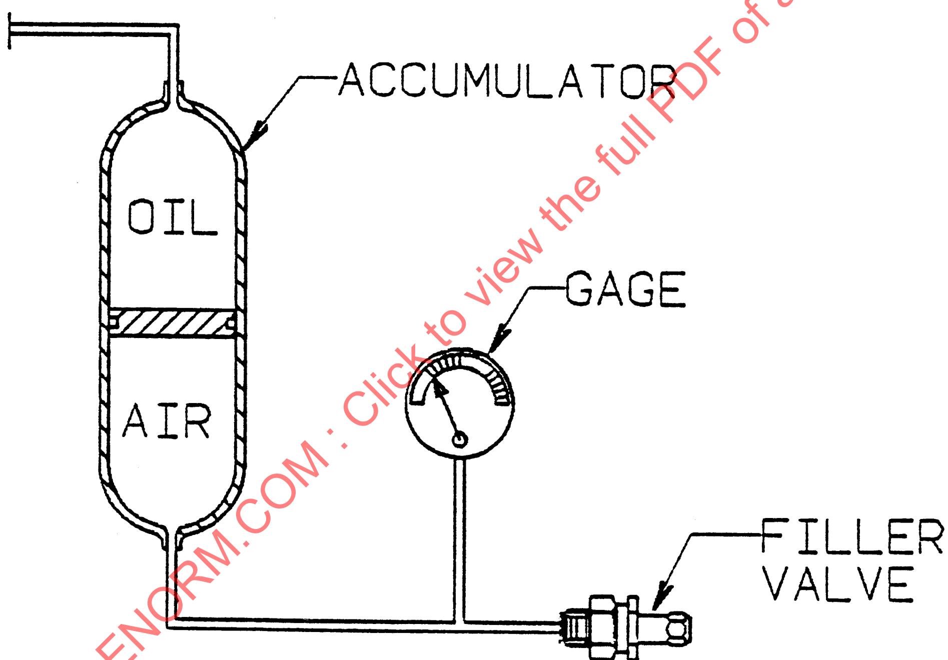
PROBLEM: FALSE ACCUMULATOR GAGE READING

ISSUE:

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

2. The fill valve had been relocated after initial sizing. The accumulator piston bottomed before reaching 3000 psi.

ILLUSTRATION:



SOLUTION: The accumulator precharge was recalculated and adjusted to the correct value.

FIGURE 3.2.1.3 - False Accumulator Gage Reading

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.

- 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.

- 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.

- Failed accumulators were standard MIL-A-5498 type.

REVIEW:

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

- 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.

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

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

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

FIGURE 3.2.1.4 - Hydraulic Accumulator Safety

SOLUTION:

- A minimum in-service inspection program is recommended.

  AIR4150 procedure for in-service inspection written.

- On existing aircraft the installation to be evaluated for:

  - Effect of environment on accumulator

  - Safety of surrounding structure, services and personnel

- 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

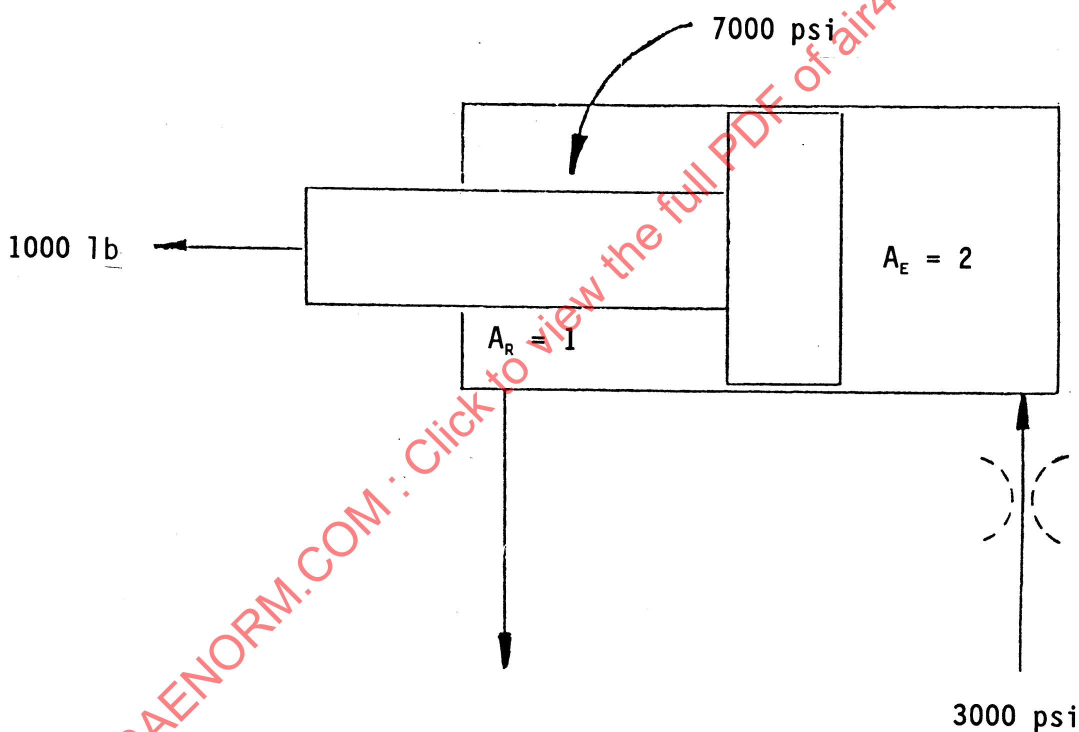
FIGURE 3.2.1.4 (Continued)

3.2.2 Servoactuators and Cylinders: This section contains lessons learned topics relating to hydraulic servoactuators and cylinders. The lessons learned are presented in Figures 3.2.2.1 through 3.2.2.25.

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:



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

FIGURE 3.2.2.1 - Pressure Intensification in Extending Cylinder

PROBLEM: SERVOVALVE SEPARATION FROM THE MANIFOLD DUE TO SERVOACTUATOR VELOCITY LIMITING

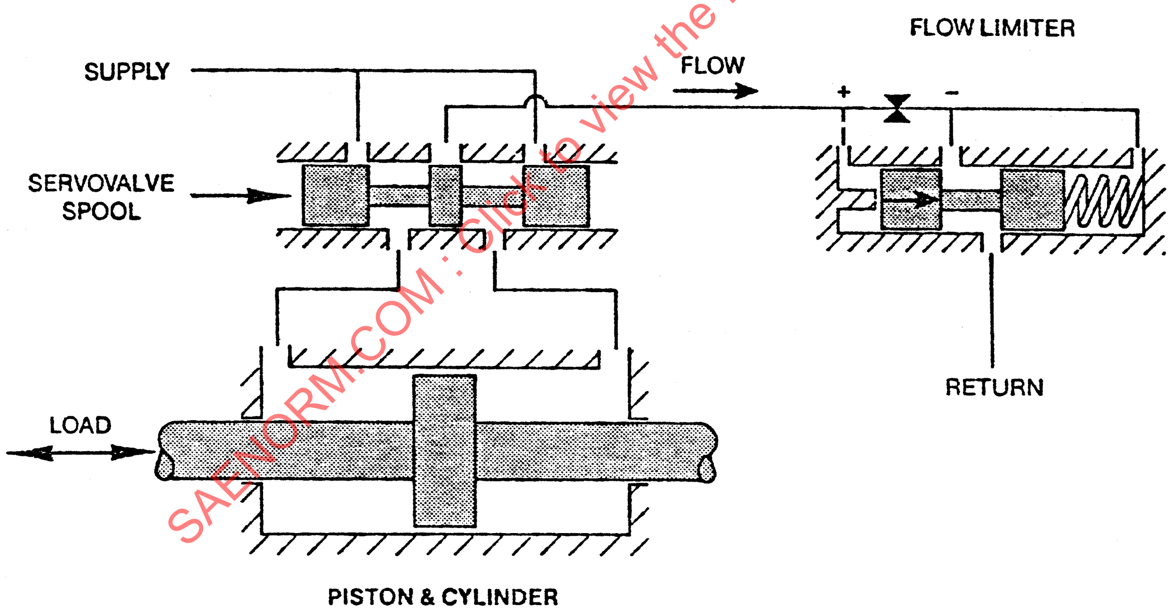
ISSUE:

1. 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.

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

3. 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:

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

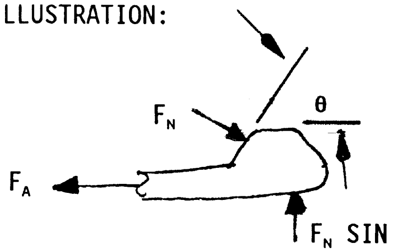
2. A long-term solution has been to add cylinder bypass (pressure relief) valves to dissipate energy of hard engine/rocket motor starts.

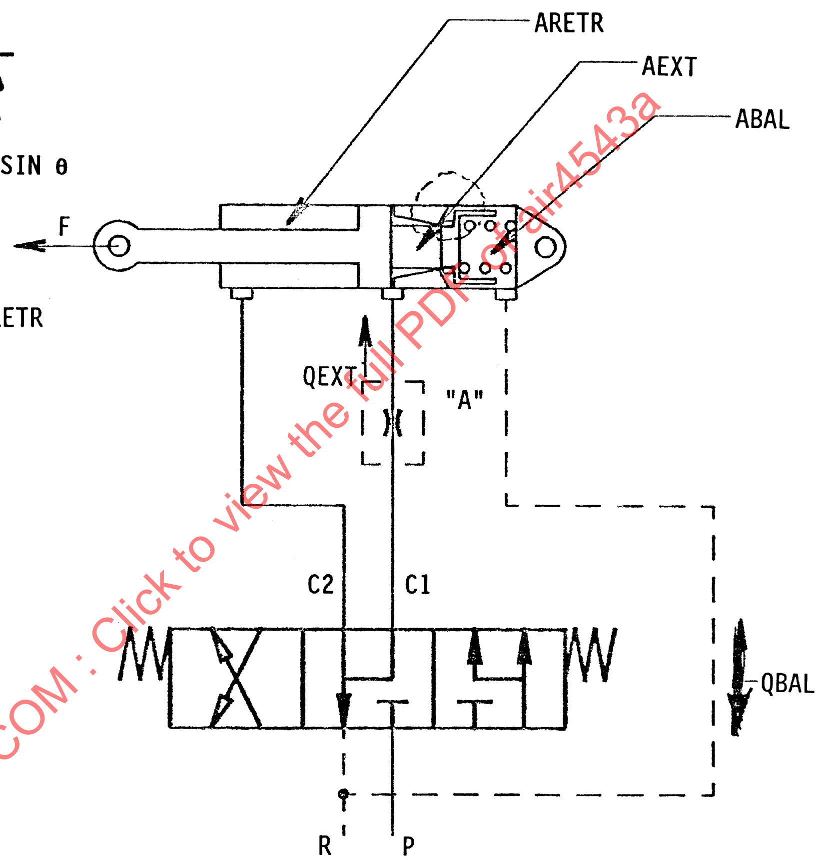
FIGURE 3.2.2.2 - Servovalve Separation From the Manifold Due to Servoactuator Velocity Limiting

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:





$F_A = F + P \times AEXT - P \times ARETR$

SOLUTION:

1. Size port and line to ABAL for high transient flow.
2. Relieve external F on fingers by application of retract pressure before application of extend pressure.
3. Choke the inlet flow QEXT at "A" to reduce the rate of pressure rise to AEXT.
4. Use one selector valve per actuator.
5. Guide and lock piston hardness to be much higher than lock fingers.

FIGURE 3.2.2.3 - Galling of the Cylinder Lock Mechanism When Unlocked Under High Loads

PROBLEM: FATIGUE FAILURES IN CYLINDERS WITH PARTING LINE 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).

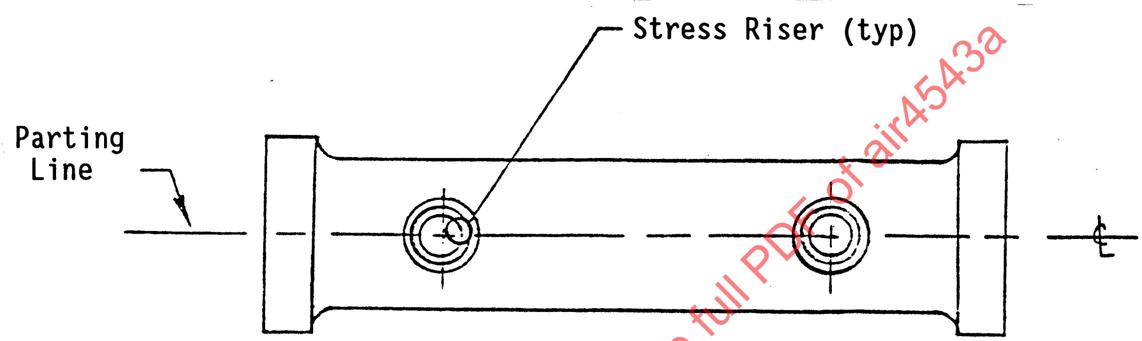


ILLUSTRATION 1

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

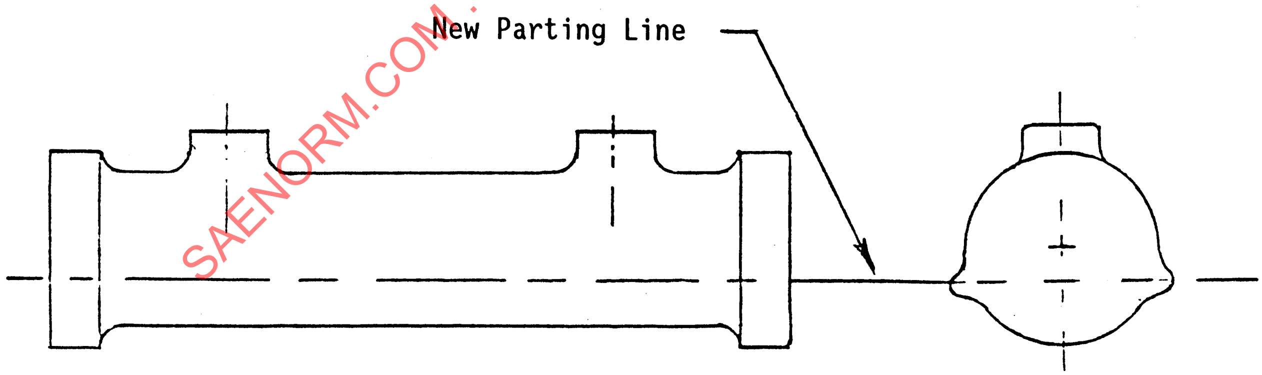


ILLUSTRATION 2

FIGURE 3.2.2.4 - Fatigue Failures in Cylinders With Parting Line Through Ports

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.

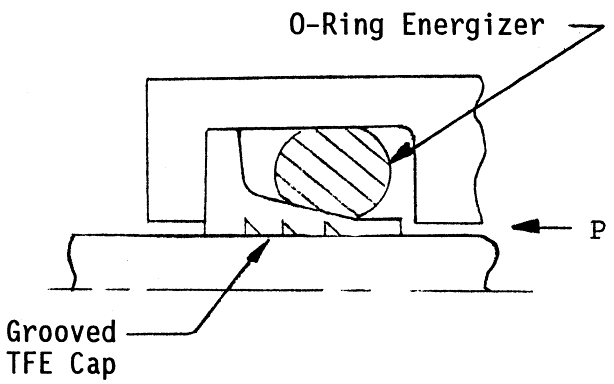


ILLUSTRATION 1

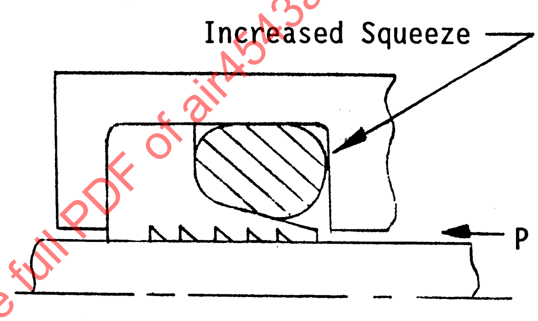


ILLUSTRATION 2

SOLUTION:

1. 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.)

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

3. 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.

4. 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# limit.

FIGURE 3.2.2.5 - Rejection of Low Friction Actuators for Dynamic Leakage

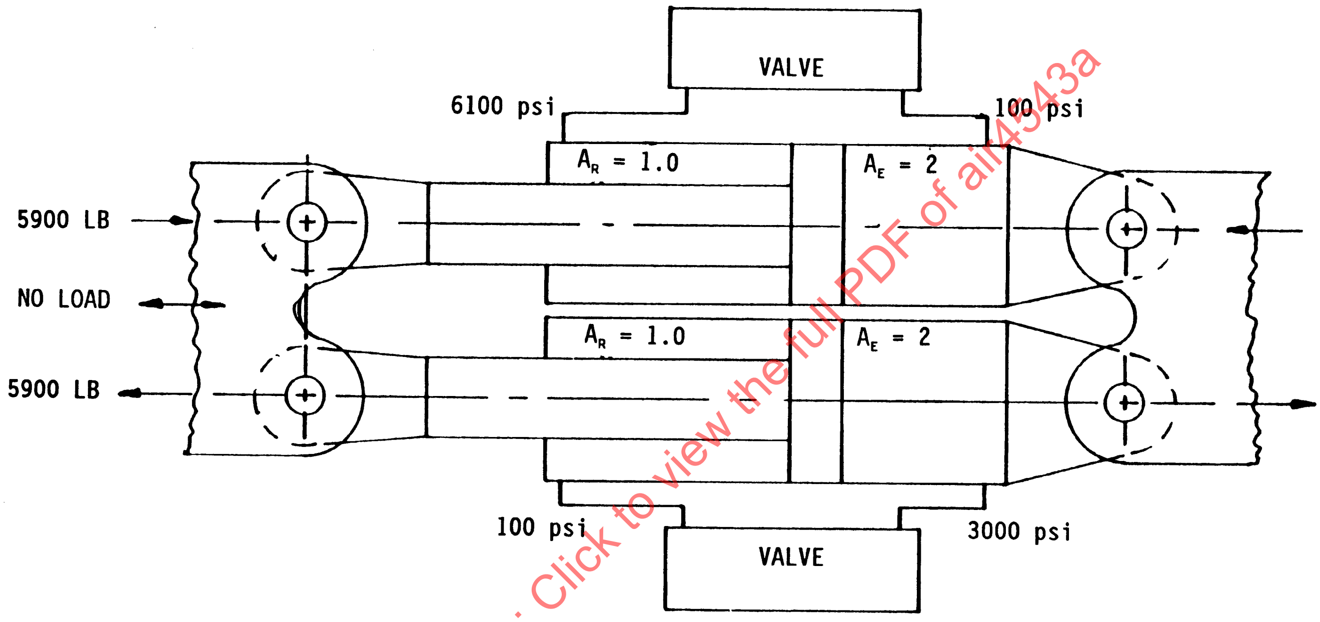
TABLE 1 - Seal Design Considerations

1. Be wary of friction calculations. Eccentricities can double calculated friction.
2. Provide smooth rod finishes - better than 8 μin.
3. Design actuator to minimize piston cocking and side loads with resultant rod scratching.
4. Use largest possible seal groove cross section for maximum design flexibility.
5. Seal energizing forces should be sufficiently high and consistent over expected temperature and life. Take advantage of all the friction that is allowable.
6. Seal cap material should be homogeneous - not grainy as is possible with some filled TFE.
7. Annular grooves in the cap helps reduce hydroplaning at higher piston velocities and reduces static leakage as well.

PROBLEM: FATIGUE DAMAGE TO LOADED STRUCTURE AND CYLINDERS

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

ILLUSTRATION:



SOLUTION:

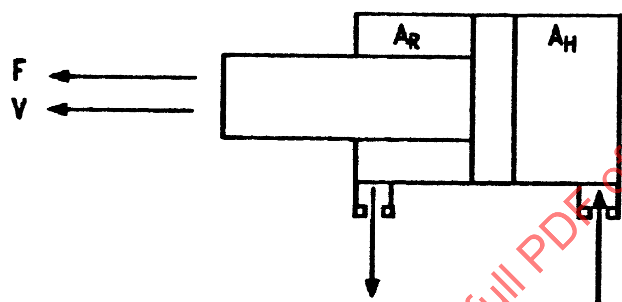
1. Use a single tandem valve for control pressure matching.
2. Mechanically couple separate valves for control pressure matching.
3. Use pressure transducers and electronic equalization to match control pressures.
4. Use underlapped separate servovalves to match control pressures at null (may still have a dynamic force fight problem).

FIGURE 3.2.2.6 - Fatigue Damage to Loaded Structure and Cylinders

PROBLEM: VELOCITY CONTROL OF EXTENDING ACTUATOR

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

ILLUSTRATION:



SOLUTION:

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

CHECKS:

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

F positive is aiding load

F negative is opposing load

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

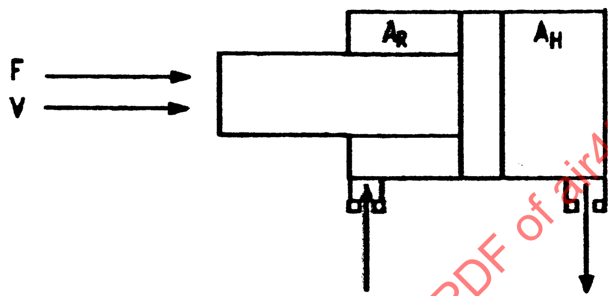
Head end pressure $= \left[P_S - \frac{A_H^2 V^2 C_1^2}{C_H^2}\right]$

FIGURE 3.2.2.7 - Velocity Control of Extending Actuator

PROBLEM: 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

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

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

FIGURE 3.2.2.8 - Velocity Control of Retracting Actuator

FORCE EQUATION TERMS

F = Actuator force, lb (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 = .26

P_R = Return pressure, psi

P_S = System pressure, psi

$C_R = 29.8\ C_{DR}\ d_R^2/\sqrt{S}$

C_{DR} = Coefficient of discharge, rod end restrictor

d_R = Diameter, rod end restrictor, in

$C_H = 29.8\ C_{DH}\ d_H^2/\sqrt{S}$

C_{DH} = Coefficient of discharge, head end restrictor

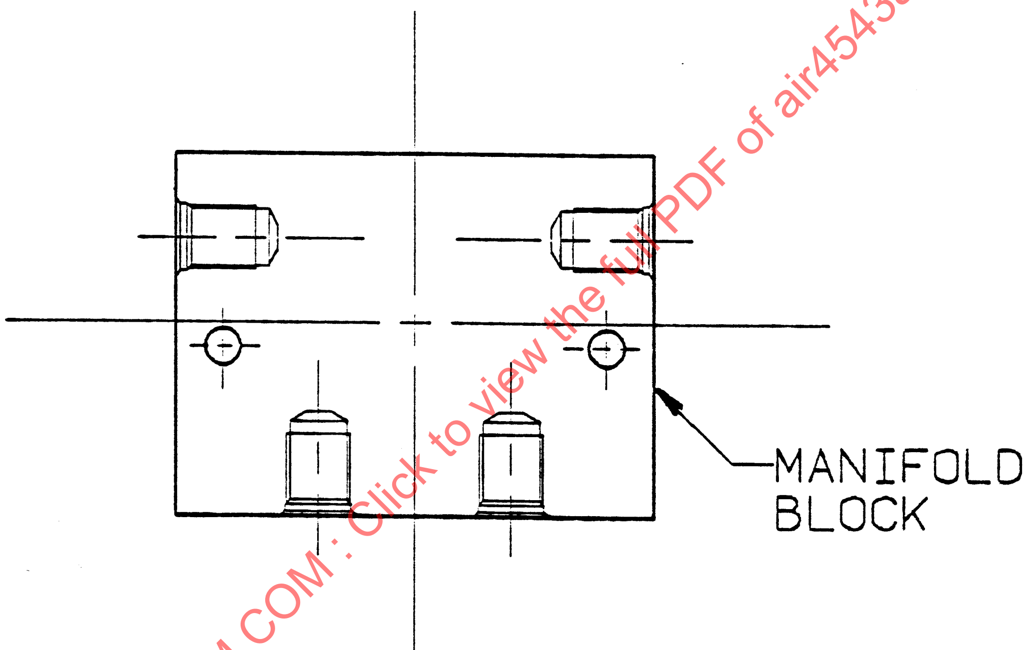
d_H = Diameter, head end restrictor, in

S = Fluid specific gravity

PROBLEM: AILERON 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.

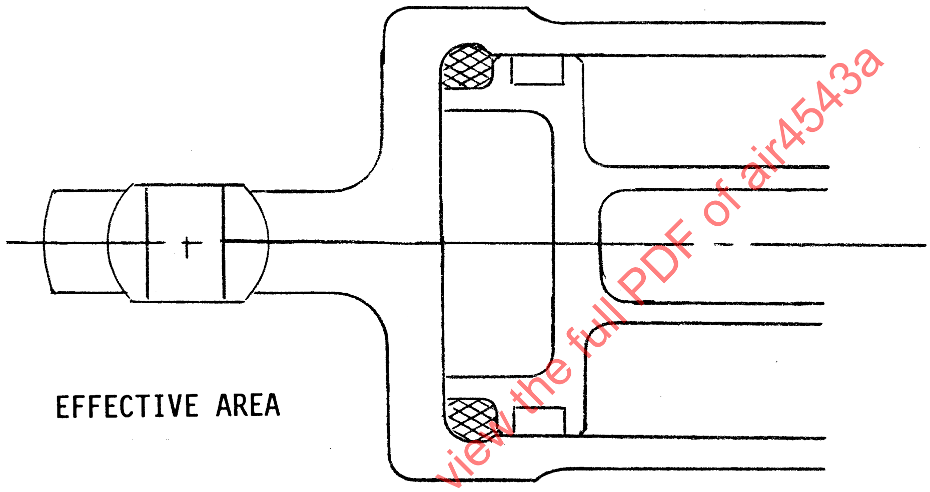
Never take for granted that any part is per the drawing.

FIGURE 3.2.2.9 - Aileron Failed to Operate

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:

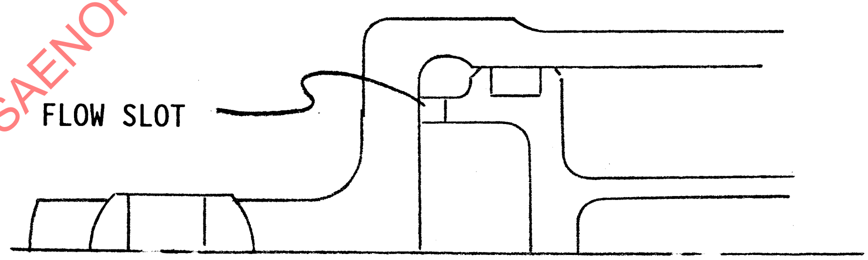


FIGURE 3.2.2.10 - Fully Retracted Actuator Fails to Extend Against Load

PROBLEM: QUALIFICATION TEST IMPULSE FAILURES OF VALVES AND ACTUATORS

ISSUE: Components are not designed for fatigue.

SOLUTION: Fatigue life should be considered when designing pressure vessels. Stress risers, such as sharp inside corners, tool marks, or small inside radii, should be avoided. If actual operating conditions are known or can be predicted, such as peak pressure, pressure rise rate, fluid temperature, and number of cycles, these values should be used for testing instead of Military Specifications or ARP requirements.

FIGURE 3.2.2.11 - Qualification Test Impulse Failures of Valves and Actuators

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:

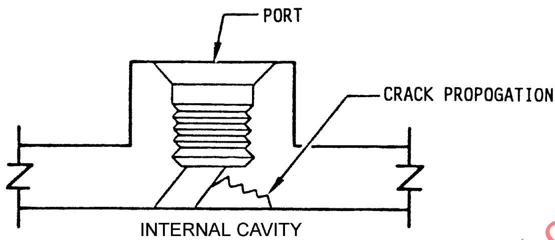
1. Return system tubing should be sized with consideration for ΔP required to accelerate the fluid.
2. ΔP switches and ΔP operated valves should be designed with damping to prevent problems from pressure transients.

FIGURE 3.2.2.12 - Inadvertent Actuation of ΔP Switches or ΔP Operated Valves in Servoactuators for High Response Rate Flight Controls

PROBLEM: ALUMINUM MANIFOLD FATIGUE

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

ILLUSTRATION:



SOLUTION:

1. Keep hole intersection angles as near 90° as possible (Illustration 1)
2. Break edges at hole intersections (Illustration 1)
3. Add a localized compressive stress layer if feasible (various methods) (Illustration 2)
4. Design to avoid - remove discontinuities where possible (Illustration 3)

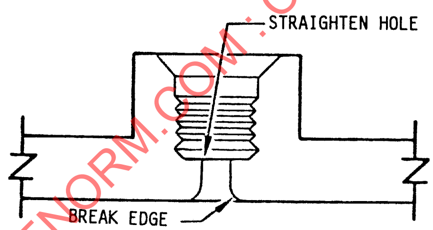


ILLUSTRATION 1

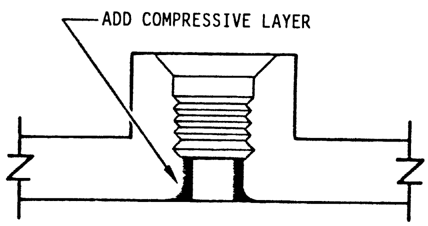


ILLUSTRATION 2

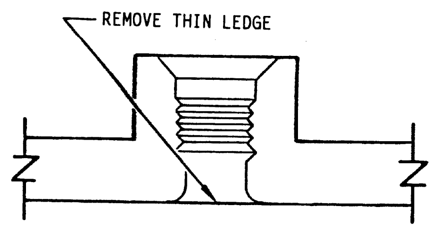


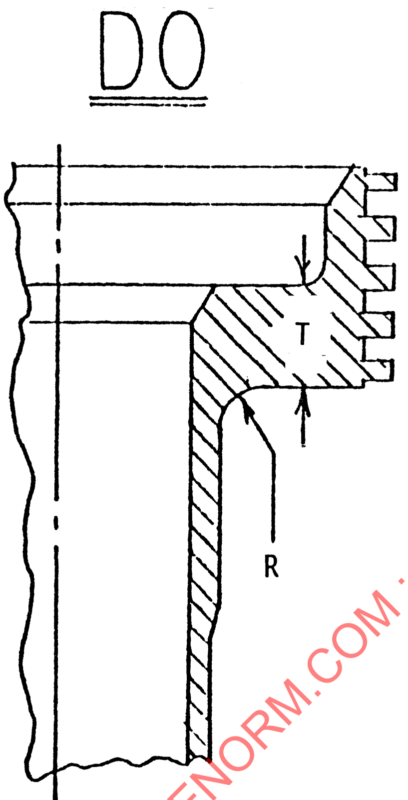
ILLUSTRATION 3

FIGURE 3.2.2.13 - Aluminum Manifold Fatigue

PROBLEM: PISTON HEAD SEPARATION

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

ILLUSTRATION:



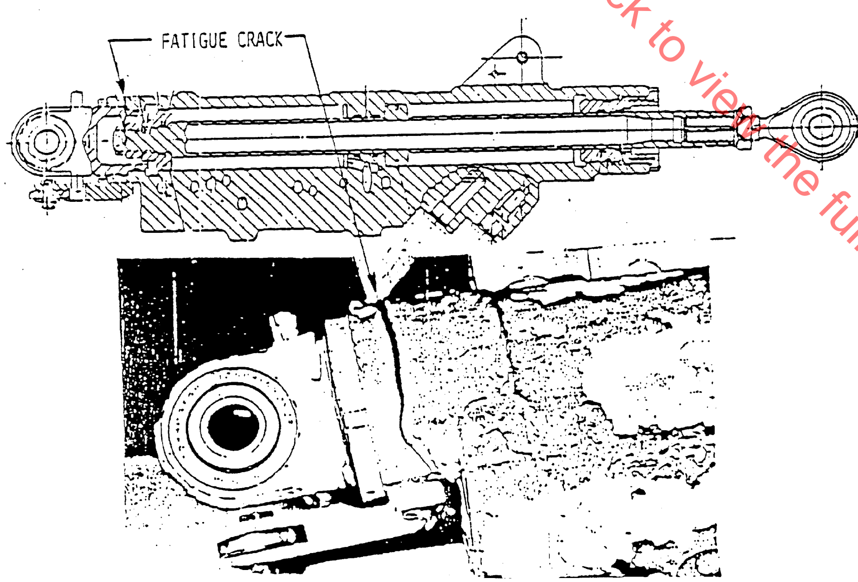
SOLUTION: Design pistons with ample wall thickness (T) and generous filet radii (R). Consideration must be given to peak transient loading. The type loading, whether pure tensile or combined tensile and bending, must be considered in relation to the stress distribution at the piston head filet to prevent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

FIGURE 3.2.2.14 - Piston Head Separation

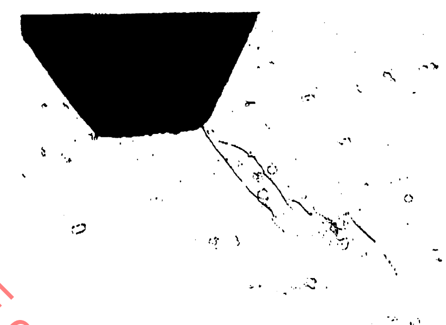
PROBLEM: FATIGUE FAILURE OF INTERNAL THREADS IN ALUMINUM COMPONENTS

ISSUE: Uncontrolled root radius of internal threads cause stress risers and consequent crack initiation sites.

SOLUTION: Specify minimum root radii when designing aluminum components with internal cut threads.



Thread root cross section of Actuator #1
Mag = 200x.
ACTUATOR S/N 380



Thread root cross section of Actuator #2, note crack at radius. Mag = 200x.
ACTUATOR S/N 278

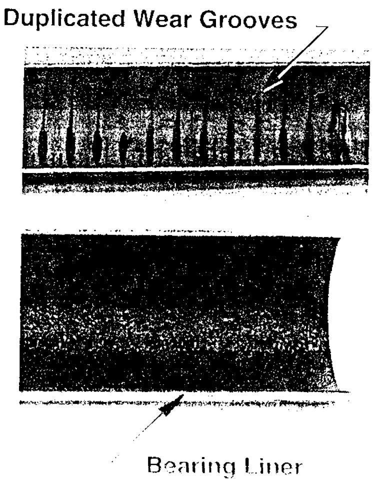
FIGURE 3.2.2.15 - Internal Threads in Aluminum Actuators and Components

PROBLEM: SPRINGS WEARING THROUGH BUNGEE TUBES

ISSUE:
- Flight Control Bungees sustaining severe wear of bungee tubes caused by vibration of internal springs.

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

Uncoated and Coated Test Articles



Sectioned Worn Bungee Assembly

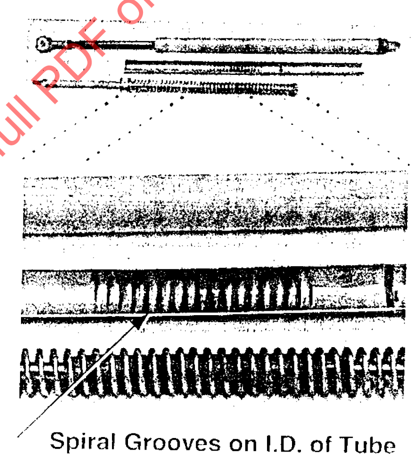


FIGURE 3.2.2.16 - Long-Term Wear of Flight Control Bungee Assemblies

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:
1. Split outer race allowing higher ball retention force.
2. Caged bearing design if space permits.

FIGURE 3.2.2.17 - Ball Bearing Failures on Servo Input Shaft on Actuator

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: Derate the bearing load capacity to account for this unusual usage. Assure that the actuator qualification verifies bearing endurance life.

FIGURE 3.2.2.18 - Ball Bearing Failure on Servo Input Shaft

PROBLEM: INADVERTENT ACTUATION OF ΔP SWITCHES OR ΔP OPERATED VALVES IN SERVO ACTUATORS 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 servo actuator.

SOLUTION:

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

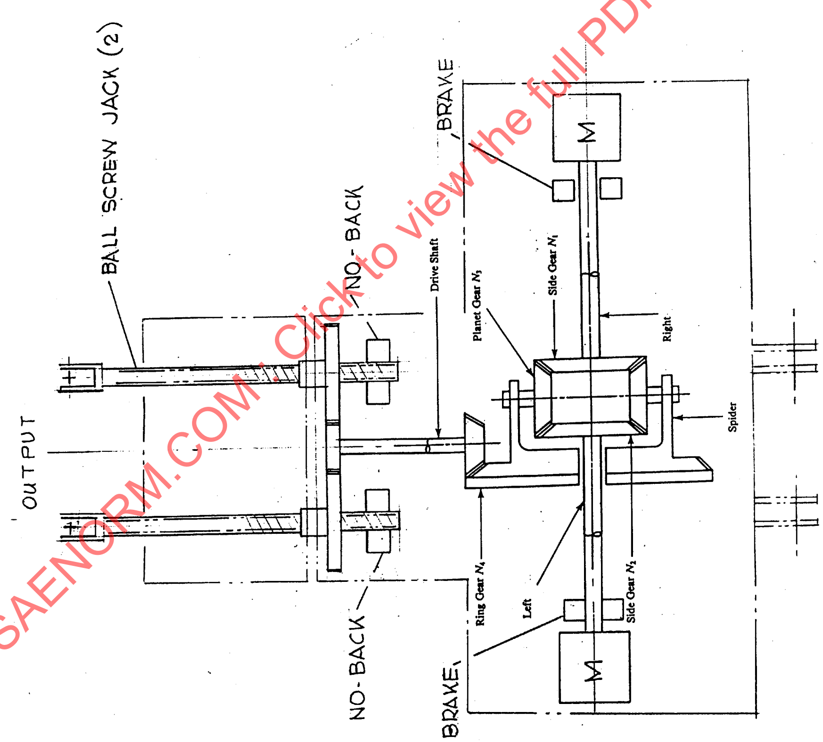
2. ΔP switches and ΔP operated valves should be designed with damping to prevent problems from pressure transients.

FIGURE 3.2.2.19 - Premature Actuation of Switches and Valves

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:



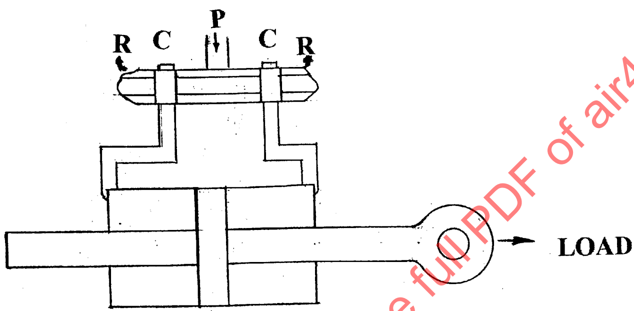
SOLUTION: Carefully examine the component or system for latent failures. If found, the best solution is simply to alter the design and eliminate the latent failure mode. The next best approach is to properly take into account the exposure of the latent failure mode in determining the actual system failure 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.

FIGURE 3.2.2.20 - Latent Failures in the Design of Flight Control Components

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 servo valve overlap conditions and continuous small movement commands which kept the valve within the overlap region.

ILLUSTRATION:



To meet low valve 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 psi extend force pressure load could require the resisting cylinder chamber pressure to approach 4500 psi in a 3000 psi 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 psi to 4500 psi 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.

SOLUTION:

1. Minimize overlaps but specify overlap balance conditions if a low leakage is required.
2. 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 psi range in the entire overlap region.
3. It is important to know what the FCC is requiring the actuator to do.
4. Minimize the frequency of small movement commands (system).

FIGURE 3.2.2.21 - Fly by Wire Actuator Pressure Fatigue Failures

PROBLEM: UNCOMMANDED INPUTS TO THE T-2C LONGITUDINAL FLIGHT CONTROL SYSTEM

- The training command was experiencing flight incidents, in the form of uncommanded pitch inputs, that led to the grounding of the training fleet due to safety-of-flight concerns.
- A task team consisting of Navy and industry and SAE experts was convened to study the problem.

ISSUE:

- Longitudinal flight control actuator utilizes "scarf cut" main piston seals. Teflon cap seal with O-ring energizer in wide groove, 1% squeeze.
- Longitudinal flight control actuator utilizes force feedback by means of balance pistons. Free floating type.

SUSPECT CAUSES OF PROBLEM:

- Main piston seal by-pass results in uncommanded actuator movement through unbalance of feedback pistons
- Potential sticking of feedback pistons result in uncommanded inputs

SOLUTION:

- Replace scarf cut main piston seal with uncut seal
- Increase seal squeeze to prevent by-pass, 10 to 12%
- Improve feedback piston design to reduce probability of sticking
- More balancing grooves, 13 versus 4
- Tighter clearance, 125 millionths
- Reduce jamming potential, reduce travel
- Straightness control

VALIDATION OF SOLUTION:

- Qualification test of new main piston seal design
- Qualification test of improved feedback pistons
- Flight test of reworked actuators with design improvements

FIGURE 3.2.2.22 - Uncommanded T-2C Flight Control System Inputs

PROBLEM: SALVAGE-HYDRAULIC CYLINDER BORES

ISSUE: Airlines periodically request information for salvage criteria of oversize bores in hydraulic cylinder barrels. The standard combination of materials used for the sliding surfaces in hydraulic cylinders is a chrome plated piston running in a bare steel cylinder bore. The cylinder bore, after years of service, eventually becomes oversize due to wear or corrosion. This causes high internal fluid leakage and may or may not reduce the force output. Some cylinder barrels have been salvage by grinding the entire cylinder bore length oversize, chrome plating to an undersize diameter and regrinding to blueprint tolerances. This is not acceptable because like materials, chrome on chrome, for adjacent sliding surfaces is susceptible to galling and is a poor choice for bearing combination. Also, thick chrome for corrosion protection provides minimal protection due to its porosity. Porosity may also reduce the fatigue life of the cylinder.

SOLUTION: Hydraulic cylinder steel bore salvage instructions:

1. Where plating is required for material buildup, use hard nickel plating.
2. Plate thickness should never exceed wear limits.
3. Plate thickness after machining back to drawing tolerances should never exceed 0.020 in per surface or be less than 0.0005 in per surface.
4. Rough machining after plating and then finish grinding to size is preferable to verify plate adhesion.
5. If corrosion in the static seal area is the only cylinder bore discrepancy, the bore may be locally ground oversize, plated undersize and remachined back to blueprint tolerances.

FIGURE 3.2.2.23 - Salvage of Hydraulic Cylinder Bores

PROBLEM: ACCEPTANCE TEST PROCEDURES - HYDRAULIC COMPONENTS

ISSUE: All hydraulic components are tested after final assembly and prior to delivery to customer. The components are tested to meet all the design requirements. Small loose parts may have been left inadvertently inside an assembled component, such as nuts, lockwire pieces, washers, etc. This may cause a malfunction such as lock linkage or binding of moving parts.

SOLUTION: All acceptance test procedures should specify:

1. The actual installed position on the aircraft.
2. The component to be in a similar position when all testing is performed.

FIGURE 3.2.2.24 - Acceptance Test Procedures - Hydraulic Components

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.

FIGURE 3.2.2.25 - Flight Control Attachment Bolt Failure - 3/16 in

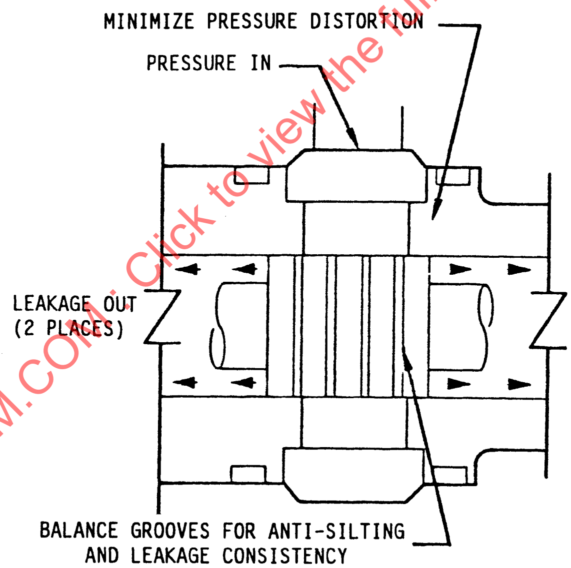
3.2.3 Filtration, Filters, and Fluids: This section contains lessons learned topics relating to hydraulic filtration, filters and fluids. The lessons learned are presented in Figures 3.2.3.1 through 3.2.3.18.

PROBLEM: VALVE REJECTIONS DUE TO SUPPLIER/CONTRACTOR FLUID FILTRATION VARIANCES

ISSUE:

1. Applies more to lap assembly valves than poppet-seat valves.
2. Finer filtration may result in significant increased leakage.
   a. Particularly if the valve is designed with a relatively thin sleeve and the designer is counting on hydraulic damming or silting to diminish leakage as a function of time.
   b. Low leakage lap assembly type valves are particularly subject to leakage increase when using finer filtered test fluid.

ILLUSTRATION:



SOLUTION:

1. Use better design practices.
   a. More rigid sleeve/spool/structure/balance grooves
   b. Balance grooves to minimize variances and prevent silting
2. Test per requirements (supplier and contractor); use the same level of filtration.

FIGURE 3.2.3.1 - Valve Rejections Due to Supplier/Contractor Fluid Filtration Variances

PROBLEM: MIL-F-8815 AIRCRAFT FILTERS PERFORM UNSATISFACTORILY IN SHIP AND SUBMARINE APPLICATIONS

ISSUE:

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

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

3. The viscosity of MIL-H-5606 aircraft hydraulic fluid varies from about 17 cst at 85°F to 14 cst at 115°F, a change of 20%. Submarine hydraulic fluid 2190-TEP (MIL-L-17331) varies from about 150 cst at 85°F to 60 cst at 115°F, a change of 250%. At normal operating temperatures the viscosity of MIL-H-5606 may increase 30% at 3000 psi 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:

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

2. 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.

3. 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.

FIGURE 3.2.3.2 - MIL-F-8815 Aircraft Filters Perform Unsatisfactorily in Ship and Submarine Applications

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 electrohydraulic 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.

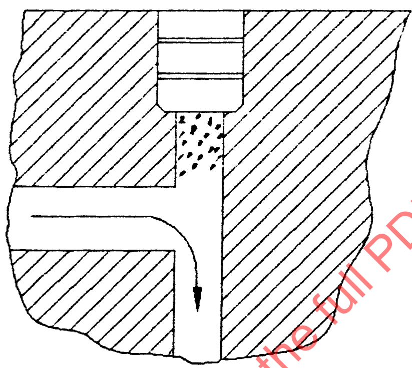
SOLUTION: Plug the bypass relief valves and use filter assemblies without bypass reliefs in new applications. If maintenance is neglected, the increased pressure drop across the filter increases and servovalve performance slowly deteriorates. The change in servovalve performance is no worse than when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

FIGURE 3.2.3.3 - Contamination Due to Filter Bypass Reliefs Lifting Results in Costly Maintenance of Servovalves

PROBLEM: ENTRAPPED CONTAMINATION

ISSUE: Dead end passages collect the contaminant.

ILLUSTRATION:



SOLUTION: 1. Reroute the flow passage to eliminate potential traps where flow will flush the contaminant through the valve.

2. Reduce the trap size.

ILLUSTRATION:

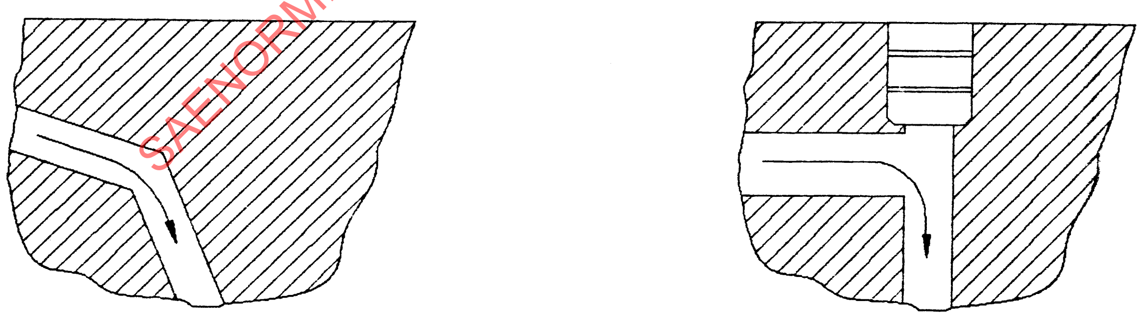
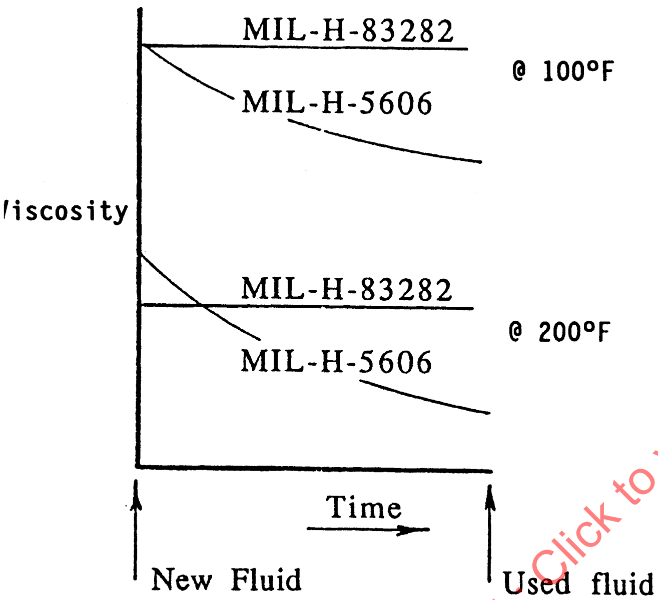


FIGURE 3.2.3.4 - Entrapped Contamination

PROBLEM: VISCOSITY EFFECTS ON PERFORMANCE

ISSUE: Acceptance tests with new fluid are unacceptable with used fluid.

ILLUSTRATION:



A low leakage requirement may test acceptably with a high viscosity, new, fluid and unacceptably with a low viscosity, used, fluid.

SOLUTION: Specify that test benches be operated for 8 h through orifices to shear fluid prior to conducting critical leakage acceptance tests.

FIGURE 3.2.3.5 - Viscosity Effects on Performance

PROBLEM: FAILURE OF MIL-F-8815/2-8 FILTER HOUSING

ISSUE: Failure due to compression ignition explosion.

Cause of Failure:

1. Air left in housing when installing replacement element

2. Rapid compression of air when shifting to emergency mode

Contributory Causes:

1. 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.

2. 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:

1. Fill filter bowls with fluid when changing elements.

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

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

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

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

FIGURE 3.2.3.6 - Failure of MIL-F-8815/2-8 Filter Housing

PROBLEM: CONTAMINATION OF HYDRAULIC FLUID DURING STORAGE

ISSUE: Although Type IV hydraulic fluids have no shelf life restriction, improper storage of hydraulic fluid can result in water and dirt contamination of the fluid, which eventually ends up in an aircraft hydraulic system. This can cause premature failure of the fluid and corrosion of system components.

SOLUTION: By following proper storage and transfer procedures of the hydraulic fluids, the contamination problem can be eliminated. The recommended practice is as follows:

a. Store drums indoors whenever possible and always keep lids well tightened except when transferring fluid. Smaller containers such as 5 gal cans and quart cases should be stored indoors only.

b. When transferring fluid take care that pumps, filters, and transfer lines are clean and dry.

c. Store drums horizontally if possible to prevent water collection on the drum lid. If the drums cannot be stored horizontally, they should at least be blocked up and tilted so that any water collection on the lid will not cover the bungs.

d. For all size containers the use of pallets, racks, or shelves is suggested to prevent contact with ground moisture.

e. Because of the possibility of contamination, fluid stored more than three years should be analyzed before use. This is not required for quart containers since they are hermetically sealed.

FIGURE 3.2.3.7 - Contamination of Hydraulic Fluid During Storage

PROBLEM: INADEQUATE PUMP LIFE AT ELEVATED TEMPERATURES

ISSUE:

1. When a hydraulic pump was run in preservative fluid, MIL-H-46170, at 230 °F severe corrosion of bronze and copper parts occurred. Pump would not operate after 200 h of endurance test time.
2. Fluid tests indicated that above 160 °F, the barium dinonaphtalene sulfonate corrosion inhibitor breaks down to form strong sulfonic acids which attack bronze in the pump. Acid number of fluid increased.
3. Corrosion inhibited hydraulic fluids were developed for recoil mechanisms of army guns where temperatures are low, less than 160 °F. Also they are used successfully in tank turrets at low operating temperatures, less than 160 °F.

SOLUTION:

1. Do not use preservative fluids containing barium dinonaphtalene sulfonate as operational fluids above 160 °F.
2. Consider banning the use of MIL-H-46170 as a shipping fluid for components as clogging of filter may result in aircraft systems.

FIGURE 3.2.3.8 - Inadequate Component Life at Elevated Temperatures

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.

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

SOLUTION: Investigation of hydraulic system revealed that the hydraulic fluid was MIL-H-5606, aircraft had not been converted to MIL-H-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-H-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.

FIGURE 3.2.3.9 - Ice Formation in Hydraulic Fluid

PROBLEM: SERVOVALVES WERE STICKING IN UH-1 HELICOPTER HYDRAULIC SYSTEMS

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

- Resulted in grounding the entire fleet of helicopters stationed at Fort Rucker - Training base for Army helicopter pilots
- Meeting of pilots, maintenance and program office personnel, Army and Air Force scientists and contractor personnel convened to discuss problem
- Training mission included hydraulics off maneuvers
- 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:

- Reduce barium content in helicopter hydraulic fluid to <15 ppm
- Remove and clean stuck valves from helicopters and re-install
- Assure that rust inhibited hydraulic fluid (MIL-H-6083) is thoroughly drained from hydraulic components before they are installed on helicopters

VALIDATION:

- No stuck valves for 1st year after implementation of above steps
- Similar deposit found in laboratory tests involving dithering cycle of poppet on seat with contaminated hydraulic fluid

FIGURE 3.2.3.10 - Fluid Additive Caused Stuck Servovalve

PROBLEM: BLOW-OUT FAILURE OF HYDRAULIC FILTER ELEMENT BASE CAP AT 3000 PSI

ISSUE:

1. A slug of bottom cap was blown out of end cap and could have drifted downstream into system.
2. Failure caused by unknown volume trap of zero pressure which created unexpected differential pressure on end cap of 3000 PSI.
3. Failure was on second source element and was not detected during qualification testing.
4. Element bottomed and aligned on dome in base of bowl.

SOLUTION:

1. Immediate fix for filters in the supply system was to machine relief groove into base cap to eliminate zero pressure trap.
2. Final fix was to require a filter element end cap that would withstand 4500 PSID.

FIGURE 3.2.3.11 - Hidden Zero Pressure Volume in Filter

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:

1. Evaluate Filter Clogging At 200 and 400 Hrs.
2. 200 Hour Replacement Recommended From Data
3. C.D. Filter Replacement Reduced Pump Failures and Improved Mean Time Between Failures From 950 Hours to 2000 Hours

FIGURE 3.2.3.12 - Case Drain Filter Replacement Interval

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 will further accelerate oil oxidation.

Run	Catalyst	Water	Hours	Total* Acid Number Change
1	None	No	3500+	0
2	None	Yes	3500+	+.73
3	Iron	No	3500+	+.48
4	Iron	Yes	400	+7.93
5	Copper	No	3000	+.72
6	Copper	Yes	100	+11.03

* 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.

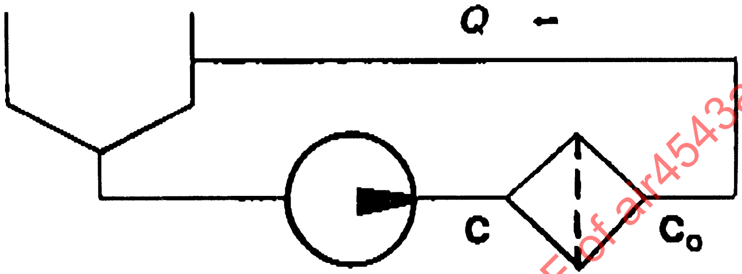
SOLUTION:
1. Operate system at lowest temperatures possible and avoid areas of excessive heating.
2. Design system to reduce possibility of ingression of water and contaminants (use closed reservoir or use reservoir vent filter/dryer).
3. Use a fine "green run" filter and a high efficiency (MIL-F-81836) GSE filter to remove built-in contaminants.
4. 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.
5. Use a fluid purifier to reduce water concentration and solid contaminants in the oil.
6. Use in-system air eliminator or automatic air bleed valve.

FIGURE 3.2.3.13 - Hydraulic Fluid Oxidation

PROBLEM: EXCESSIVE HYDRAULIC SYSTEM CLEANUP TIME

ISSUE: Hydraulic system cleanup time is excessive after contamination due to a component failure or from external contaminant ingression.

ILLUSTRATION:



Equation expressing the variation in the concentration of particles over time, assuming an initial concentration (C_I), an idealized system with instantaneous mixing in the reservoir and no contamination "traps" in the system:

$$\frac{C}{C_I} = e^{-\frac{Qt}{V}\left(1-\frac{1}{\beta}\right)}$$

C = Concentration of particles upstream of filter
C_I = Initial concentration of particles at time t = 0
C_o = Concentration of particles downstream of filter
β = Ratio of concentration upstream of filter to concentration downstream of filter (measure of filter efficiency)
Q = Flow rate
t = Time
v = System volume

SOLUTION:

1. Use a system filter with higher efficiency.
2. Use "green run" filter of higher efficiency than system filter, and high efficiency (MIL-PRF-81836) GSE filter.
3. Increase flow rate.
4. Increase fluid temperature which will increase Reynolds number, removing contaminant settled in quiescent areas.
5. Design systems without contamination traps. Use smooth transitions, eliminate unnecessary bends and corners.
6. Use of additional system filters (i.e., return line, case drain, last chance, reservoir vent, etc.).

FIGURE 3.2.3.14 - Hydraulic System Fluid Clean-Up

PROBLEM: FILTER CONDITION INDICATOR, DELTA "P" BUTTONS, WOULD EXTEND AND BE PUSHED BACK DOWN. LOADED FILTERS NOT BEING REPLACED.

ISSUE: Fleet maintainers were resetting popped Delta "P" buttons. Conflicting publications allow resetting once or three times. No tracking of resets was kept. Therefore, every reset was the first for each maintainer. By-passing filters could result in component damage.

SOLUTION: Educate fleet that when a filter button pops, the filter is to be replaced. No resets allowed. Buttons are designed for 15 G's and hard landings should not pop buttons. Filters also designed for low temperature lock-out at 100 °F. Institute forced filter removal at maintenance intervals.

FIGURE 3.2.3.15 - Delta Pressure Filter Indicators Popping

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 uncommanded pitch problem.

SOLUTION:

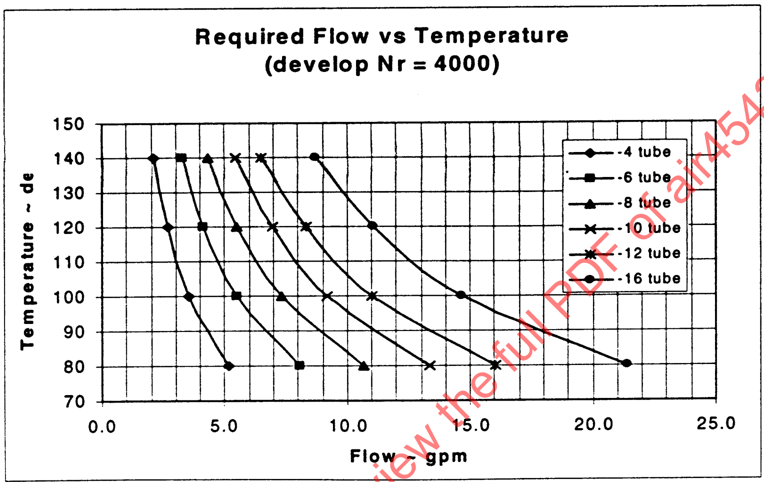
1. Require pre/post flight monitoring of filter Delta "P" buttons.
2. Change pressure filter to non by-pass type as specified in AS5440 (previously MIL-H-5440).
3. Do not allow painting over buttons during rework.
4. Periodically check operation of Delta "P" indicators.

FIGURE 3.2.3.16 - By-Pass Type Pressure Filter and Delta “P” Button Surveillance

PROBLEM: SYSTEM WOULD NOT CLEAN TO CLASS 6 WITH FLUSHING

ISSUE: It was reported that a section of hydraulic plumbing would not clean up to a NAS1638 Class 6 with flushing.

ILLUSTRATION:



SOLUTION: It was determined that the flow rate and fluid temperature being used were not per written guidelines. System cleanup is fast and repeatable when flushing flow rate creates a Reynolds number => 4000 and fluid temperature is 120 to 130 °F.

FIGURE 3.2.3.17 - Flushing System to Class 6

PROBLEM: HYDRAULIC COMPONENT ACCESSIBILITY - FILTERS

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 to never require removal of components in order to service another component.

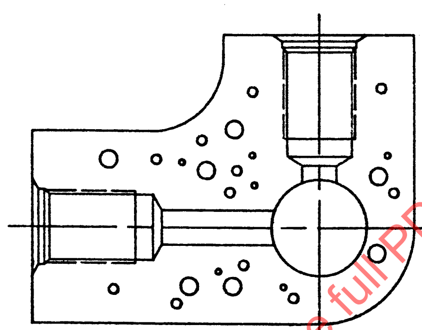
FIGURE 3.2.3.18 - Hydraulic Component Accessibility - Filters

3.2.4 Corrosion, Materials, and Processes: This section contains lessons learned topics relating to corrosion, materials, and processes. The lessons learned are presented in Figures 3.2.4.1 through 3.2.4.16.

PROBLEM: CASTING FATIGUE

ISSUE: Casting "porosity" caused short fatigue life.

ILLUSTRATION:



SOLUTION: Hot isostatic press (HIP) casting to reduce internal porosity size.

ILLUSTRATION:

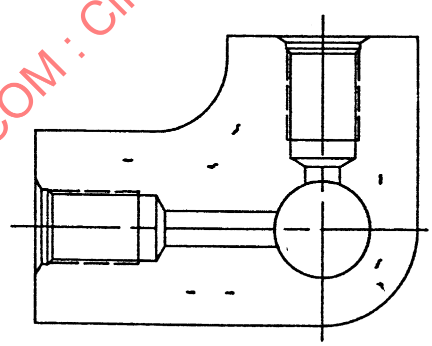
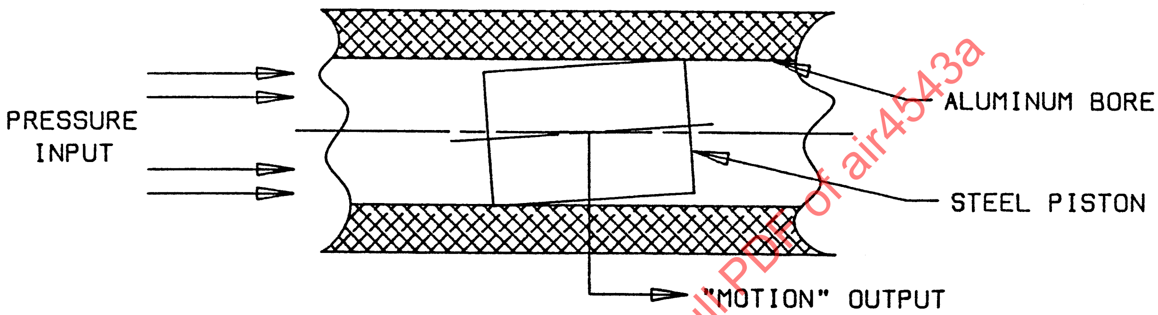


FIGURE 3.2.4.1 - Casting Fatigue

PROBLEM: SLIDING PISTON "STICKING" IN BORE

ISSUE: Steel piston "sticking" in aluminum bore.

ILLUSTRATION:



SOLUTION: Use PTFE impregnated "hard anodize" bore.

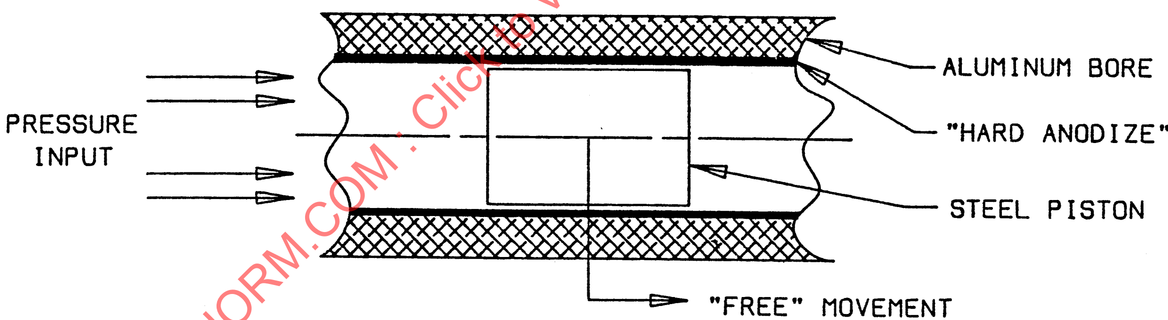


FIGURE 3.2.4.2 - Sliding Piston “Sticking” in Bore

PROBLEM: VALVE JAM CAUSED BY RETAINED AUSTENITE

ISSUE:

1. High heat treat 440C corrosion resistant or tool steels are used for critical valve lap assemblies.

2. Retained austenite in these steels may transfer to a martensitic structure with time. Size change and distortion occurs during this transition.

SOLUTION:

1. Minimize retained austenite to approximately 2% maximum by thermal cycle shocking, i.e., alternate cold (-100 to -120°F) and hot (to tempering temperature) a sufficient number of times to stabilize, usually two cycles.

2. Can determine effect by X-ray if desired.

FIGURE 3.2.4.3 - Valve Jam Caused by Retained Austenite

PROBLEM: CORROSION, CAUSED BY CHLORINE CONTAMINATION IN MIL-H-5606 HYDRAULIC FLUID CAN CAUSE ERRATIC BEHAVIOR AND FAILURE OF SERVOVALVES, ACTUATORS, AND PUMPS

ISSUE: Chlorine induced corrosion in hydraulic systems has resulted in failures of low chromium steel components, such as actuator control valves, pump compensators, and pump and transformer bearing surfaces. The corrosion is caused by chlorinated solvent contamination of the system hydraulic fluid. The contaminant has been identified as 1,1,2-Trichlorotrifluorethane (freon TF) used as a cleaning agent on hydraulic system components.

SOLUTION:

1. Prohibit the use of all chlorinated cleaning solvents in or around hydraulic system components. Use only P-D-680 Type II solvent during system maintenance, overhaul, and repair of hydraulic components.

2. Establish periodic fluid sampling and analysis of aircraft and ground support hydraulic systems. Any system found exceeding a 200 ppm chlorine limit should be drained, flushed, or decontaminated with ground purifier.

3. Utilize high chrome (greater than 12%) steels in the design of hydraulic components whenever possible.

FIGURE 3.2.4.4 - Corrosion, Caused by Chlorine Contamination in MIL-H-5606 Hydraulic Fluid Can Cause Erratic Behavior and Failure of Servovalves, Actuators, and Pumps

PROBLEM: FAILURE OF ALUMINUM AT ELEVATED PRESSURES

ISSUE:

1. Manifolds and components made from aluminum alloys fail in fatigue in high pressure systems.

2. Stresses induced into the part when operating at high pressures can exceed the endurance limits of the material. Extreme care must be taken to eliminate all stress risers, which is often impossible to control on complex shapes.

SOLUTION: Manufacture parts from a material having higher endurance limits and fracture toughness. Use titanium or stainless steel.

FIGURE 3.2.4.5 - Failure of Aluminum at Elevated Pressures

PROBLEM: SHOT PEENING OF BARRELS

ISSUE:

1. Thin shells (barrels) - potential for distortion with excessive or improper peening.
2. Potential for contamination due to incorrect peening, poor masking, or inadequate cleanup.
3. Quenched and tempered hard steel barrels
   a. Smooth hard surface - shot peening reduces fatigue life particularly in run out areas (transition to compression) (Illustration 1).
   b. Adding standard chrome plating - shot peening enhances fatigue life and forms a compression barrier under the chrome crack notches (Illustration 2).
   c. Adding thin dense chrome (in lieu of standard thick chrome) - shot peening probably does not enhance life and may actually reduce life (thin film).
4. Excessive shot peening may not allow conformance to seal finish requirements.
5. Failure to shot peen in a critical fatigue area results in a less than expected life.

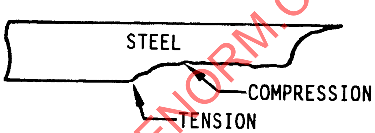


ILLUSTRATION 1

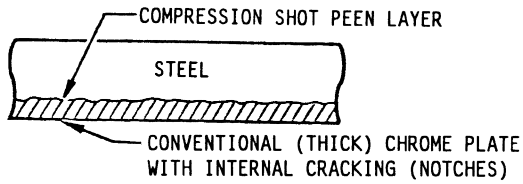


ILLUSTRATION 2

SOLUTION:

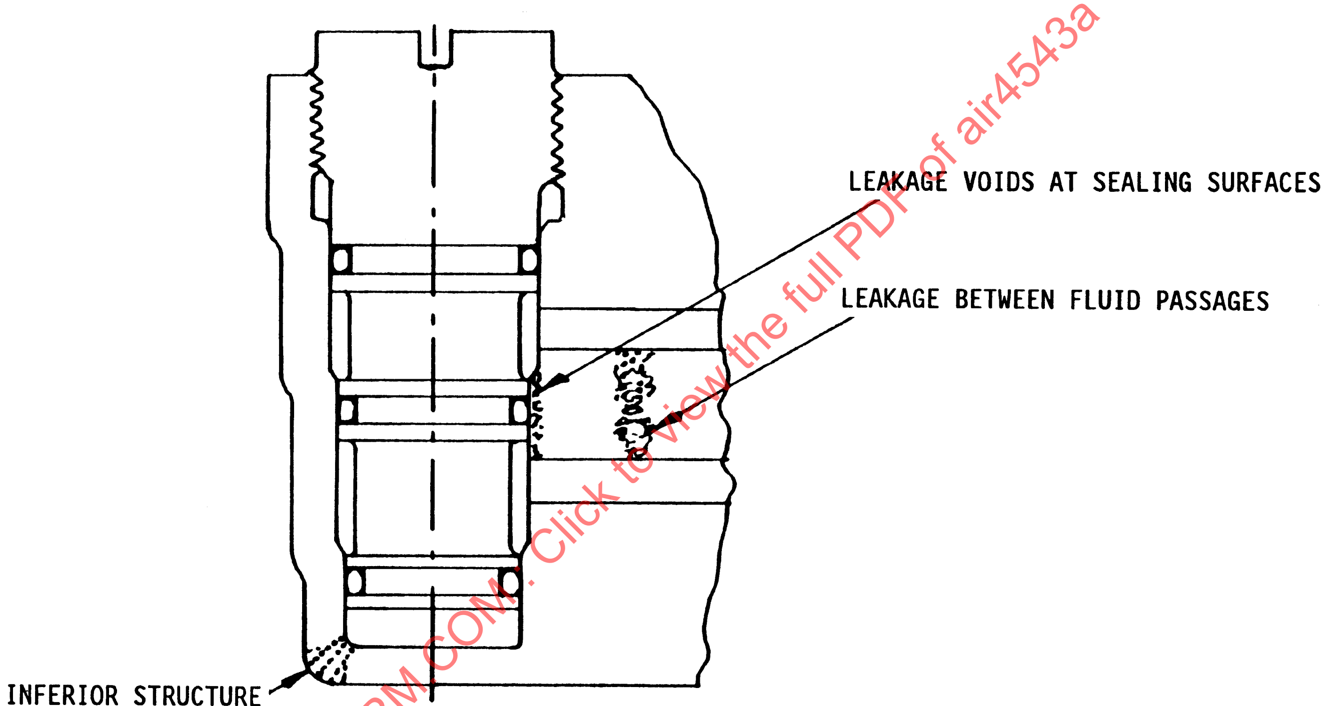
1. Use shot peening only when it is useful.
2. If shot peening is used, employ per proven procedures (as in any process oriented technique).
3. Avoid the need to count on shot peening to meet life requirements.

FIGURE 3.2.4.6 - Shot Peening of Barrels

PROBLEM: LEAKAGE AND INFERIOR STRUCTURAL INTEGRITY ASSOCIATED WITH POROSITY IN CAST MANIFOLDS

ISSUE: Trapped gases and/or inadequate gates and risers result in shrinkage voids and porosity in manifold castings.

ILLUSTRATION:



SOLUTION:

1. Improve the casting gates, risers, and vents.
2. Use the vacuum casting process to reduce trapped gases.
3. Use hipping of the castings to reduce internal porosity.
4. Impregnate the castings to improve sealing.
5. Weld repair the larger defects and voids.

FIGURE 3.2.4.7 - Leakage and Inferior Structural Integrity Associated With Porosity in Cast Manifolds

PROBLEM: VANADIUM PERMENDUR MATERIAL STRENGTH NOT AS PUBLISHED

ISSUE: Parts cracked under load.

SOLUTION:

1. Material allowable was determined by a test.
2. Parts redesigned to the determined strength rather than published strength.

FIGURE 3.2.4.8 - Vanadium Permendur Material Strength Not as Published

PROBLEM: PRESSURE TRANSDUCER FAILED FROM POTTING EXPANSION AND MOISTURE ENTRY

ISSUE:

1. Potting compound excessive thermal expansion broke solder joint.
2. Moisture is sucked into the inside.

SOLUTION:

1. Add an O-ring behind the connector.
2. Change the connector to be hermetically sealed.
3. Change the potting compound to have low thermal expansion coefficient.

FIGURE 3.2.4.9 - Pressure Transducer Failed from Potting Expansion and Moisture Entry

PROBLEM: INTERNAL CORROSION OF HYDRAULIC COMPONENTS

ISSUE:
1. Corrosion from chlorine/water contaminant.
2. Chlorine source is cleaning solvent residue.
3. Water source is condensation during fill operation and from vented GSE reservoirs.

SOLUTION:
1. Dry parts cleaned with solvent (1 h minimum).
2. Use air or vacuum oven.
3. Flush complex components with clean oil.
4. For operating aircraft
   a. Establish a 150 ppm maximum water limit.
   b. Establish a 25 ppm maximum chlorine limit.
5. Decontamination procedures
   a. Vacuum purge system reservoirs to 25 in-hg.
   b. Drain, refill, flush, or decontaminate until acceptable levels are reached.
6. Material design
   a. Use corrosion resistant steels - 440C, BG-42, etc. when possible.
7. Close or pressurize support equipment reservoirs.

FIGURE 3.2.4.10 - Internal Corrosion of Hydraulic Components

PROBLEM: LOW FATIGUE LIFE WITH ALUMINUM 7075-T73

ISSUE: Anodize causes early failures, reduced fatigue life.

MINIMUM LIFE (7500 PSI, 200°F, 3 HZ, 0.3125 INCH BORE)	
ANODIZE	BARE
X CYCLES	2X CYCLES

NOTE: SAMPLE SIZES OF FIVE

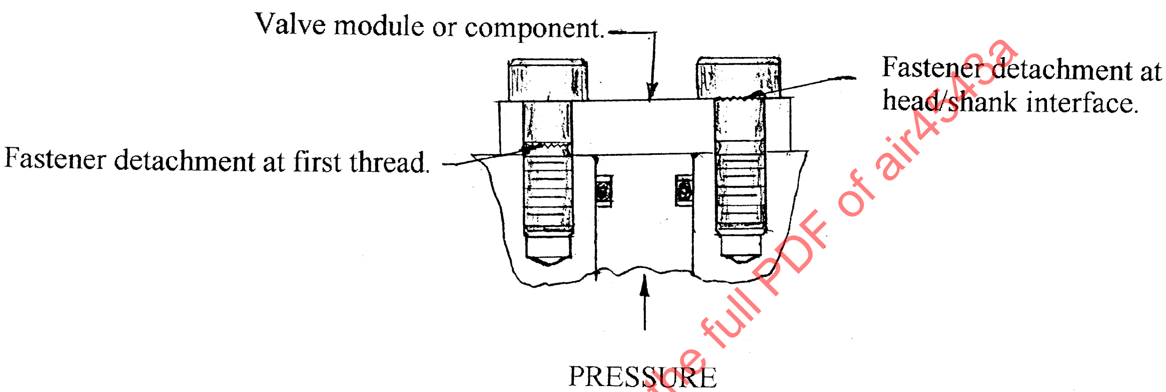
SOLUTION: Use bare internal passages.

FIGURE 3.2.4.11 - Anodize Effect on Fatigue Life of Aluminum

PROBLEM: FIELD FAILURES OF MOUNTING BOLT AND SCREW TYPE FASTENERS

ISSUE: Early "in-service" failures of fasteners due to the process of hydrogen embrittlement.

ILLUSTRATION:



Hydrogen embrittlement may occur when atomic hydrogen is present in a metal under load and diffuses over time to the higher stressed areas, promoting crack growth until failure. It is usually associated with high strength steel materials and electroplating, i.e., a process that involves hydrogen and a material that is susceptible to the process.

1. There have been several incidents wherein all four heads of a four screw or bolt pattern have "popped off" and the hydraulic system was lost.
2. Numerous incidents have occurred wherein heads "popped off", while installed on a unit and sitting on the shelf, due simply to initial, proper preloading plus the presence of diffused hydrogen induced during the fastener manufacturing process.
3. Proper processing primarily involves timely baking after plating for a proven time period and insuring that all fasteners in the baking lot are properly processed.

SOLUTION:

1. Incorporate sufficient manufacturing controls to insure proper processing and verify in accordance with ASTM 1940 (process control verification).
2. Better Solution: Use fastener materials that are not subject to hydrogen embrittlement process (such as A286, a CRES steel that does not require plating), i.e., remove the process sensitivity by removing the need for the process.

FIGURE 3.2.4.12 - Hydrogen Embrittlement of Hydraulic Component Bolts

PROBLEM: COUNTER DRILL HOLE DESIGNS LEAD TO THIN WALL CONDITIONS AND DANGEROUS SAFETY-OF-FLIGHT SITUATIONS IF NOT ADDRESSED IN THE DESIGN, MANUFACTURING, AND PARTS INSPECTION PHASE

ISSUE:

- Inadequate safeguards during machining operations
- Inadequate evaluation of tolerance factors
- Inadequate inspection of detail parts
- Tolerance pile-up causes thin wall conditions
- Uncontrolled machining of detail parts

SOLUTION:

- Avoid dimensional tolerances that rely on proper machining to prevent dangerous conditions
- Use drill stops when machining counter drilled holes
- Define counter drilled depths and associated external shoulder dimensions as "critical dimensions"

FIGURE 3.2.4.13 - Counter Drill Hole Designs Lead to Thin Wall Conditions

PROBLEM: UNRELIABLE AND TIME CONSUMING DEBURR OPERATIONS

ISSUE: Substandard impulse life on fielded parts

- Improper radii (not representative of qual unit)
- Nonpolished surfaces
- Damaged surfaces

APPROACHES:

- Manual deburring
- Slurry deburring
- Electrolytic deburring

MANUAL DEBURRING

- Recognize deburr as a precision machining operation
- Train deburr people
- Certify deburr people
- Provide excellent working conditions
- Provide excellent deburr tools
- Require 100% inspection/buy-off on deburr radii and finishes

SLURRY DEBURRING (EXTRUDE HONE)

- Pumps silicon/carbide slurry under pressure through passages to remove burrs, imperfections, EDM recast, etc.
- Reliable method
- Provides polished finish
- Multiple radii can be deburred simultaneously (2 h/circuit for 0.030 °R)
- Sealing surfaces can be affected
    - Plug off sealing surfaces
    - Finish sealing surfaces after hone
- Fluid flow sensitive (velocity & directional)
    - Removes more mtrl on inside radii of corners
- Slurry must be completely removed
    - Contamination potential
    - Multiple washes needed
    - 100% inspection for cleanliness
    - Time consuming
- Process must be controlled
- Require 100% inspection/buy-off on key deburr radii to verify proper processing occurred
- Freeze the planning

FIGURE 3.2.4.14 - Deburring Approaches

ELECTROLYTIC DEBURRING

- Metal is removed by reverse plating
- Does not affect parent material metallurgy
- Works on most metals
- Repeatable method
- Short duration (approx 60 s)
- Can simultaneously deburr multiple radii
- Operation can be in-house
- Does not affect sealing surfaces or passages
- Limited radii size: trade off size vs finish
    - 0.020 in radius typical (polished finish)
    - 0.060 in radius with 16 micro finish achievable
- Requires tooling
    - Tools center on bores
    - Negligible tool wear
- Require 100% inspection/buy-off on deburr radii to verify proper processing occurred
- Freeze the planning

FIGURE 3.2.4.14 (Continued)

PROBLEM: LACK OF TUNGSTEN CARBIDE ADHESION FOLLOWING TUNGSTEN CARBIDE REWORK

ISSUE: Incomplete stripping of tungsten carbide will yield unacceptable tungsten carbide recoat. Hydraulic pressure propagation lifts recoat significantly. Verification of complete strip on stainless steels requires trained eye.

SOLUTION:

- Internally approved documented rework process
- Separate rework documentation for each stripping and re-coating episode
- Fully traceable rework of part to original lot
- Documented reason for rework
- Certified and trained stripping inspectors
- Identify and record operator performing stripping and post strip inspector. Also verify inspector is certified.
- Recorded results of copper sulfate test
- Require copper sulfate solution check on entire stripped surface for nonstainless steel parts. A copper color staining is positive indication that the coating has been completely removed.

FIGURE 3.2.4.15 - Tungsten Carbide Coating Rework

PROBLEM: ROTATING JOINT LUBRICATION

ISSUE: Lubrication problems in normal aircraft service are normally voiced as follows:

- Accessibility - do not have room to get the lubrication gun in place to do the job or the grease fitting is at the top and it is impossible to get to it.
- Grease is not flowing in and the joint is frozen or dried up grease has prevented the joint from being lubricated.
- Is a frequent lubrication cycle on certain specific joints necessary?

SOLUTION: Lubrication requirements vary in relation to duty cycle, degree of joint rotation, type of bearing materials being lubricated, and environment conditions. Items to consider are as follows:

1. Insure that lubrication can take place by proper location of the grease fitting considering all possible positions of the component and surrounding equipment when installed.
2. Use the right type of grease for the conditions the component is going to operate in. For example, do not use low temperature grease on a joint that is normally subjected to high temperatures. Do not use graphite-loaded grease in high temperature bearing applications because it dries up and interferes with proper bearing function.
3. Do not install lube fittings in areas that will cause the joint to fail due to high stress concentration.
4. Avoid long lubrication paths. The oil in the grease tends to run out and the grease base material dries up clogging the path. This prevents the grease from lubricating the joint.
5. On critical monoball bearing applications provide lubrication for the outside and inside ball diameters. The use of two lube fittings, one on the rod end and the other on the bolt is recommended. One lube fitting can be used provided the groove and four holes on the OD of the ball leading to the ID permit the flow of grease. This would require a more frequent lube cycle to prevent the grease from drying up.

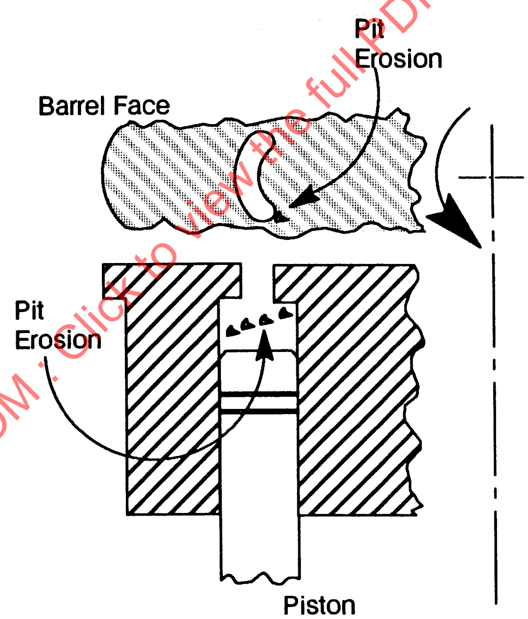
FIGURE 3.2.4.16 - Rotating Joint Lubrication

3.2.5 Pumps: This section contains lessons learned topics relating to hydraulic pumps. The lessons learned are presented in Figures 3.2.5.1 through 3.2.5.15.

PROBLEM: PUMP CAVITATION/EROSION

ISSUE:
1. Implosion/erosion in pumps
2. Low output flow
3. Noise
4. Vibration/excessive wear

ILLUSTRATION:



SOLUTION:
1. Dynamic analysis of inlet filling component
2. Streamline inlet filling (lines)
3. Size inlet lines to match pump requirements (pressure)
4. Contour filling of pump inlet coverage
5. Increase inlet pressure

FIGURE 3.2.5.1 - Pump Cavitation/Erosion

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.

FIGURE 3.2.5.2 - Specification Performance Requirements at Extended (Low and High Speed/Pressure) Conditions

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.

FIGURE 3.2.5.3 - Catastrophic Pump/Motor Failure at Start-Up

PROBLEM: SHORT PUMP LIFE DUE TO LOW INLET PRESSURE

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

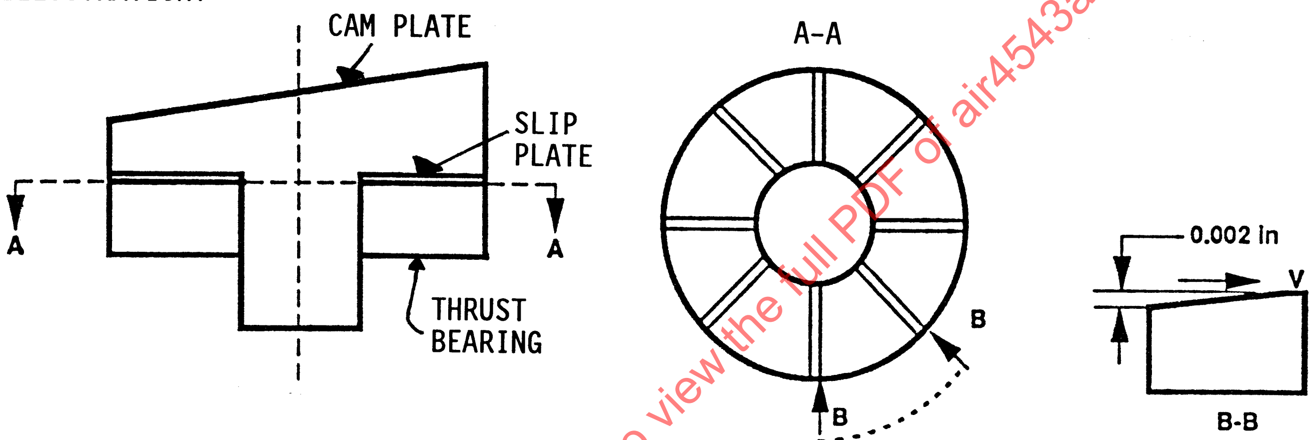
SOLUTION: Reduce pump response/increase reservoir pressure.

FIGURE 3.2.5.4 - Short Pump Life Due to Low Inlet Pressure

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.

FIGURE 3.2.5.5 - Early Pump Failures Due to Rapid Air-Turbine Starts of the B-52 Hydraulic Packs (Ten Per Shipset)

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

1. Setting one pump compensator (slave) 200 psi below the other (master) pump

2. Incorporating a 50 to 200 psi check valve on one pump discharge line (slave) to let the other dominate until higher flow by system is demanded of the slave pump

FIGURE 3.2.5.6 - Force Fight Between Two Pumps in the Same System

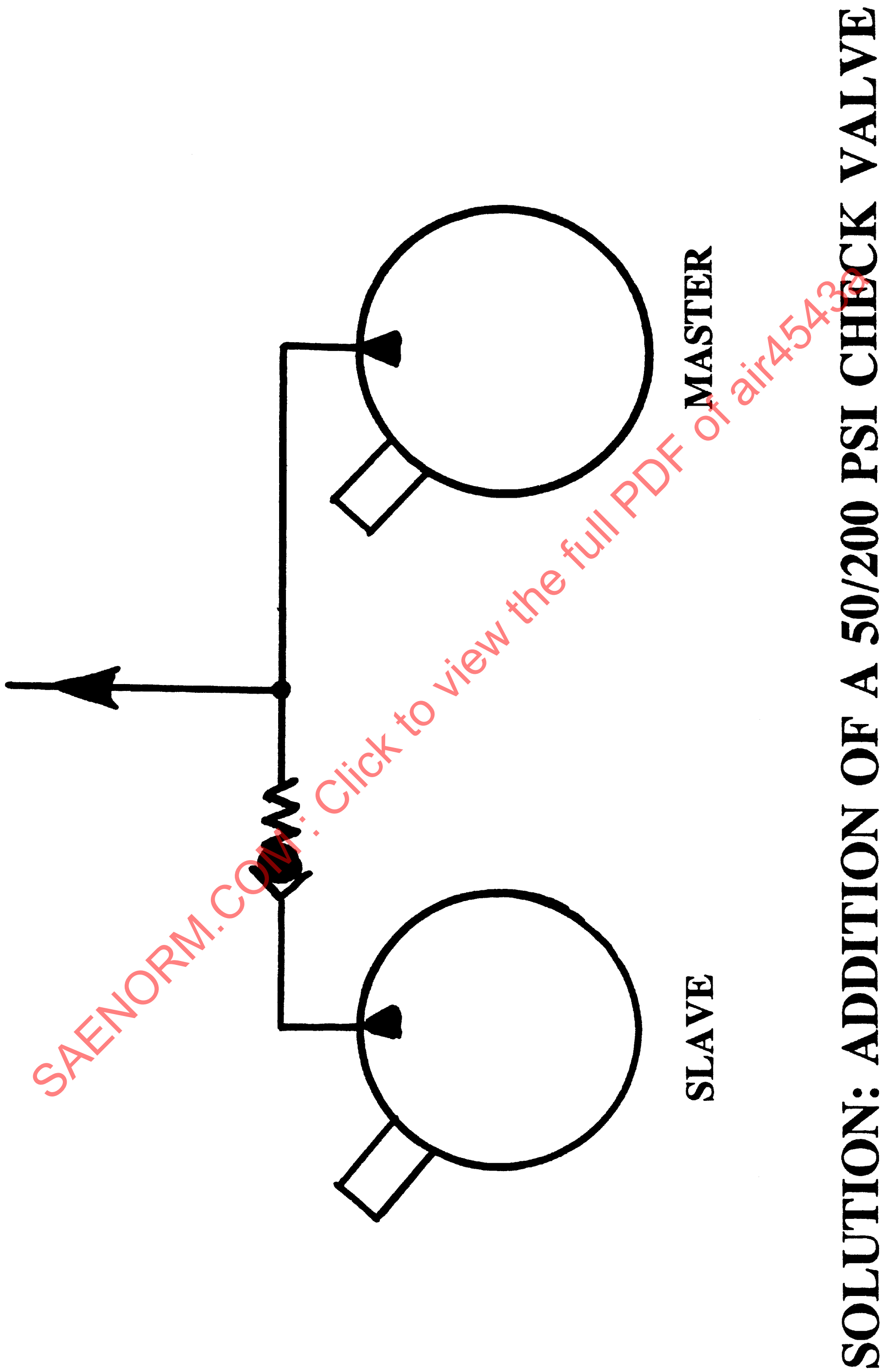


FIGURE 3.2.5.6 (Continued)

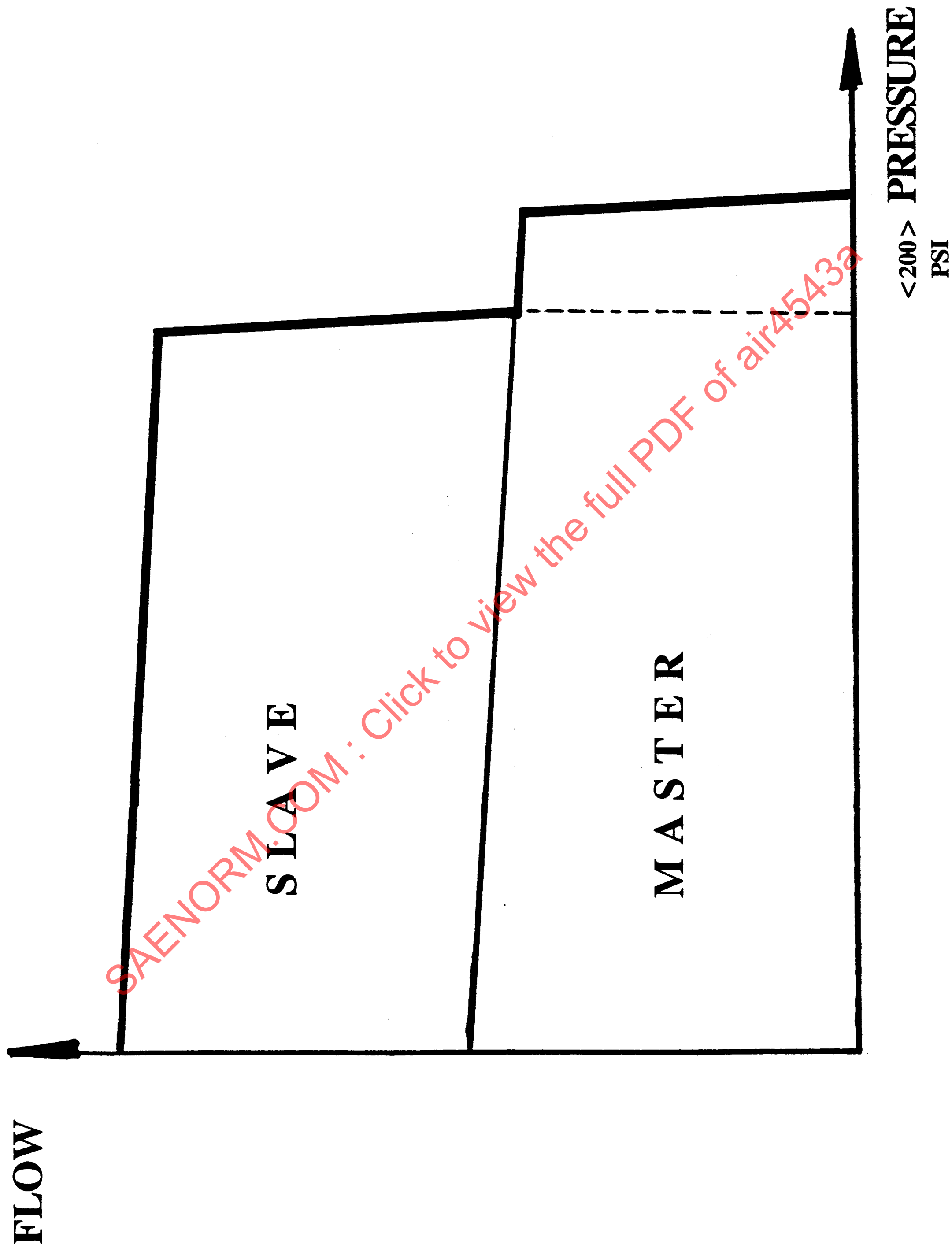


FIGURE 3.2.5.6 (Continued)

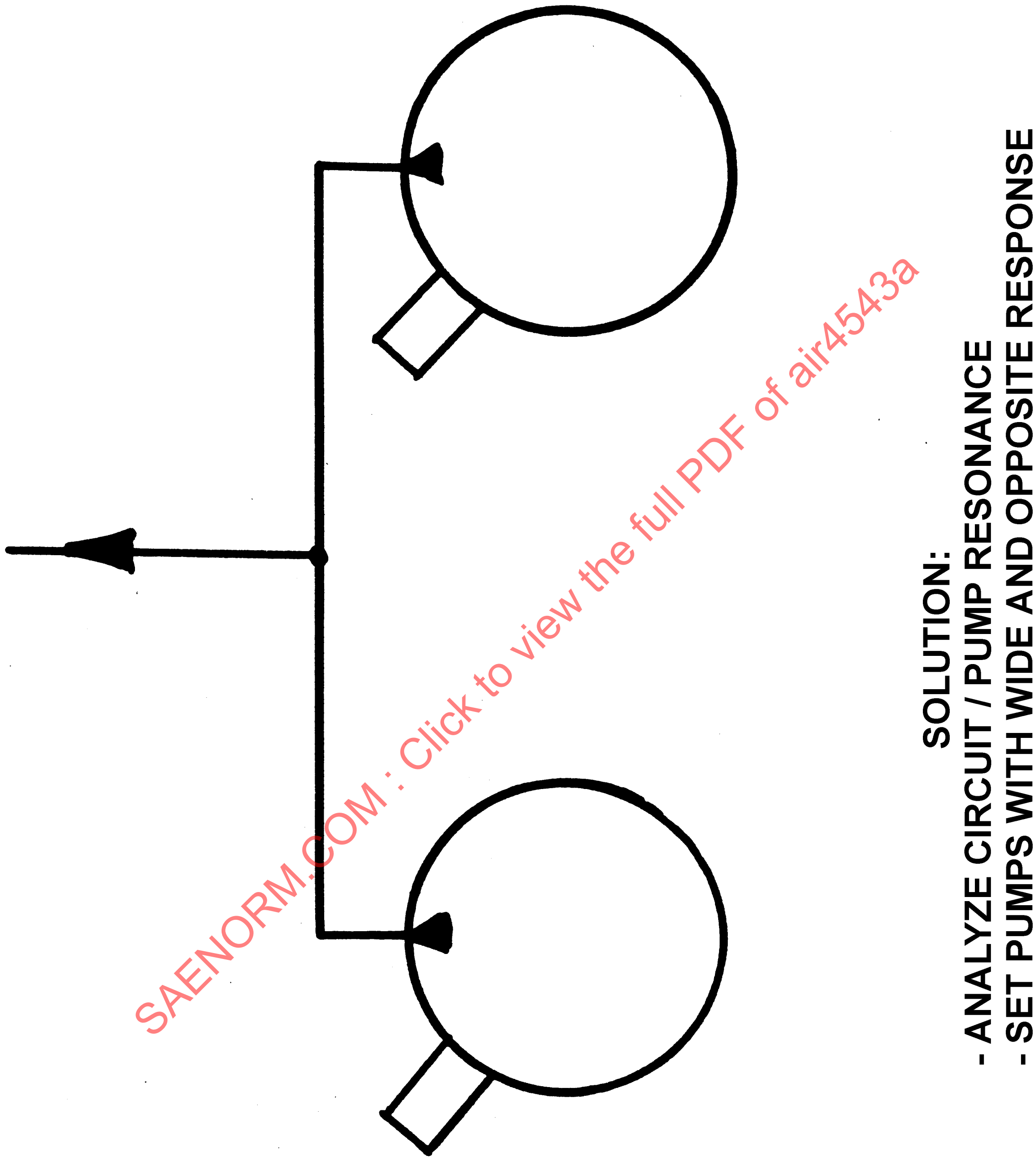


FIGURE 3.2.5.6 (Continued)

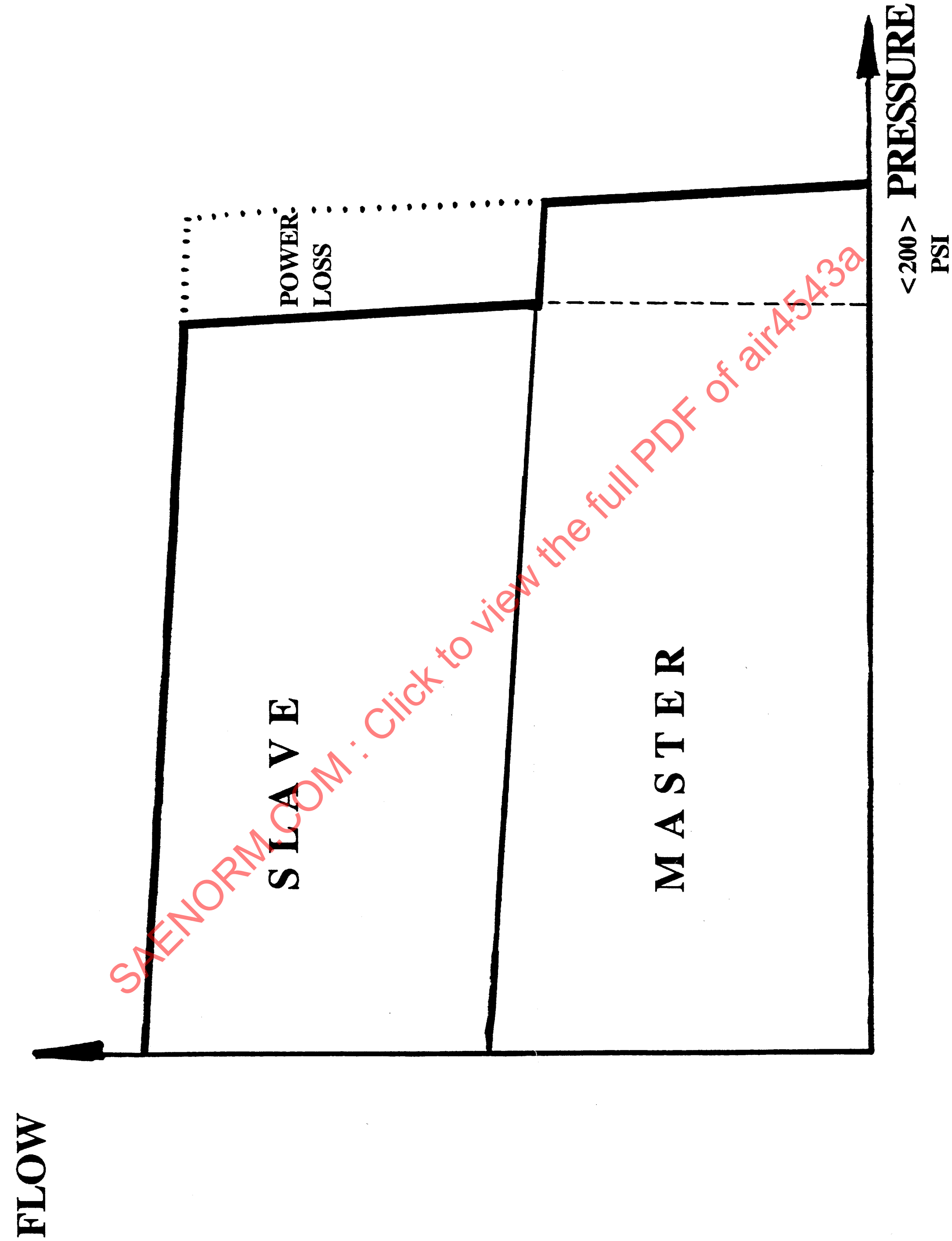


FIGURE 3.2.5.6 (Continued)

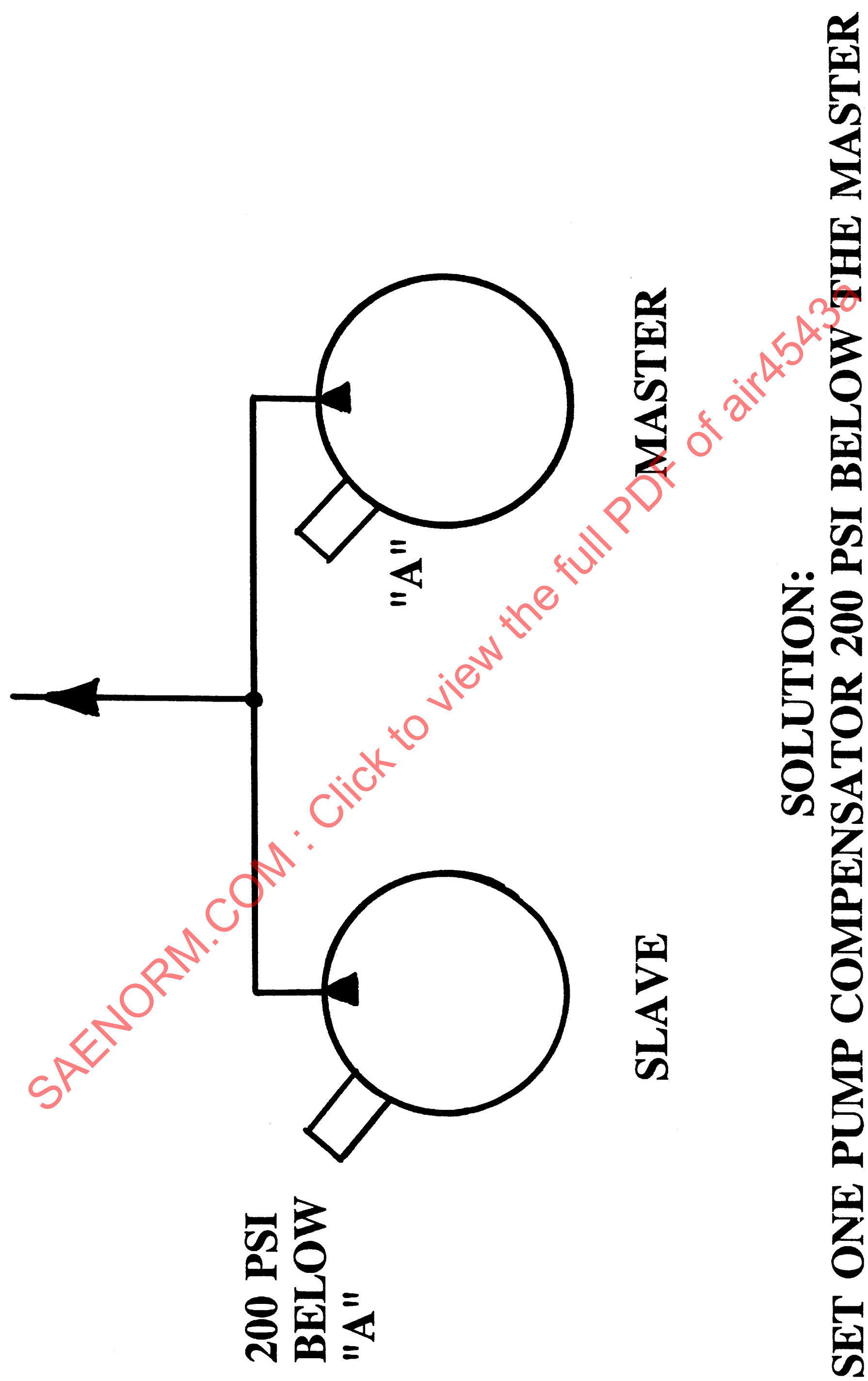
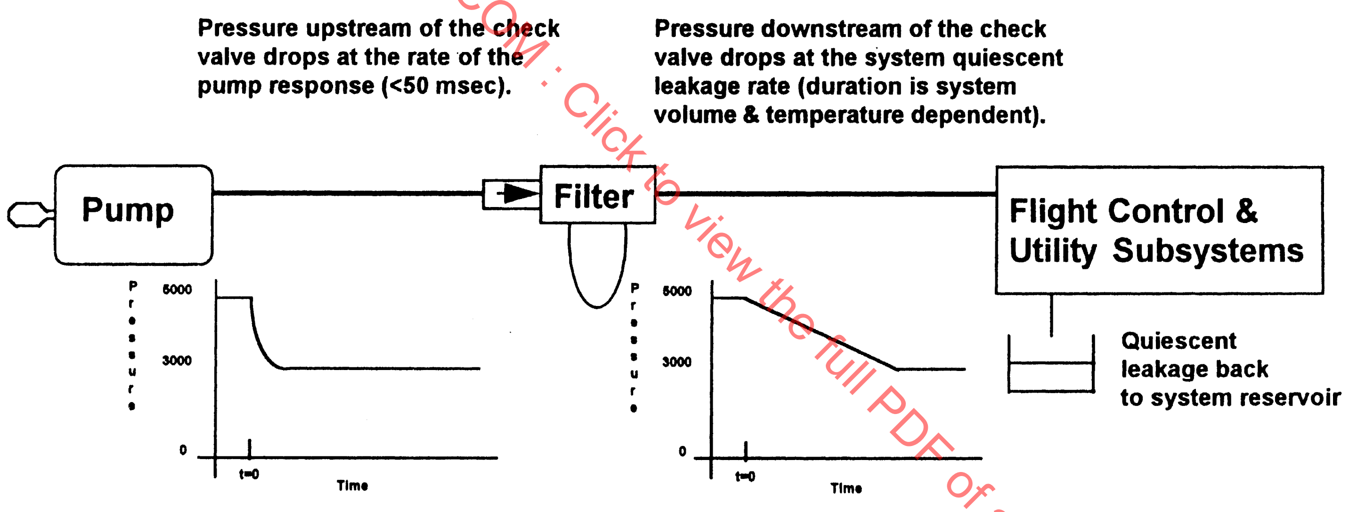


FIGURE 3.2.5.6 (Continued)

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.



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

FIGURE 3.2.5.7 - Dual Pressure Pump Instability

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.

- Discharge Pressure
- Shaft Speed
- Displacement
- Fluid Buld Modulus (Temperature, Air Content)

SOLUTION:

- 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).

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

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

FIGURE 3.2.5.8 - Timing of Dual Pressure Pumps

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-H-5606, MIL-PRF-83282, and MIL-PRF-87257 do not have this problem as their additives are different.

FIGURE 3.2.5.9 - Pump Performance With Phosphate Ester Fluid

PROBLEM: A FIGHTER AIRCRAFT EXPERIENCED 30 IN-FLIGHT FAILURES A YEAR, 8% OF TOTAL PUMP REMOVALS

ISSUE: 50% 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.

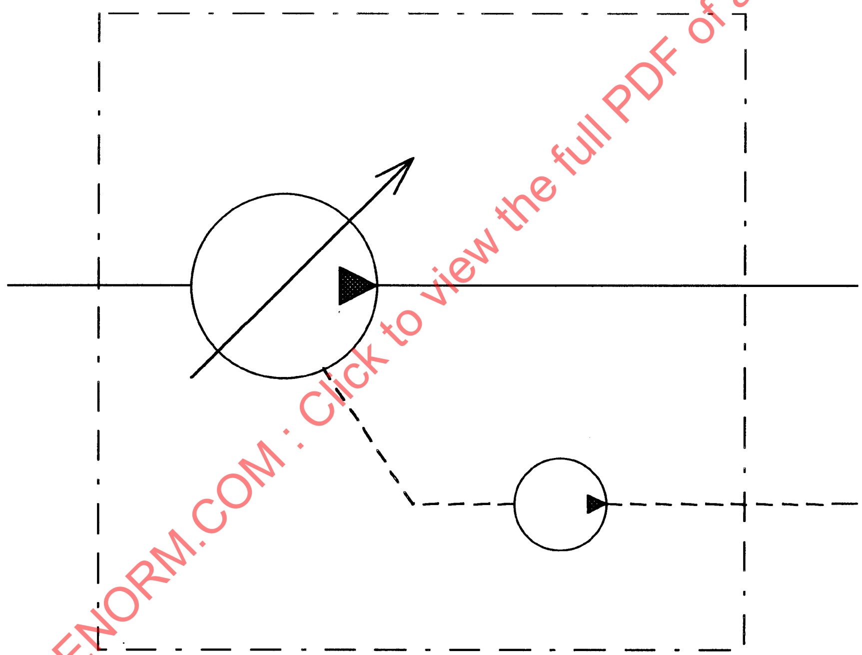
Check valves in case drain manifold removed to avoid jamming and back pressure on pump.

FIGURE 3.2.5.10 - Hydraulic Pump Fires

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.

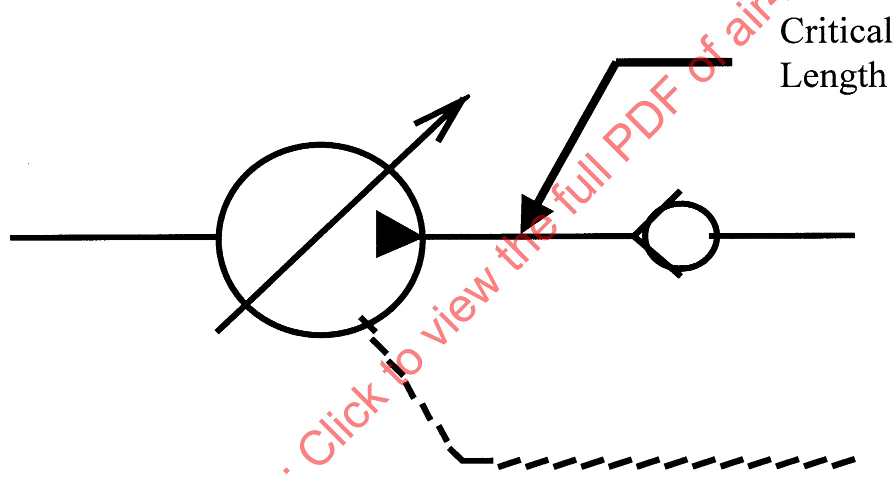
a. Improved heat rejection
b. Insulate pump case from back pressure
    1. Reduced pressure loading
    2. Improved fatigue life
    3. Steady outlet pressure
c. Reduced case temperature
d. Improved reliability

FIGURE 3.2.5.11 - Pump Case Scavenge Device

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:

1. Locate check valve as far away from pump outlet as possible.
   a. Improves stability
   b. Allows for lower internal leakage
   c. Improves performance
   d. Increases life
2. If check valve must be located close to pump, then special design considerations are required.
   a. Yoke over stroke angle
   b. Compensator spool clearances
   c. Compensator spool overlap
   d. Compensator spring rate
3. Check valve location and associated system volumes must be known early in pump design process to prevent problems and delays later in the program.

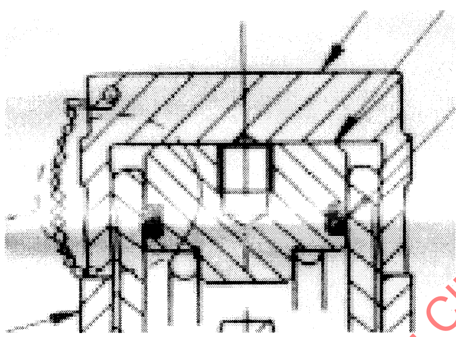
FIGURE 3.2.5.12 - Pump Instability Caused by Check Valve

PROBLEM: REQUIREMENT TO PASS A 5 min, 2000 °F, FLAME TEST WAS ADDED AFTER PUMP WAS DESIGNED

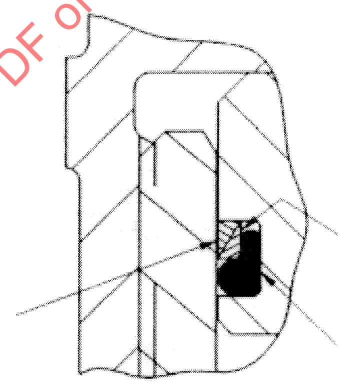
ISSUE: Because of the orientation of the pump in the aircraft, the compensator valve would be directly exposed to the flame. Seal damage and resulting external leakage would result.

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:

- Least cost approach
- No additional envelope required
- No added weight
- Easy upgrade of fielded pumps

FIGURE 3.2.5.13 - Pump Fire Test

PROBLEM: BRONZE CYLINDER BLOCK OF A 5000 psi HYDRAULIC MOTOR CRACKED AFTER A LOW NUMBER OF FLIGHT HOURS

ISSUE:

1. Cracks in a solid bronze hydraulic motor used at 5000 psi were determined to be from fatigue. Time in service was 25 h.
2. FEA analysis determined that the bronze material did not have adequate fatigue endurance properties for 5000 psi.
3. Qualification tests on three bronze cylinders of 500 h indicated successful operation of bronze cylinder block.
4. Design of cylinder block of motor was changed to steel block with bronze piston bore liners.
5. Similar cracking was experienced with an 8000 psi pump by the Navy in 1977 (bronze cylinder block design).
6. Solid bronze cylinder blocks have been used up to 4000 psi

SOLUTION: Use steel cylinder blocks with bronze cylinder bore liners for hydraulic pumps and motors at 5000 psi or higher pressures.

FIGURE 3.2.5.14 - Bronze Pump and Motor Cylinder Blocks

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.

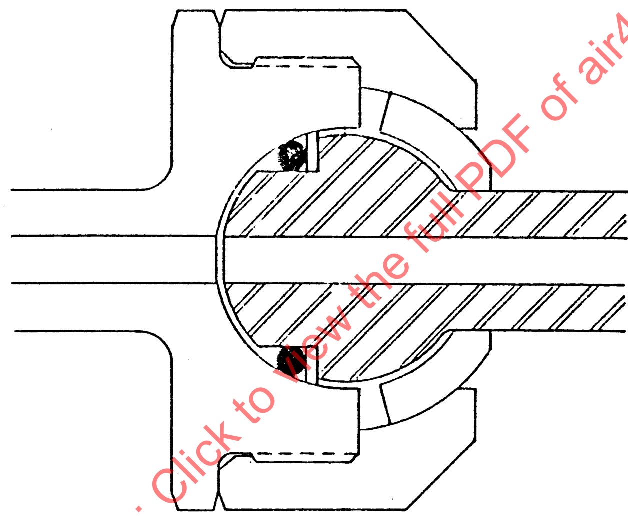
FIGURE 3.2.5.15 - Pump Gearbox Installation - Shaft Seal Leakage

3.2.6 Seals: This section contains lessons learned topics relating to hydraulic seals and seal design. The lessons learned are presented in Figures 3.2.6.1 through 3.2.6.26.

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:

1. Use coiled tube rather than spherical swivels, if possible.
2. Increase seal squeeze, and reduce ball to housing clearance as much as possible in the design of the swivel.

FIGURE 3.2.6.1 - Spherical Ball Swivel Leakage

PROBLEM: SERVOACTUATOR PISTON SEAL BLOWBY

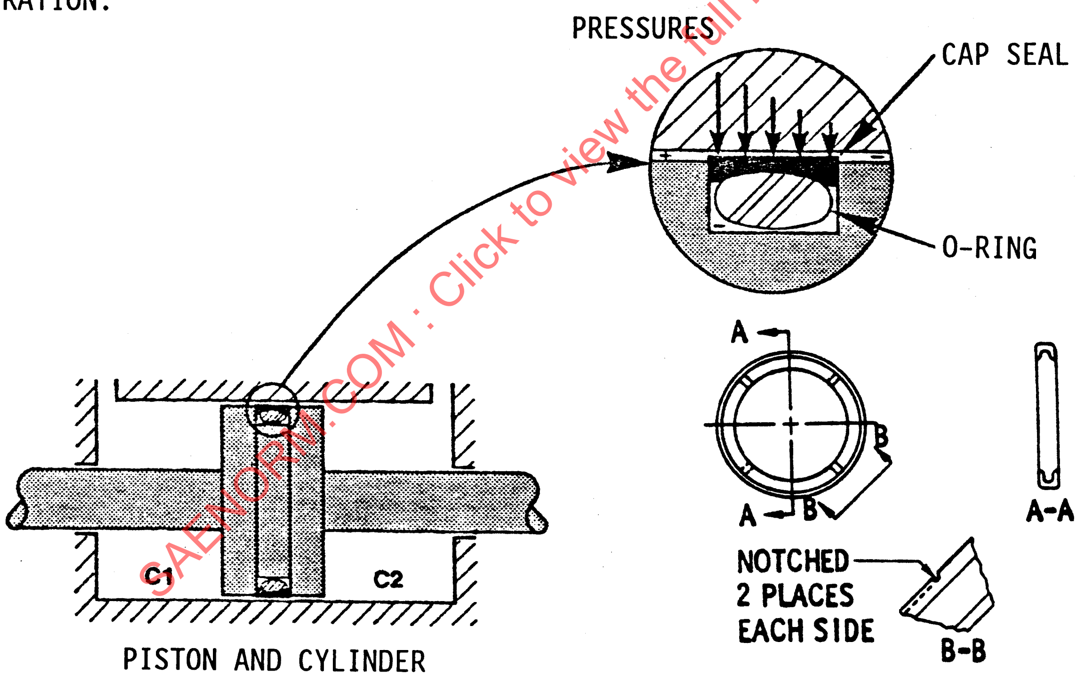
ISSUE:

1. 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.

2. 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.

3. 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.

FIGURE 3.2.6.2 - Servoactuator Piston Seal Blowby

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. Nonvented designs offer a smaller envelope size and reduced complexity. Primary seals in unvent applications must have an anti-extrusion device on both sides.

FIGURE 3.2.6.3 - Tight Rod Seal Leakage Requirements Are Difficult to Achieve Using a Single 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.

[1] SEM (scanning electron microscope)

FIGURE 3.2.6.4 - Rapid Seal Wear of a PTFE Piston Head Seal

PROBLEM: USE OF SMALL CROSS-SECTION O-RINGS (LESS THAN 0.103 IN CROSS-SECTION DIAMETER)

ISSUE: Nitrile O-rings with a cross-section diameter that is less than 0.103 in, 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 or larger.

Problem	Affect
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 has inadequate seal squeeze and inadequate groove width on some several glands.	Leaks, groove overfill.

SOLUTION: Avoid the use of small cross-section seals. It may be possible to use a -102 through -109 O-ring (0.103 in 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 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.

FIGURE 3.2.6.5 - Use of Small Cross-Section O-rings
(Less Than 0.103 in Cross-Section Diameter)

PROBLEM: USE OF SMALL CROSS-SECTION O-RINGS

ISSUE: Small cross-section static seals, typically 0.070 in diameter, per MS28775 or MIL-P-83461 (nitrile) 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 (see Illustration 1).

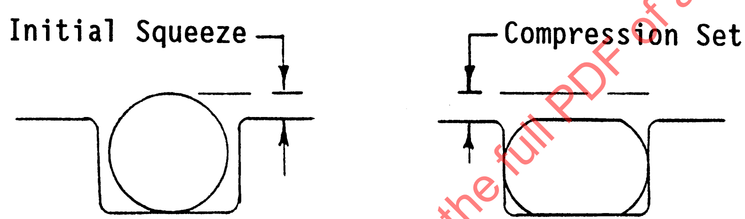


ILLUSTRATION 1

SOLUTION: 1. Where external seepage would result, avoid small cross-section seals, especially for sizes above a 2 in diameter. Where they must be used, the elastomer compound should be changed:

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

b. Phosphonitrilic fluoroelastomer (PNF) per MIL-P-87175 has been successful solving problems where squeeze must be maintained down to -65°F.

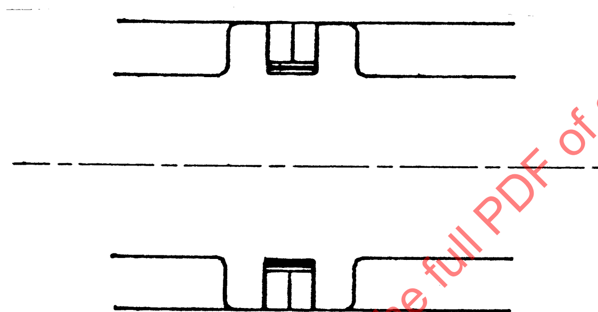
c. Fluorosilicone per MIL-R-25988 may prove to be an acceptable lower cost alternate to PNF in certain applications such as face seals.

FIGURE 3.2.6.6 - Use of Small Cross-Section O-rings

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:

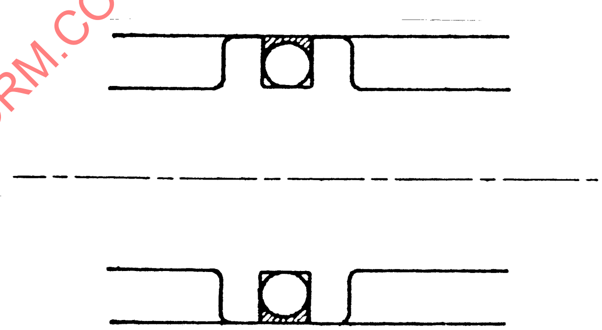


FIGURE 3.2.6.7 - Small Actuator Piston Ring Leakage Blowby

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.

FIGURE 3.2.6.8 - Deterioration of O-rings Exposed to Different Fluids, e.g., Pump Coupling Shaft in Flooded Gearbox

PROBLEM: FLUID LEAKAGE IN GUIDED MISSILES

ISSUE:
1. Small volumes - no leakage tolerable
2. -65°F to +275°F storage and operation
3. Immediate full performance required
4. No warm-up
5. MIL-P-25732 nitrile seals found lacking due to permanent set at +275°F

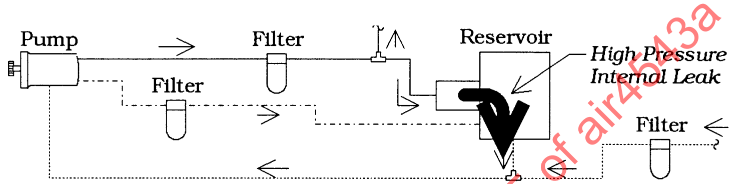
SOLUTION:
1. Use fluorosilicone O-rings.
2. Shore 70 minimum for toughness and wear.

FIGURE 3.2.6.9 - Fluid Leakage in Guided Missiles

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.**

ILLUSTRATION:



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

FIGURE 3.2.6.10 - Reservoir High Pressure Seal Leakage

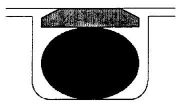
PROBLEM: WEAR ON SOFT ANODIZED BORES BY UNFILLED MODIFIED PTFE SEALS

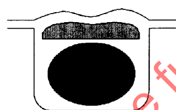
ISSUE: Unfilled modified PTFE seals subjected to large numbers of cycles.

Loose sealing integrity due to:

- Wear of soft anodized aluminum bores
- Wear on energizing O-rings
- Wear of PTFE cap strips

ILLUSTRATION:





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

- Hard anodized or nickel plated bores
- Mineral filled PTFE seals, non-abrasive
- Viton O-rings (if temperature hardening is evident)

FIGURE 3.2.6.11 - Wear of Aluminum Bore by PTFE Seals

PROBLEM: FAILURE OF NYLON LOW LEAKAGE POPPET SHUT-OFF VALVES

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

- Embrittlement from overheating
- Moisture degradation
- Accumulated temperature cycles
- Unexpectedly high frequency of operation
- Combinations of all of the above

Results in:

- Failure to stop flow when poppet is activated
- Migration of seat material to downstream components

SOLUTION: Step 1 - Assure that life requirements adequately cover service usage

- Temperature range
- Cycles of operation
- 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

FIGURE 3.2.6.12 - Failure of Nylon Poppet Value Seat

PROBLEM: HYDRAULIC SEAL FAILURE AND LEAKAGE CAUSED BY HIGH SYSTEM TEMPERATURES

ISSUE:

1. A major fighter aircraft was experiencing O-ring, cap strip seals, and poppet seat failures at about 2000 flt-hrs into a 6000 flt-hrs useful aircraft life.

2. Higher than normal temperatures, up to 260 °F, caused by degrading components was suspected as the major cause.

3. Qualification test procedure review indicated no endurance or impulse test cycling above 225 °F. MIL-H-8775 specifies 275 °F or tests at expected temperatures.

SOLUTION:

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

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

3. In known hot areas, specify high temperature seals, i.e. MIL-R-83485 (Viton).

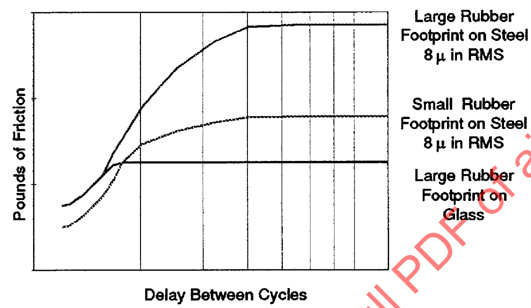
4. Tests at 275 °F have revealed two seal failures on a new aircraft program.

FIGURE 3.2.6.13 - Seal Degradation from High System Temperatures

PROBLEM: PUMP CAVITATION ON START-UP

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.

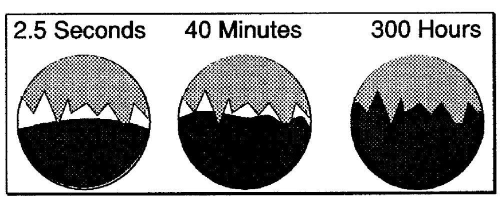
Loss of pressurization was traced to the sticking of the reservoir piston rubber O-ring seal on the reservoir housing.



The reservoir housing material is aluminum with hardcoat (Type 3) anodize. The hardcoat 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.

Hardcoat anodize and a large rubber footprint is an unfavorable combination. The rubber extrudes into the voids and crevices of the hardcoat. 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.



SOLUTION: Be cognizant of the micro-surface, its relationship with seals and select seals and finishes appropriately. For this particular application, a new loss 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.

FIGURE 3.2.6.14 - Stiction of a Rubber Seal on a Reservoir Piston Caused Pump Cavitation

PROBLEM: LEAKAGE OF FACE SEALS

ISSUE: When to use face seals versus quill 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 psi) can result in a weight penalty, especially on multi-port installations.

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

F18 E/F had to use quills because of weight considerations and 5000 psi pressure level.

FIGURE 3.2.6.15 - Use of Face Seals or Quills (Transfer Tubes) at Hydraulic Interfaces

PROBLEM: DYNAMIC SEALS LEAKAGE DUE TO HYDRAULIC PUMP RIPPLE

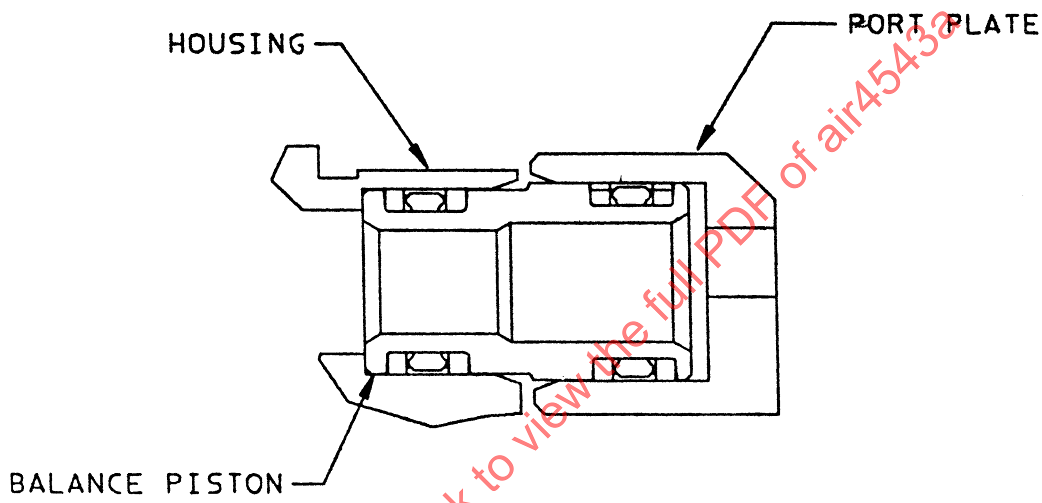
ISSUE: Dynamic seals can be damaged when exposed to hydraulic pump ripple. The elastomer is typically eaten away from under the cap.

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-H-5606 or MIL-H-83282 is Viton GLT per MIL-R-83485.

FIGURE 3.2.6.16 - Damage to Seals from Hydraulic Pump Ripple

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.



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

FIGURE 3.2.6.17 - Unbalanced Transfer Tube to Eliminate Seal Wear

PROBLEM: RESERVOIR RELIEF 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 face seal and teflon coated seat demonstrated greatly increased life at 275 °F compared with nitrile face seal and anodized aluminum seat.

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

FIGURE 3.2.6.18 - Reservoir Relief Valve Poppet Seal Failures

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 backrup rings.

FIGURE 3.2.6.19 - Uncut Back-Up Rings Cause Pressure Trap

PROBLEM:	HAD STATIC O-RING SEAL LEAKAGE AT -65 °F EVEN THOUGH SQUEEZE WAS GREATER THAN CALLED FOR BY MIL-G-5514F (5.36%/0.011 INCH MINIMUM SQUEEZE)
ISSUE:	Leakage occurred at 4,000 psi when barrel would be expected to breathe by approximately 0.001. Parker Handbook says high temperature nitrile good to -20 °F.
SOLUTION:	Changed to DAA3122D123 seal wherein cross section was enlarged by .010 inch. Minimum squeeze now 9.71% or 0.021 inch passed test.

FIGURE 3.2.6.20 - Nitrile O-ring Leakage at -65 °F

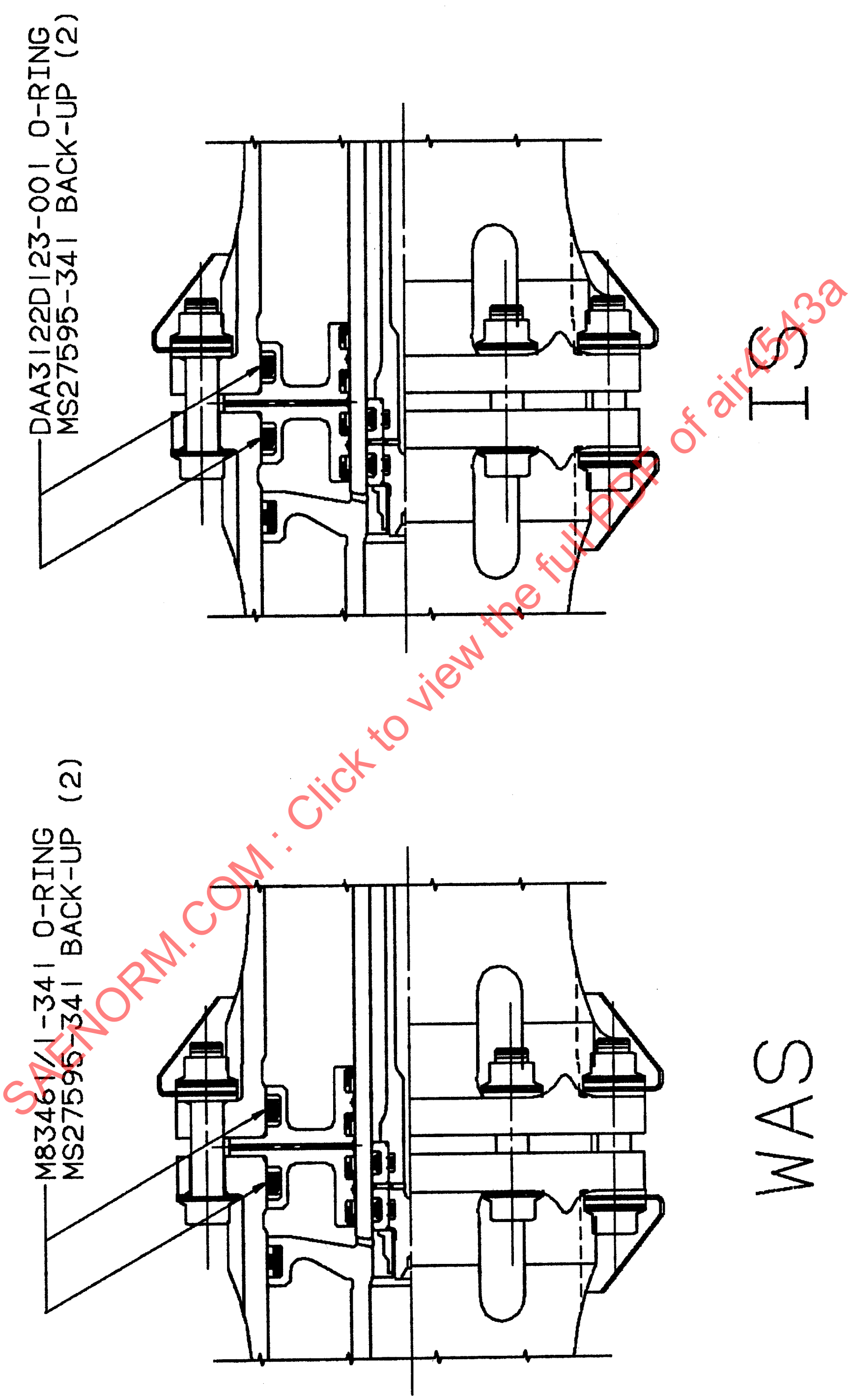


FIGURE 3.2.6.20 (Continued)

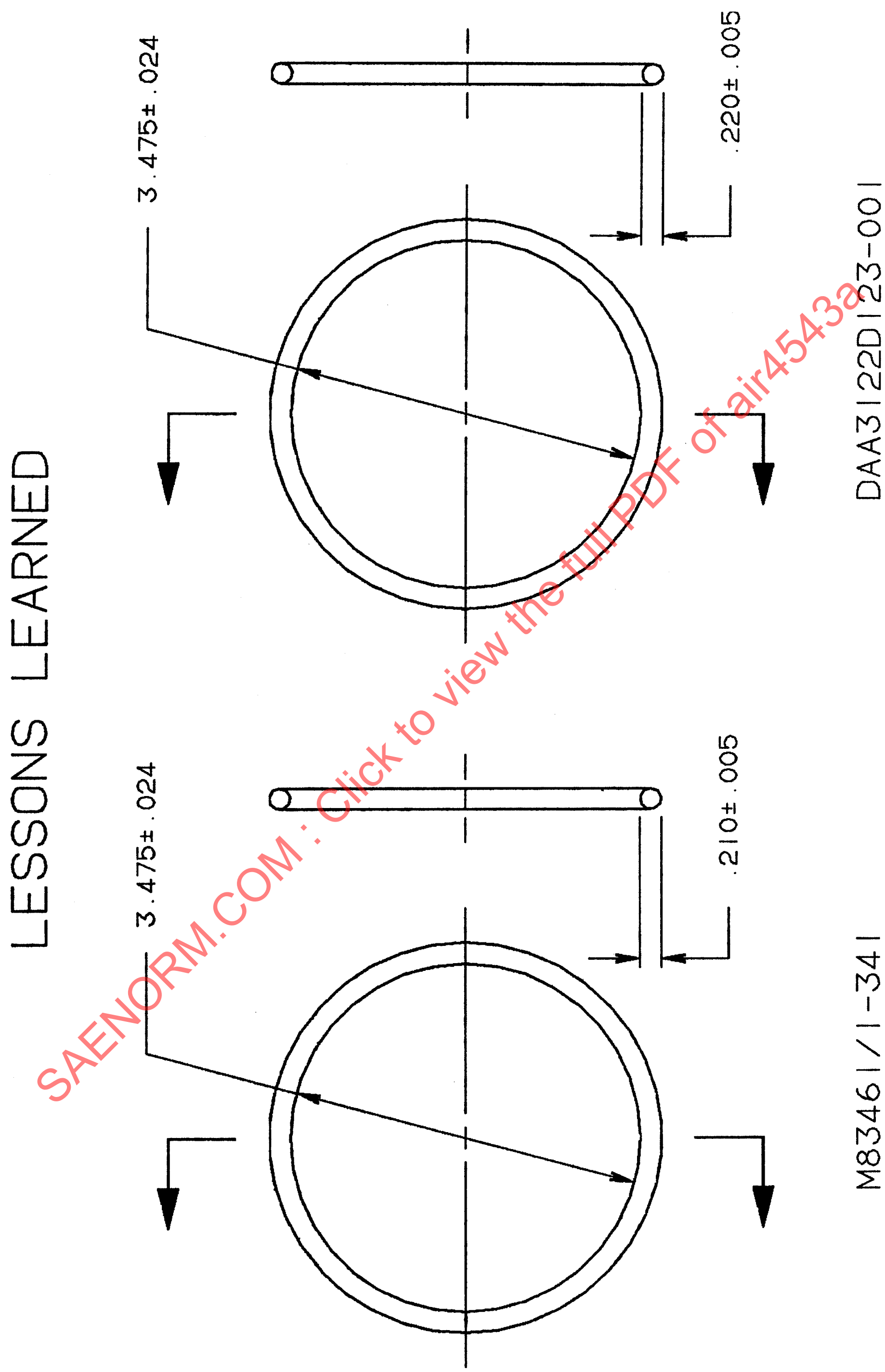
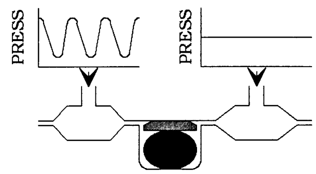


FIGURE 3.2.6.20 (Continued)

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 misphased oscillating pressures of similar magnitude can be expected.

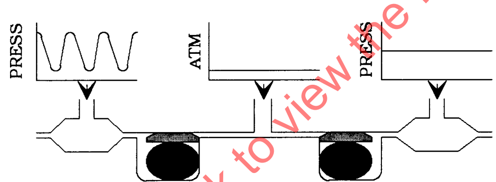
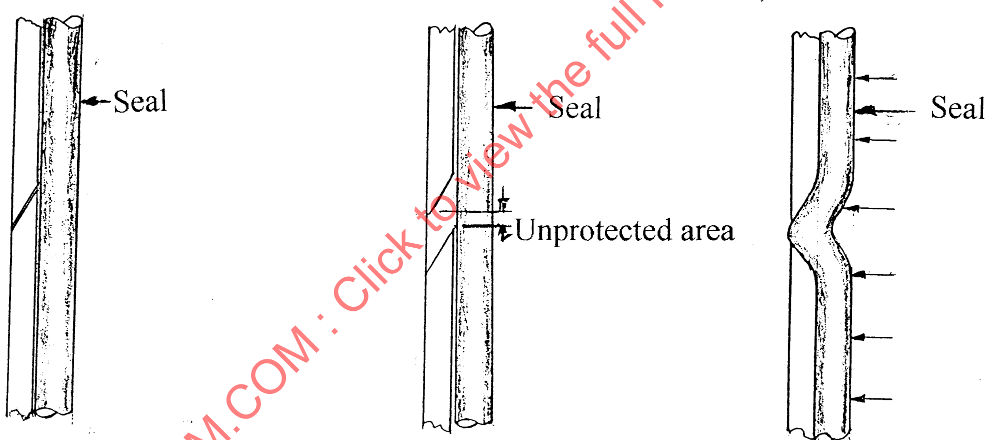


FIGURE 3.2.6.21 - Seal Wear and Intersystem Leakage

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 bores and larger sizes), 0.052/0.045 in thickness MS28774 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 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:



Installation

Installation plus cold soak while unpressurized

Pressurized seal deforms tang and is subject to local nibbling or blowout.

SOLUTION:

1. Use uncut Teflon backups for larger bore sizes. Do not use MS28774 which are not designed for current gland standards.
2. An applied axial seal pressure of approximately 100 psi 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.

FIGURE 3.2.6.22 - Seal Nibbling Caused by Large Scarf Cut Backup Ring

PROBLEM: UNCUT PTFE BACKUP RINGS PER MIL-G-5514 ACT AS A SEAL AND TRAP PRESSURE WHEN AS4716 SEAL GROOVE IS USED. HIGHER PRESSURES, 5000 psi, MAY EXACERBATE THIS PROBLEM.

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:

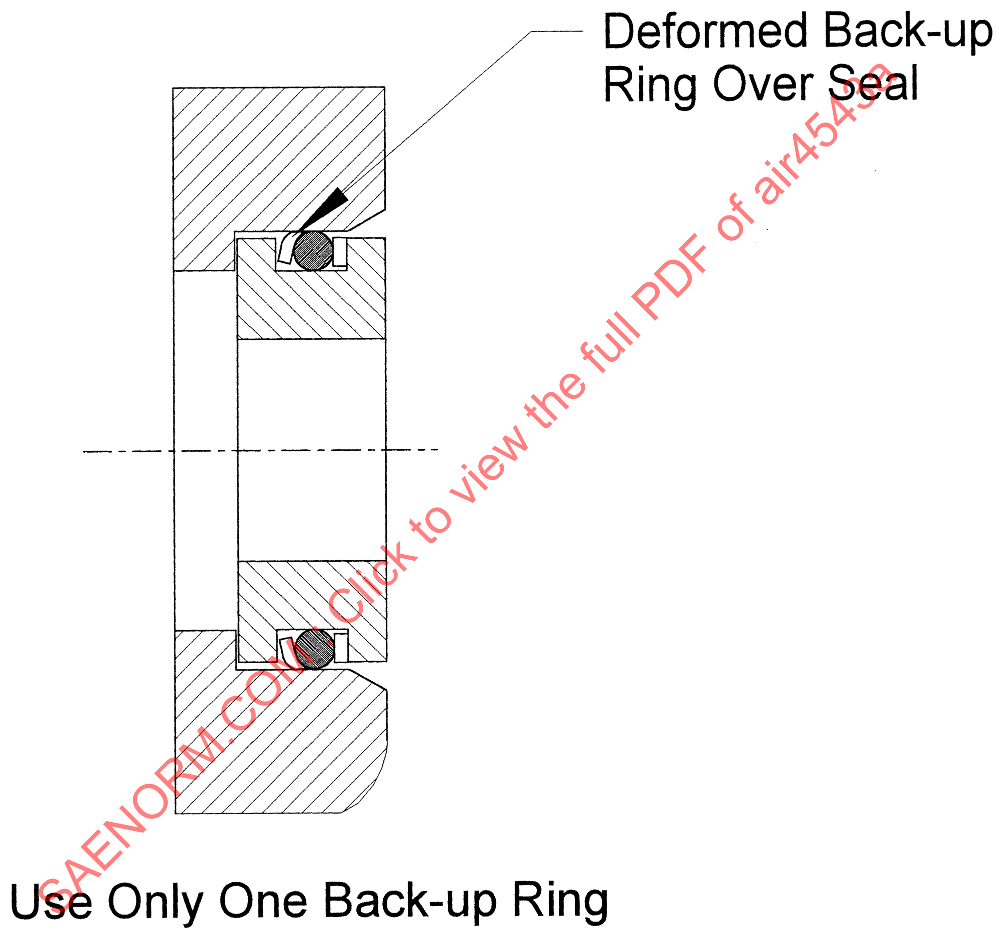
1. On aluminum plug, redesign to MIL-G-5514 groove with scarf cut backup rings.
2. On static seals on spool/sleeve assembly, redesign to AS4716 solid delta backups with pressure relieving notches. Also notched sleeves on several designs.

FIGURE 3.2.6.23 - Pressure Traps Caused by PTFE Backup Rings in AS4716 Seal Grooves

PROBLEM: BACKUP RING DEFORMATIONN DURING ASSEMBLY

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

ILLUSTRATION:



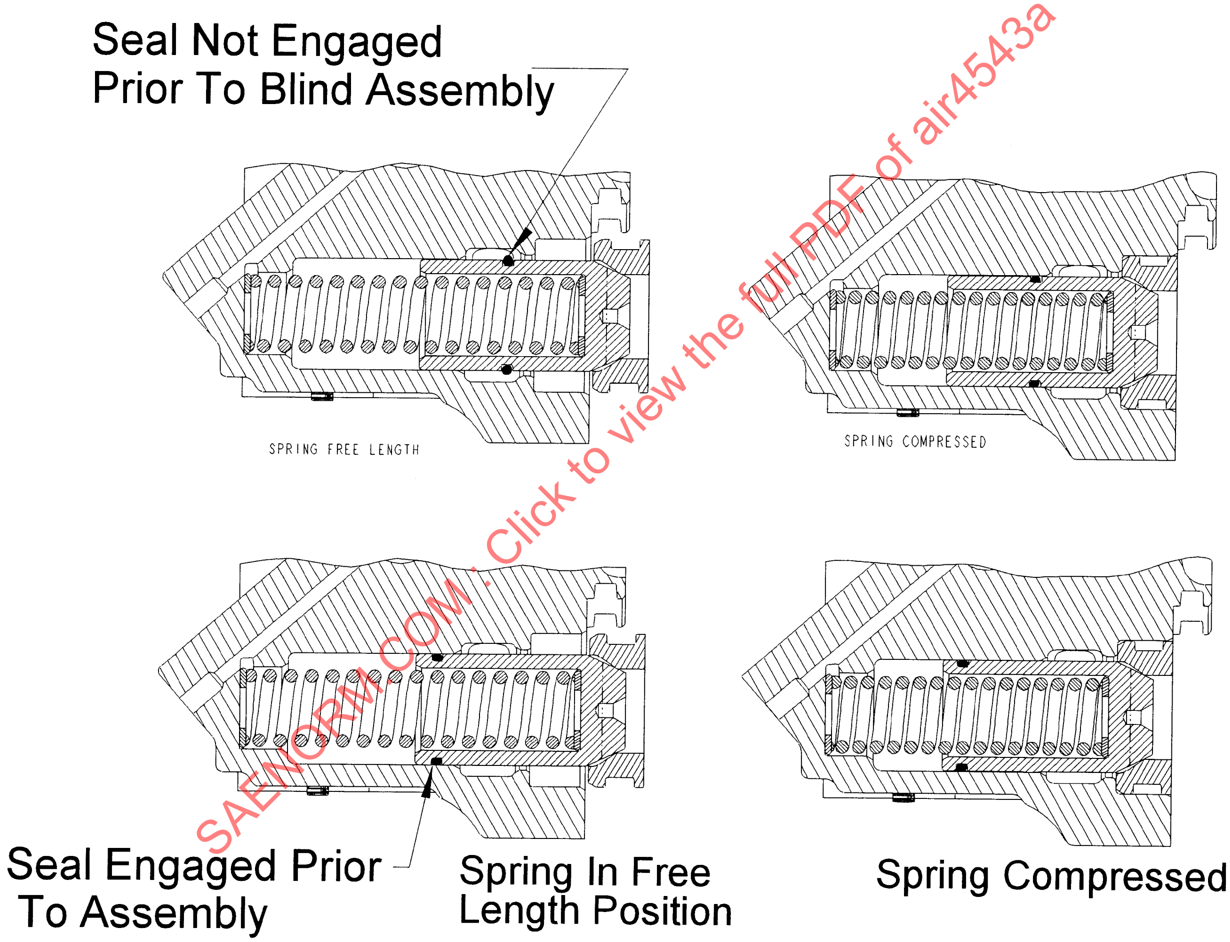
SOLUTION: Change design to eliminate upstream backup, i.e., single backup design which has short insertion of backup ring.

FIGURE 3.2.6.24 - Damaged Backup Ring

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:



Engagement of Seals Should Occur Prior to Assembly

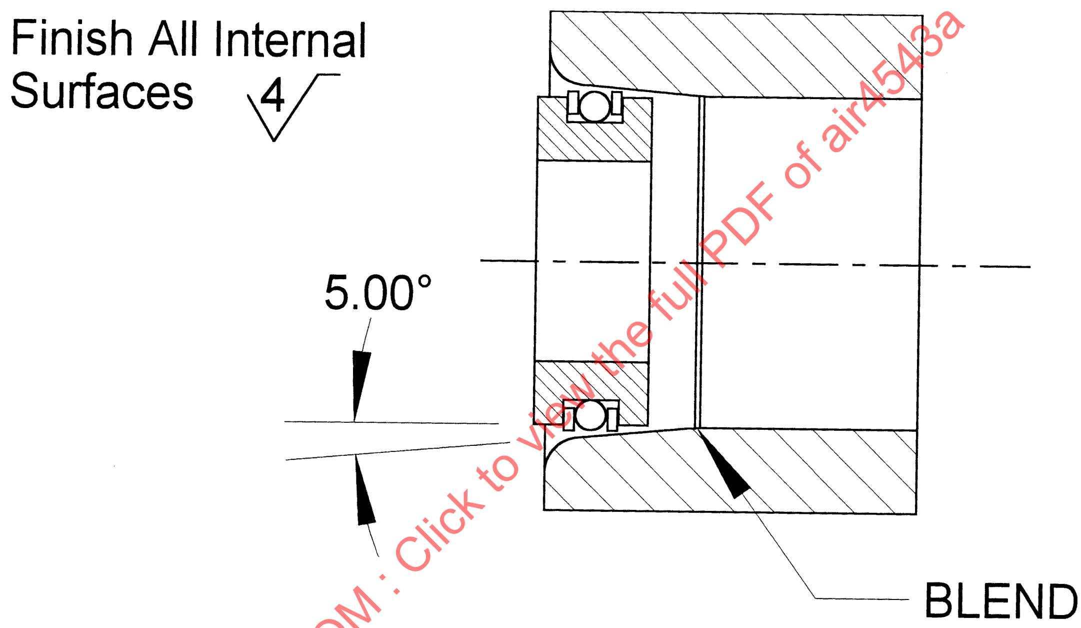
SOLUTION: Design components so that seals engage bores prior to assembly of mating components, i.e., compressing springs. Assembly is simplified and avoids damage to seals.

FIGURE 3.2.6.25 - Seal Engagement

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:



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

FIGURE 3.2.6.26 - Compressing Solid Backup Rings

3.2.7 Tubing, Fittings, Bosses, and Hoses: This section contains lessons learned topics relating to hydraulic tubing, fittings, bosses, and hoses. The lessons learned are presented in Figures 3.2.7.1 through 3.2.7.13.

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 ocurred 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.

FIGURE 3.2.7.1 - Hydraulic Tubing Failures Due to Excessive Ovality in the Bends

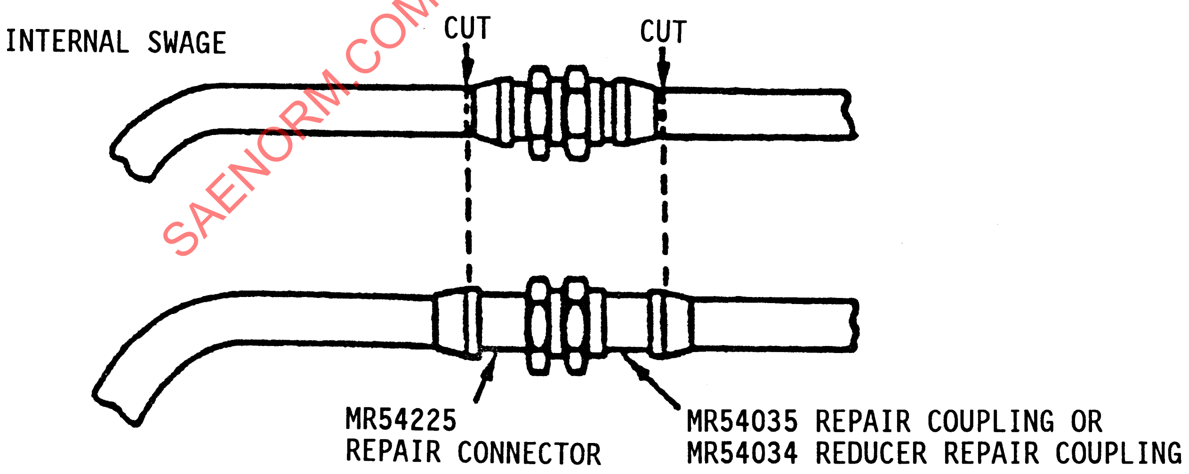
PROBLEM: LEAKING FITTINGS

ISSUE:

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

2. 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.

FIGURE 3.2.7.2 - Leaking Fittings

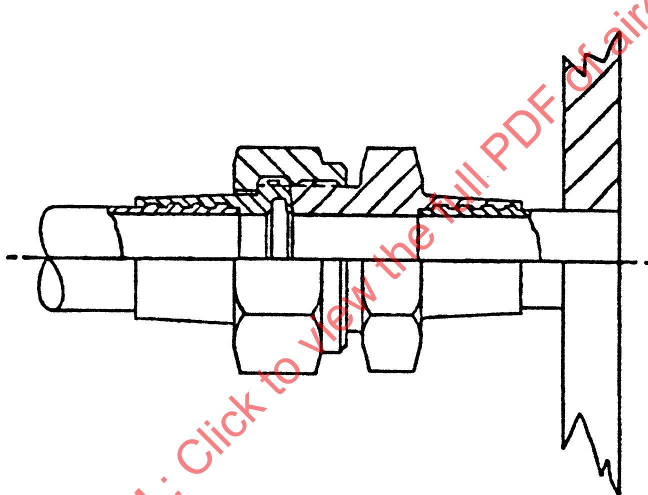
PROBLEM: MALE/FEMALE LIP SEAL CONNECTION DAMAGE

ISSUE:

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

2. 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.

FIGURE 3.2.7.3 - Male/Female Lip Seal Connection Damage

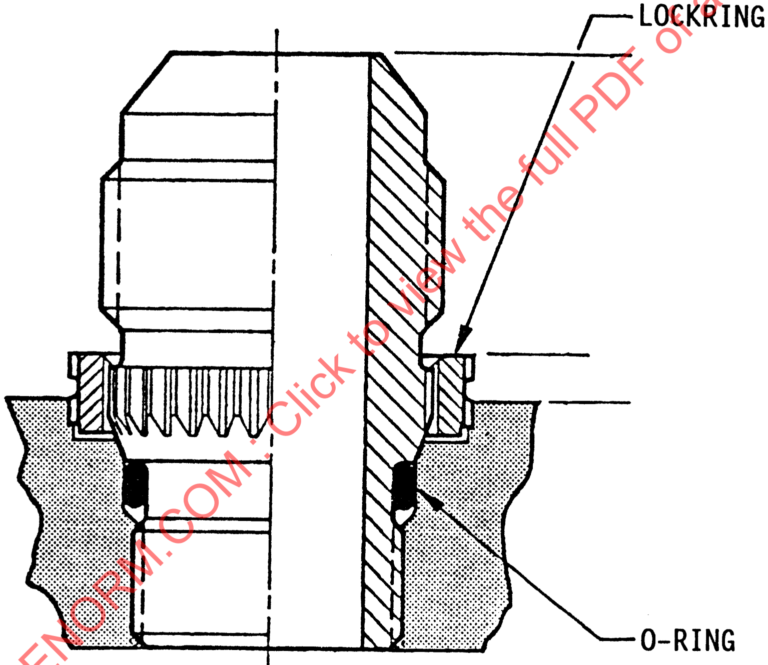
PROBLEM: DAMAGED PORTS

ISSUE:

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

2. 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.

FIGURE 3.2.7.4 - Damaged Ports

PROBLEM: TEFLON COVERED TUBE CLAMP

ISSUE:

1. Subject to burnishing, chaffing, and uneven wear leading to pin hole or large wear hole in tube.
2. Teflon cover is only 0.020 in 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:

1. Add thick additional teflon or plastic grommet between clamp and tube to provide longer wear.
2. Replace the loop clamp with a two bolt plastic clamp block.
3. Change the clamp to one that has a cover of nitrile butadiene rubber, 65-75 durometer with a minimum thickness of 0.060 in.

FIGURE 3.2.7.5 - Teflon Covered Tube Clamp

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. Industry Standards would permit a length of 48.000 in ± .500 in.

Hose assembly specifications allow hose elongation or contraction while under operating pressure to typically vary ±2% (±.96 in 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:

±.125 in for lengths under 18 in
±.250 in for lengths from 18 to 36 in exclusive
±.500 in for lengths from 36 to 50 in exclusive
±1% for lengths of 50 in and over

FIGURE 3.2.7.6 - Hose Assembly Tolerances

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:

- Hose stiffness can make installations difficult due to the force required to bend the hose.
- For lower pressure hose assemblies with less reinforcement, kinking is possible.
- 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.

FIGURE 3.2.7.7 - Short Hose Assemblies

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 unmanufacturable designs requiring exceptions to be taken or drawing revisions to be made.

EXAMPLE OF TYPICAL PAGE TOLERANCES:

Decimals	.X in	±.1 in
	.XX in	±.02 in
	.XXX in	±.010 in
Angles		±0°30'

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.

Example of Tube Tolerancing:

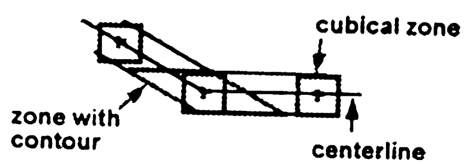


FIGURE 3.2.7.8 - Hose Drawing Tolerances

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 ±.035 in.

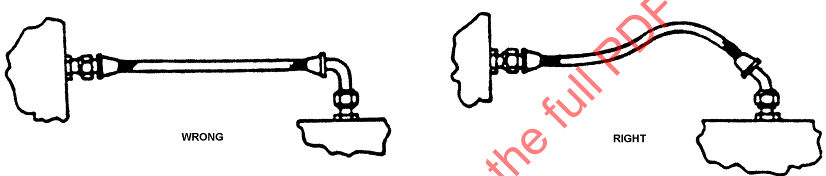
For multi-bend elbows (which often have charted coordinates), specify the coordinates as BASIC dimensions and call out a contour tolerance for the bent tube region. Typical tolerances for tube contours are ±.060 in.

FIGURE 3.2.7.8 (Continued)

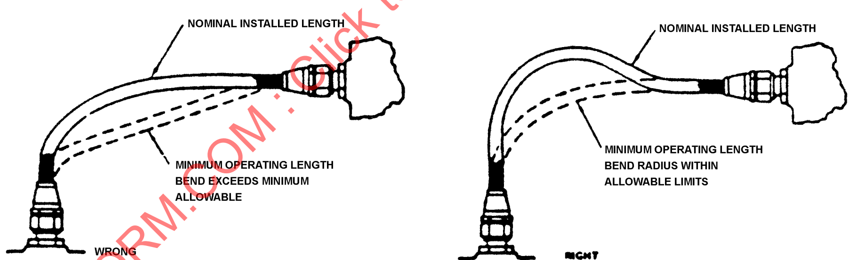
PROBLEM: HOSE ASSEMBLY INSTALLATION PRACTICES

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

- Proper length including at least one hose bend should be used to accommodate worst case of tolerance stackup.
- Proper routing to withstand motions/deflections.
- Proper installation to eliminate twisting.
- Proper use of elbows to provide most direct routing and minimize opportunity for abrasion.

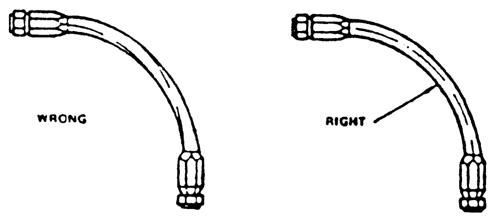


Proper length including at least one hose bend

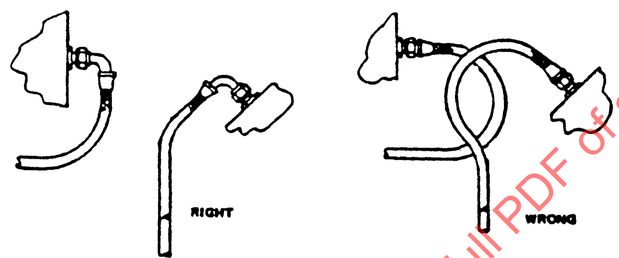


Proper routing to withstand motions/deflections

FIGURE 3.2.7.9 - Hose Installation Practices



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 AIR1659, "Handling and Installation Practice for Aerospace Hose Assemblies" for additional information.

FIGURE 3.2.7.9 (Continued)