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AEROSPACE INFORMATION REPORT

Submitted for recognition as an American National Standard

SAE AIR4543

Issued 1992-01-13

AEROSPACE HYDRAULICS AND ACTUATION LESSONS LEARNED

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1. SCOPE:

This Aerospace Information Report (AIR) contains technical information on aerospace hydraulic and actuation 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 three Lessons Learned Symposia in 1989 and 1990 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 3.1.21 present these lessons learned.

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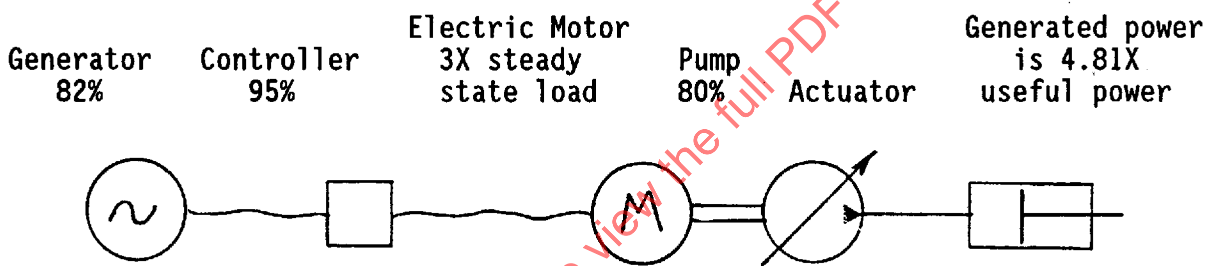
PROBLEM: POWER CONVERSION EFFICIENCY

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

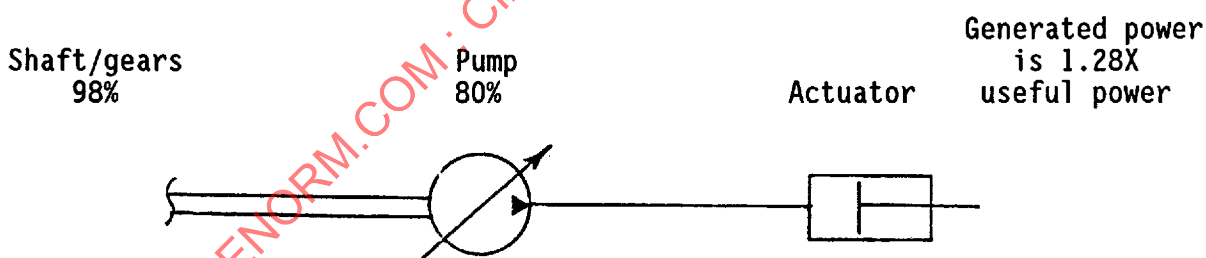
a. mechanical
b. electrical
c. hydraulic
d. pneumatic

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

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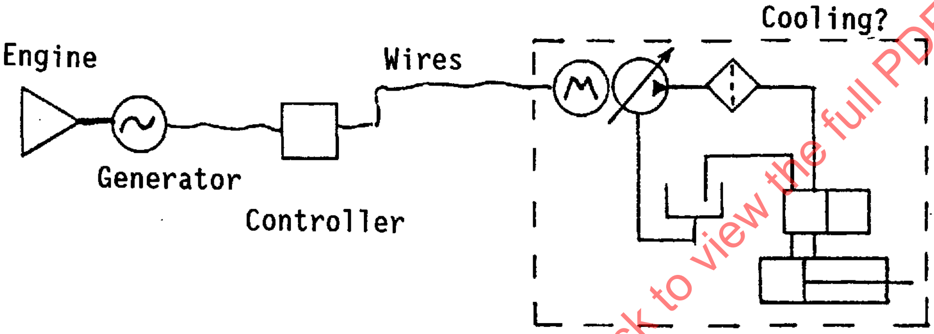
PROBLEM: BIASED TRADE STUDIES

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

ILLUSTRATION:

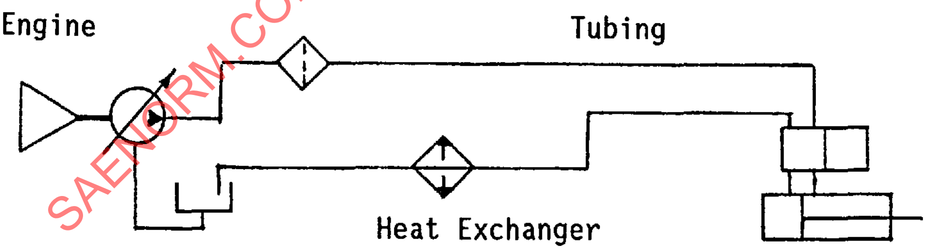
Distributed hydraulic packages



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

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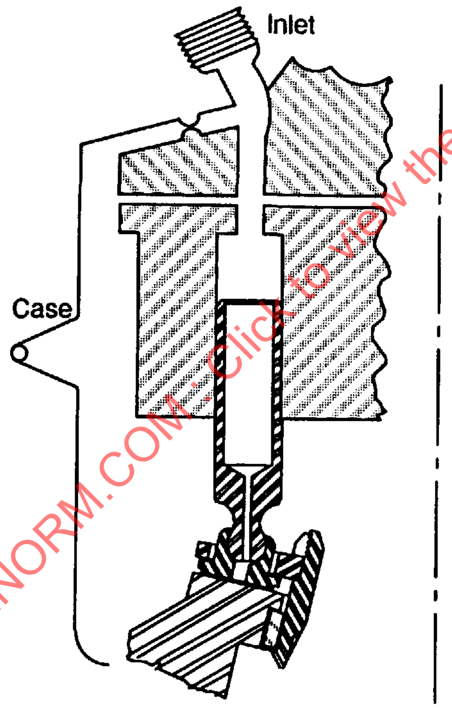
PROBLEM: PUMP OVERHEAT AND RAPID WEAR

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:



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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

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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%

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

FIGURE 3.1.4 - Excessive Pump Pressure Ripple in System

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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

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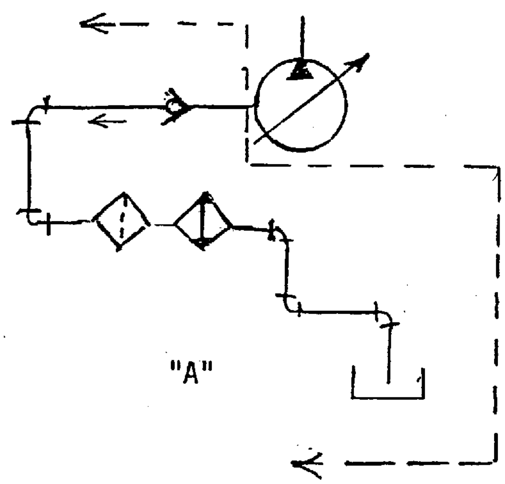
Design pressure (pump/motor case) must be higher than any combination of A:



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

A = Pressure Drop Consideration

Typical "Case Circuit"

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)

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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

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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

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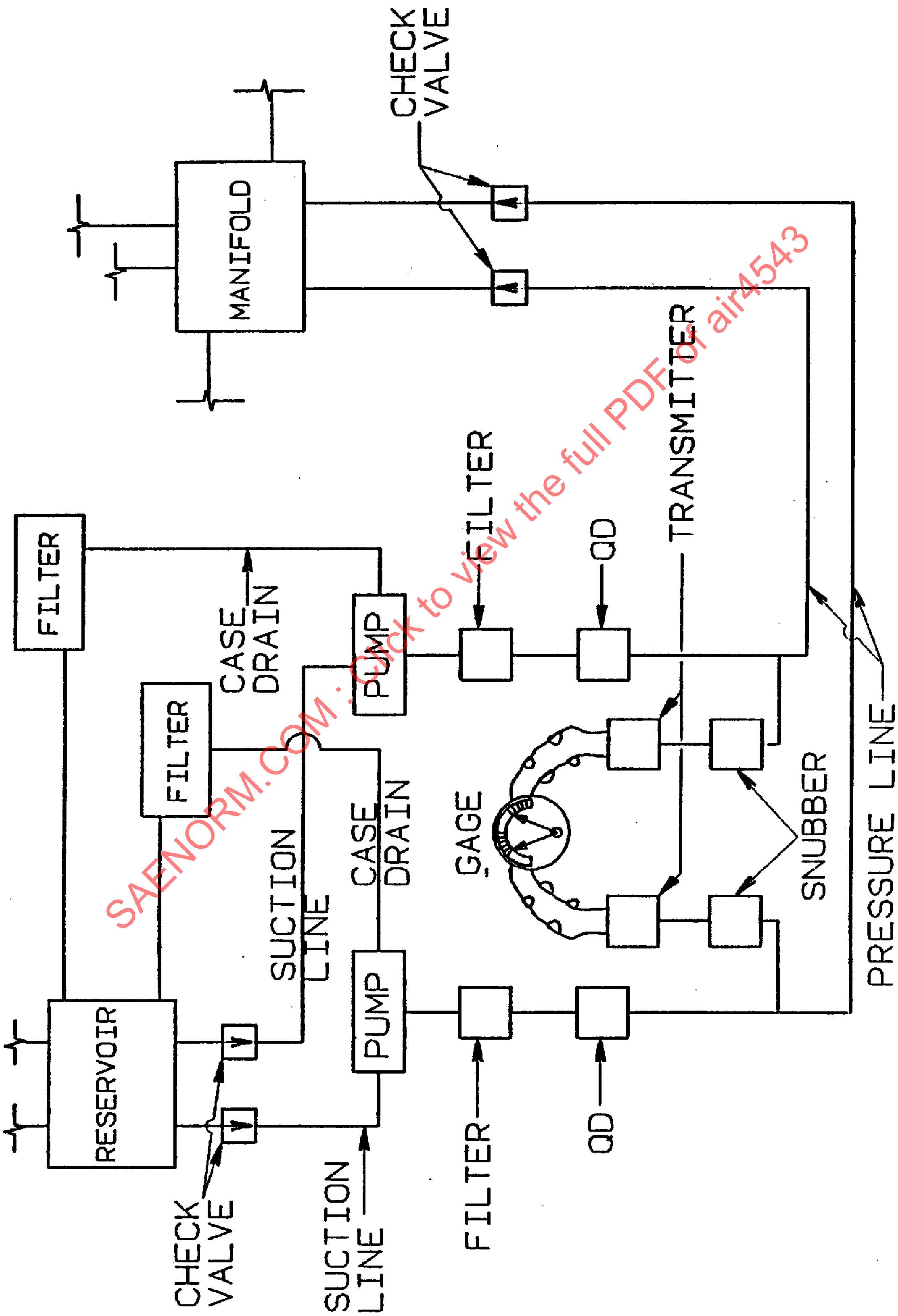
FIGURE 3.1.7 (Continued)



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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:

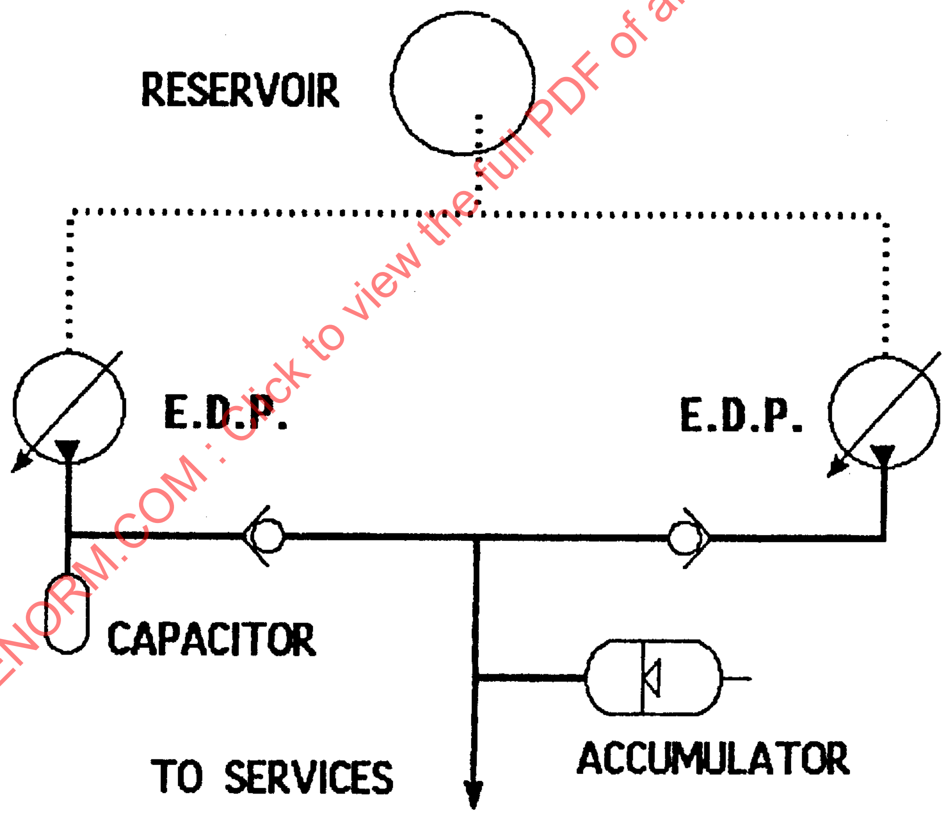
   a. Earlier production aircraft;

   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



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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

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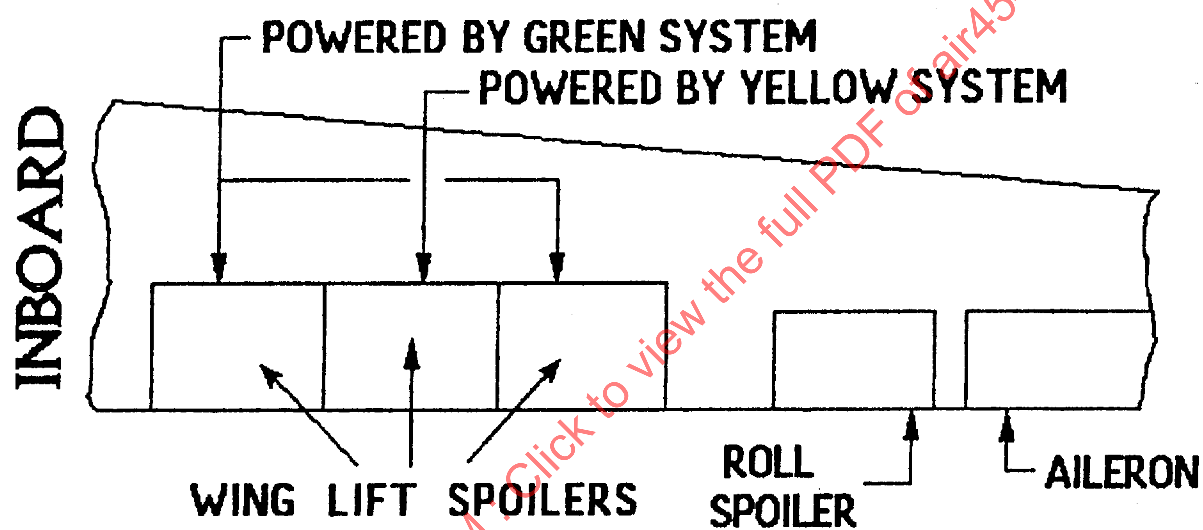
PROBLEM: EFFECT OF JAR/FAR REQUIREMENT ON FLIGHT CONTROL SURFACE OPERATION

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:

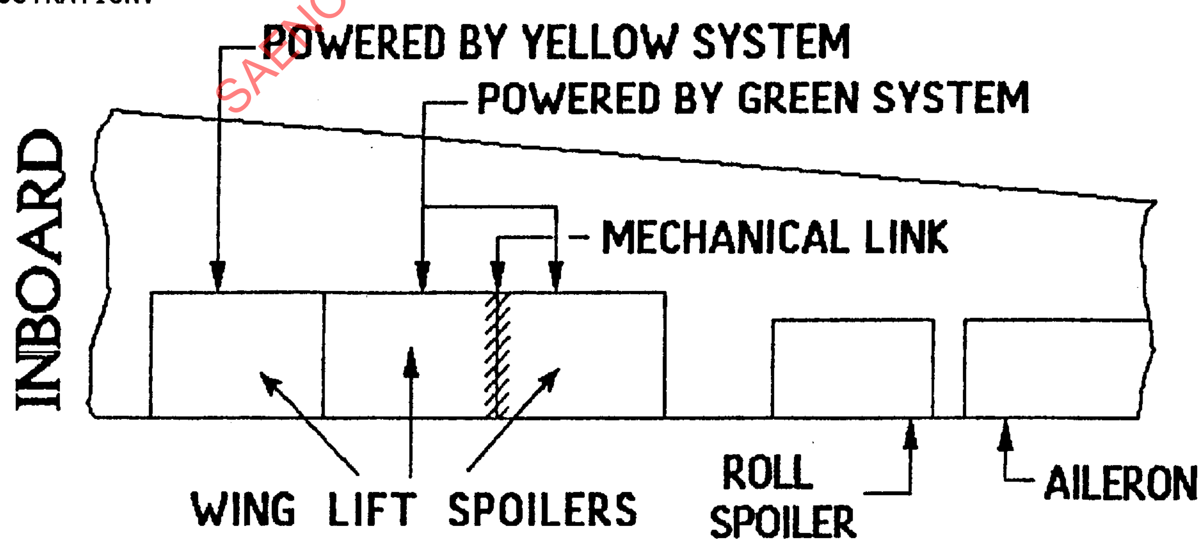


FIGURE 3.1.9 - Effect of JAR/FAR Requirement on Flight Control Surface Operation

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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

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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.

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FIGURE 3.1.11 - Low Pressure Cavitation During Valve Endurance Test

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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.)

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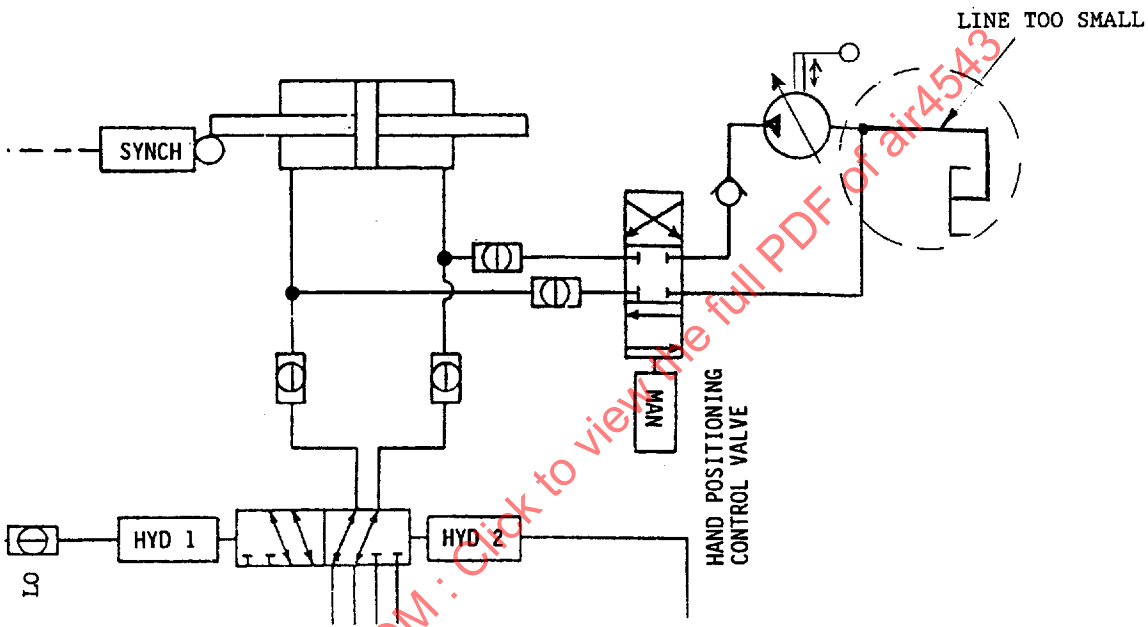
PROBLEM: INADEQUATELY SIZED HAND PUMP SUCTION LINE

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:



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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

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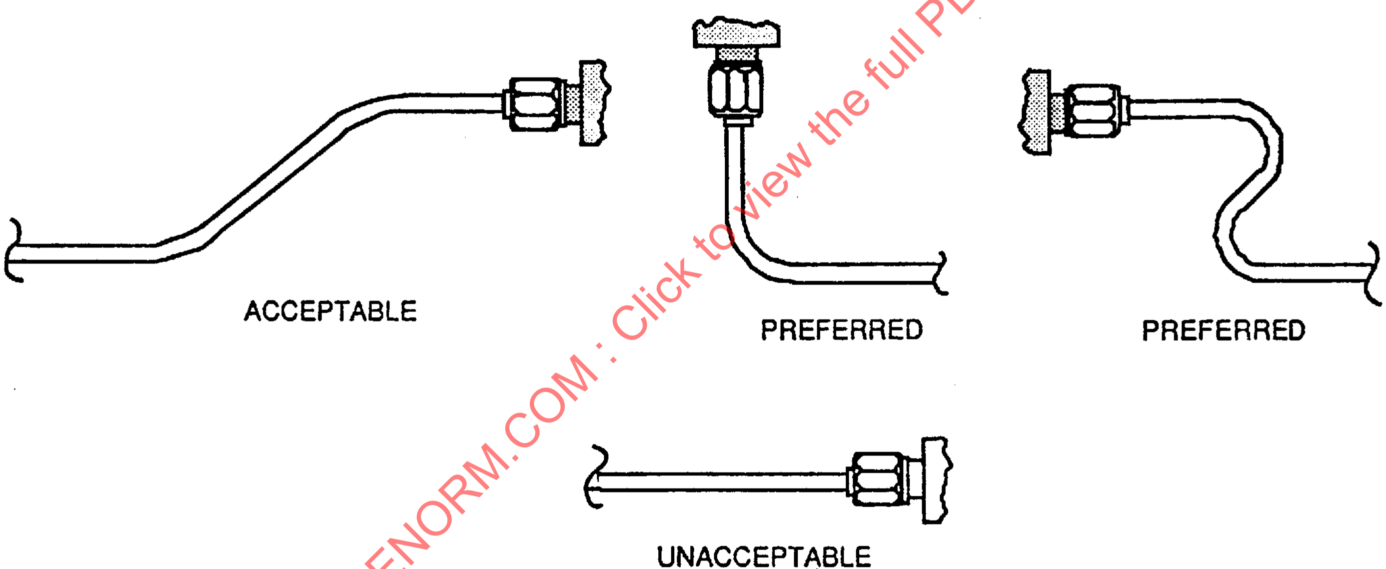
PROBLEM: HYDRAULIC COMPONENT INSTALLATION/TUBING TOLERANCES

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:



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

FIGURE 3.1.14 - Hydraulic Component Installation/Tubing Tolerances

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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 Pressure Due to Air in the System

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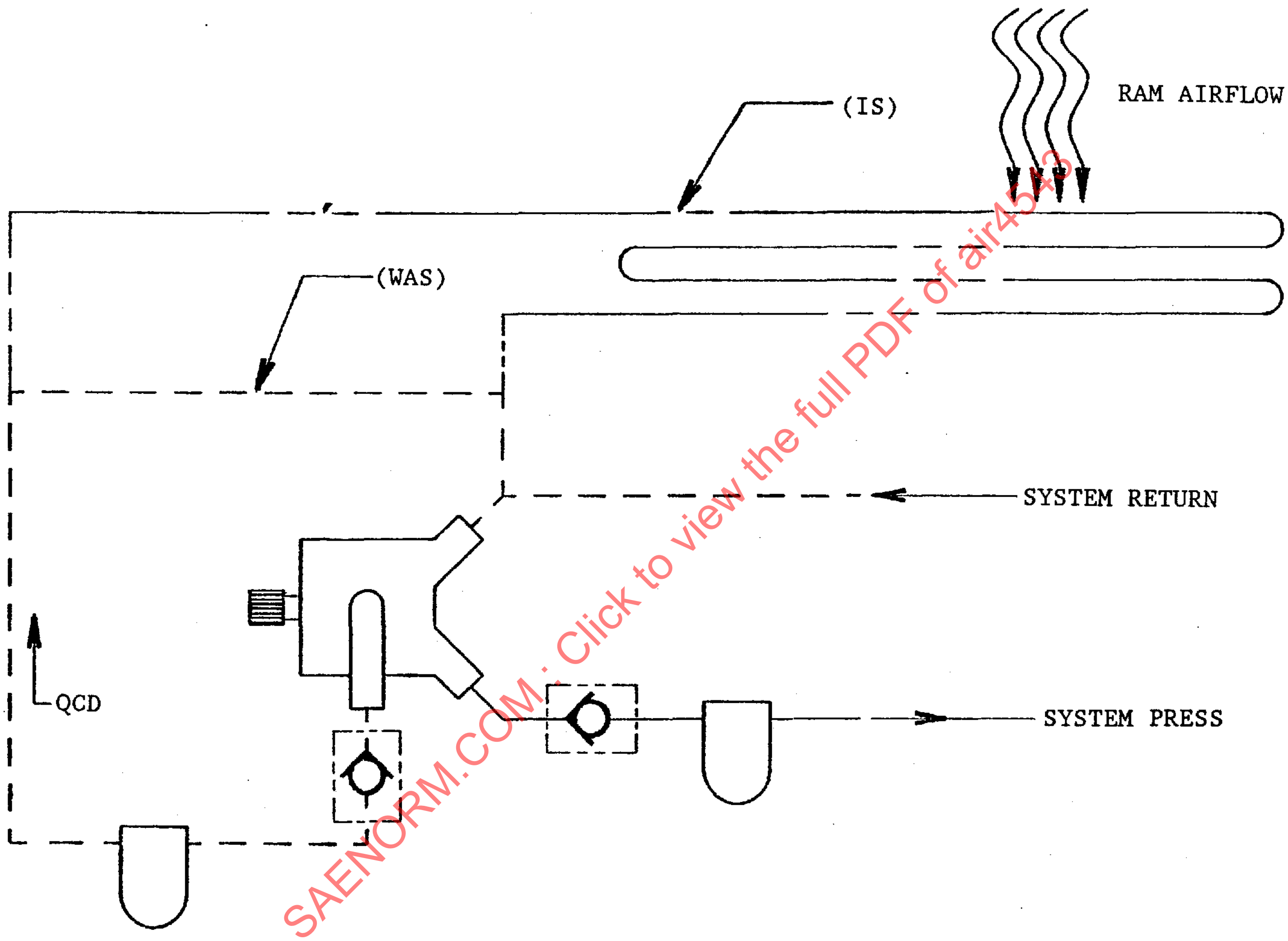
PROBLEM: SYSTEM OVERHEAT CAUSED BY INEFFICIENT PUMP AND LACK OF HEAT EXCHANGER

ISSUE: System can exceed 275°F

ILLUSTRATION:



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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

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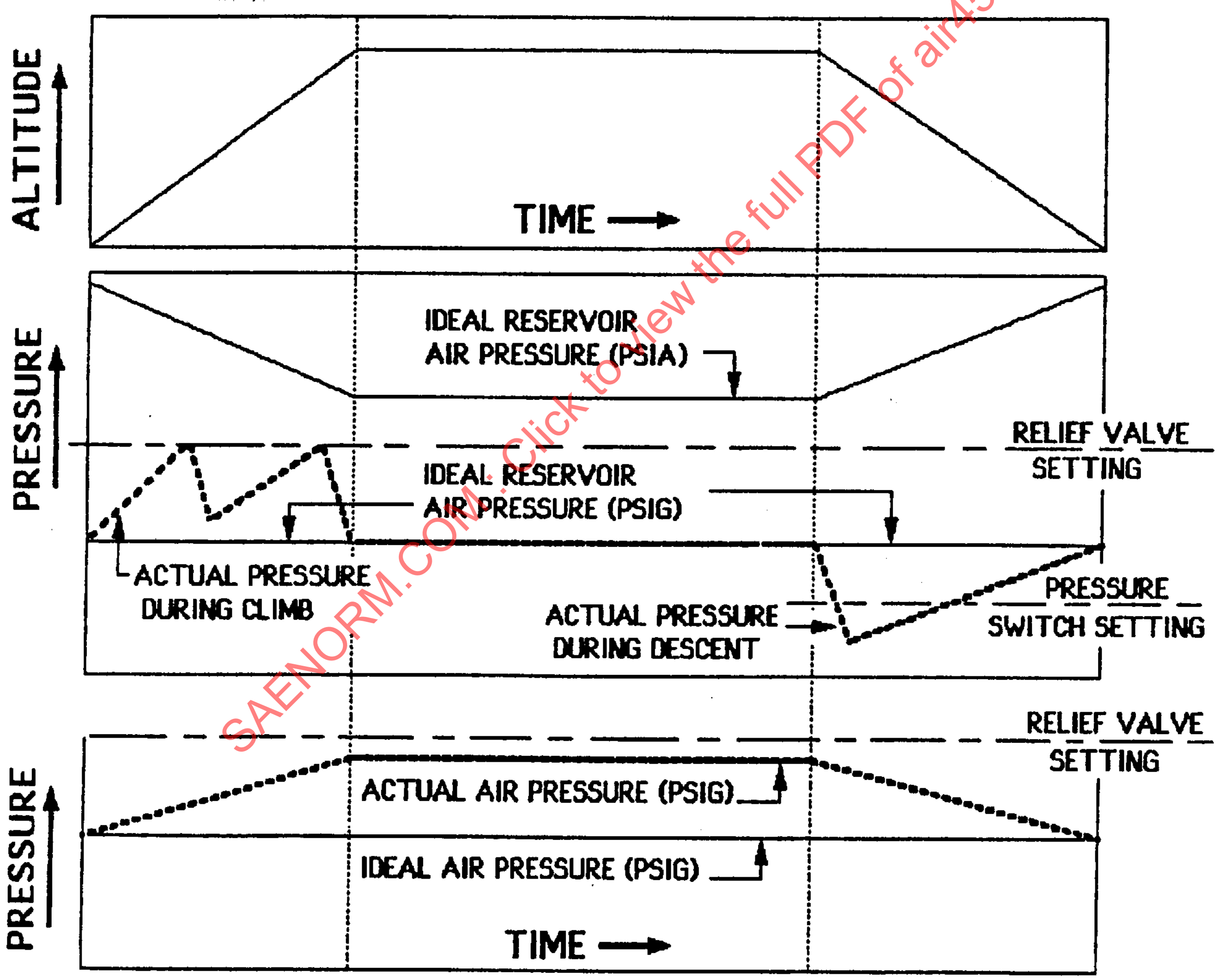
PROBLEM: SPURIOUS HYDRAULIC RESERVOIR LOW AIR PRESSURE WARNING

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:



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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

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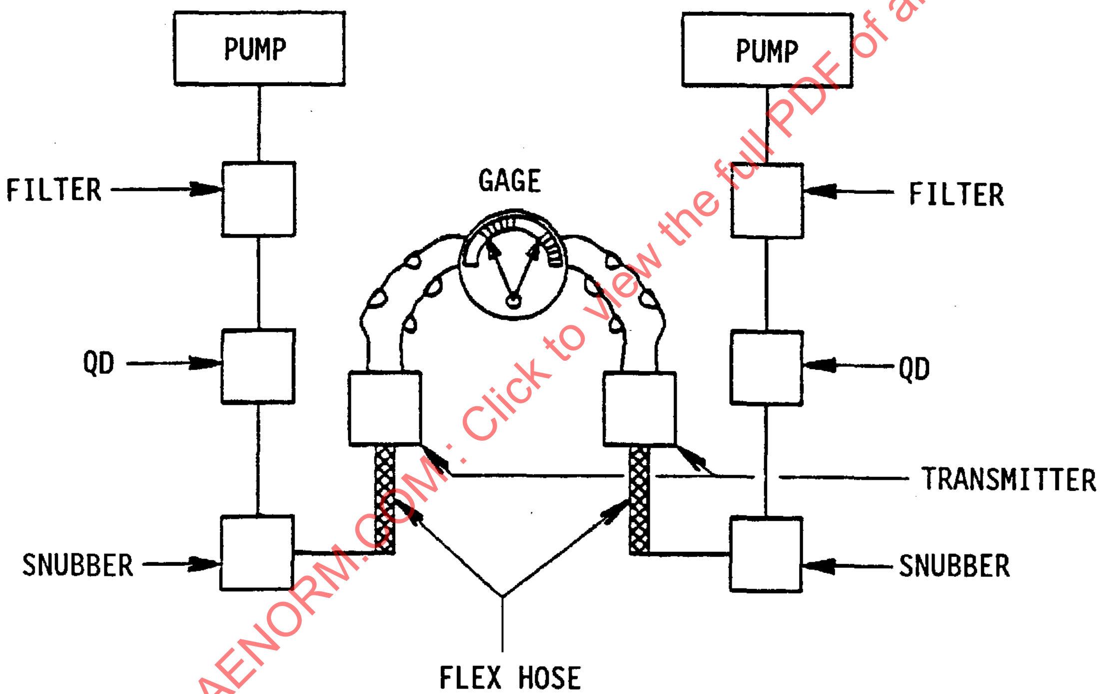
PROBLEM: FALSE PRESSURE TRANSMITTER INDICATION

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:



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SOLUTION: Replace the flex hose.

FIGURE 3.1.18 - False Pressure Transmitter Indication

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PROBLEM: HYDRAULIC SYSTEM FAILURES DUE TO TIRE BURSTS IN THE WHEEL WELL

ISSUE: Tire overheat due to hard braking or a 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 even blow off the wheel well doors.

SOLUTION: Since this event cannot be prevented, the hydraulic components and tubing must be located and routed to provide the best possible protection.

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

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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

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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

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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 3.2.1.3.

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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

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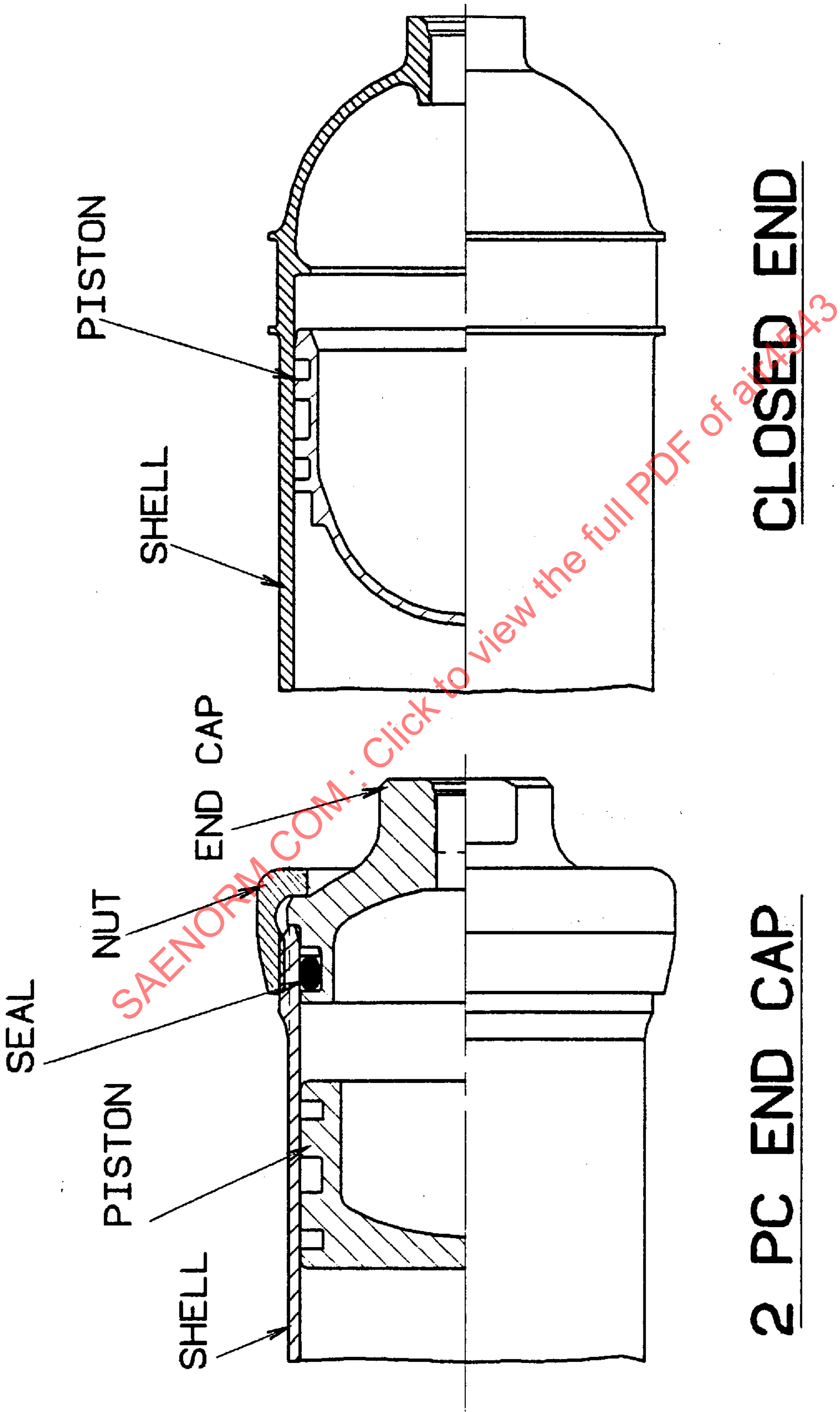
FIGURE 3.2.1.1 (Continued)



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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

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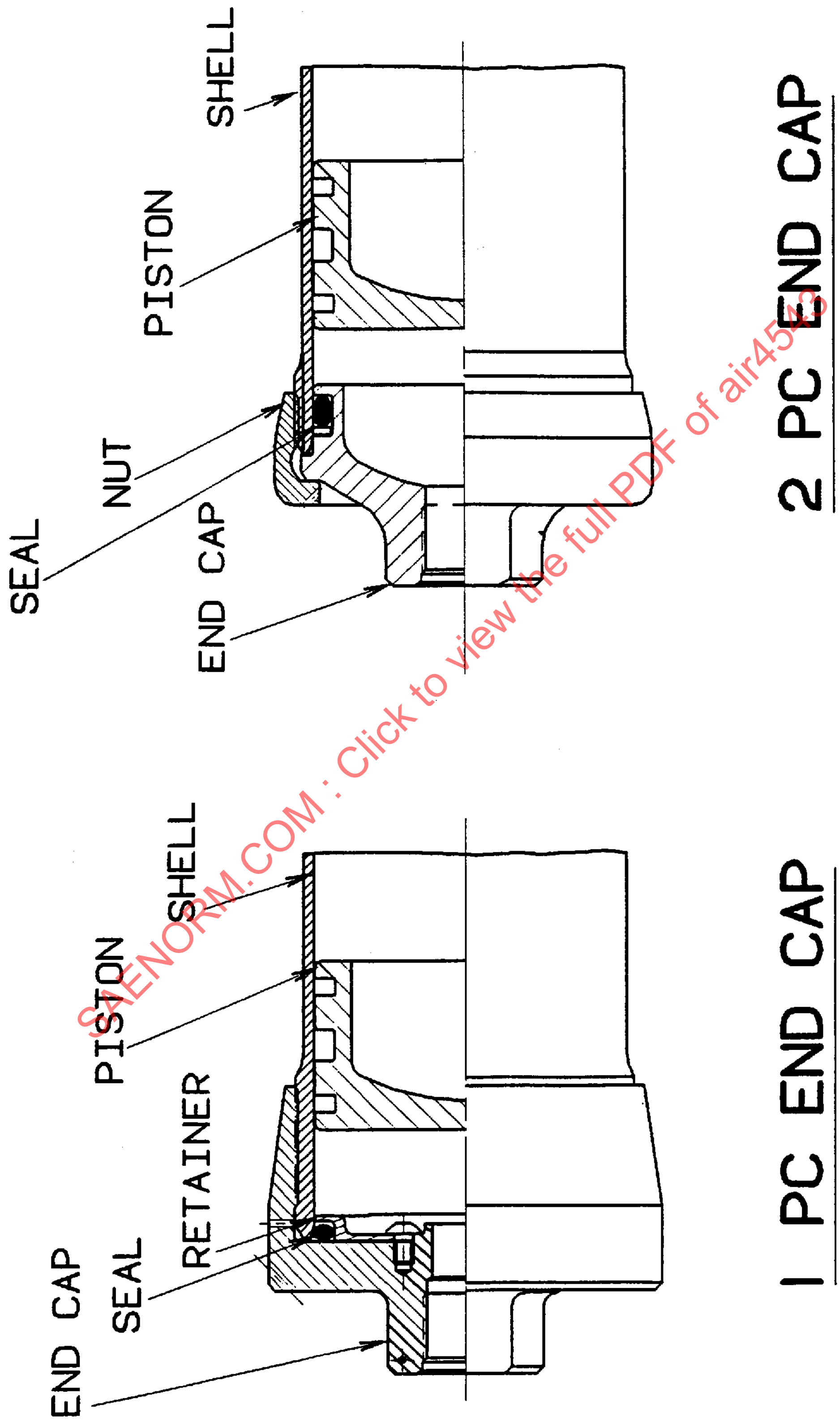
FIGURE 3.2.1.2 (Continued)



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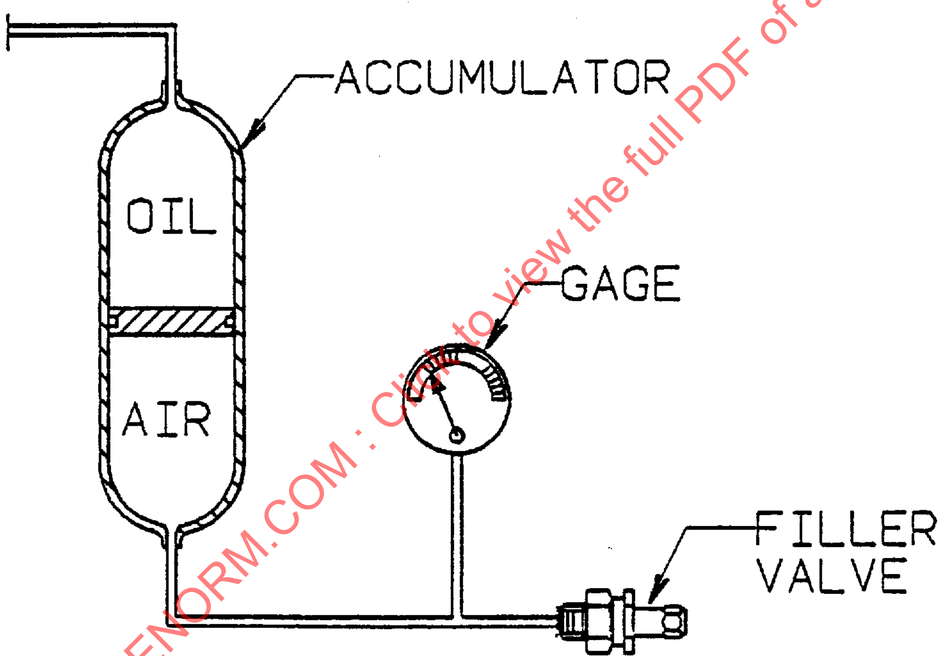
PROBLEM: FALSE ACCUMULATOR GAGE READING

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:



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

FIGURE 3.2.1.3 - False Accumulator Gage Reading

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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 3.2.2.14.

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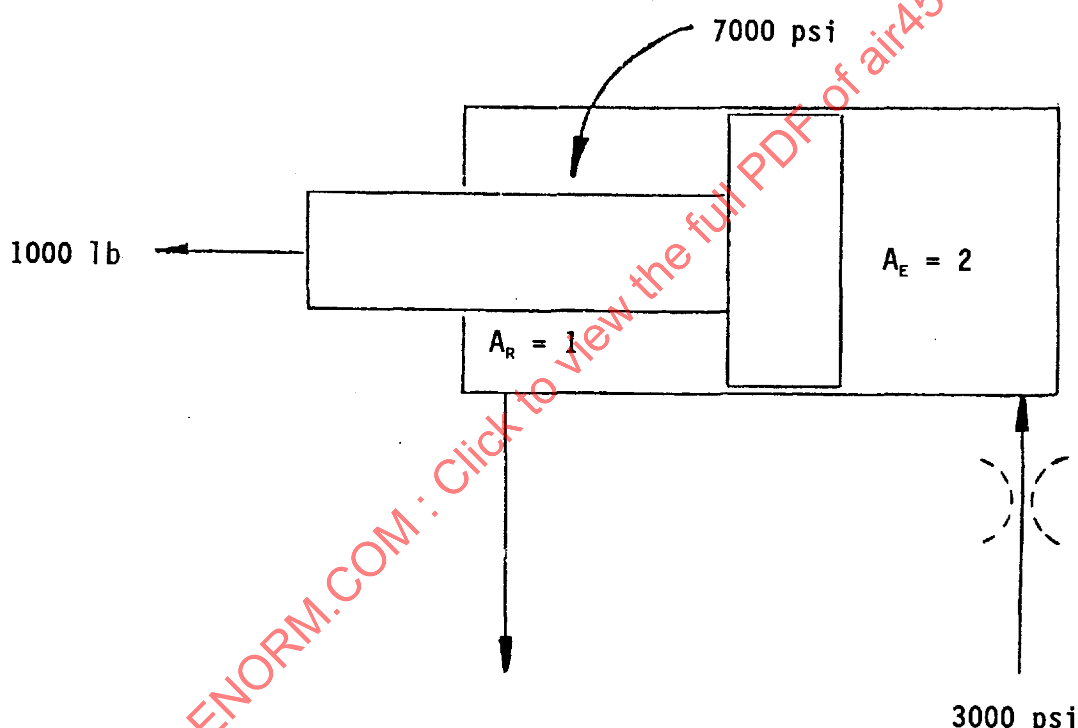
PROBLEM: PRESSURE INTENSIFICATION IN EXTENDING CYLINDER

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:



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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

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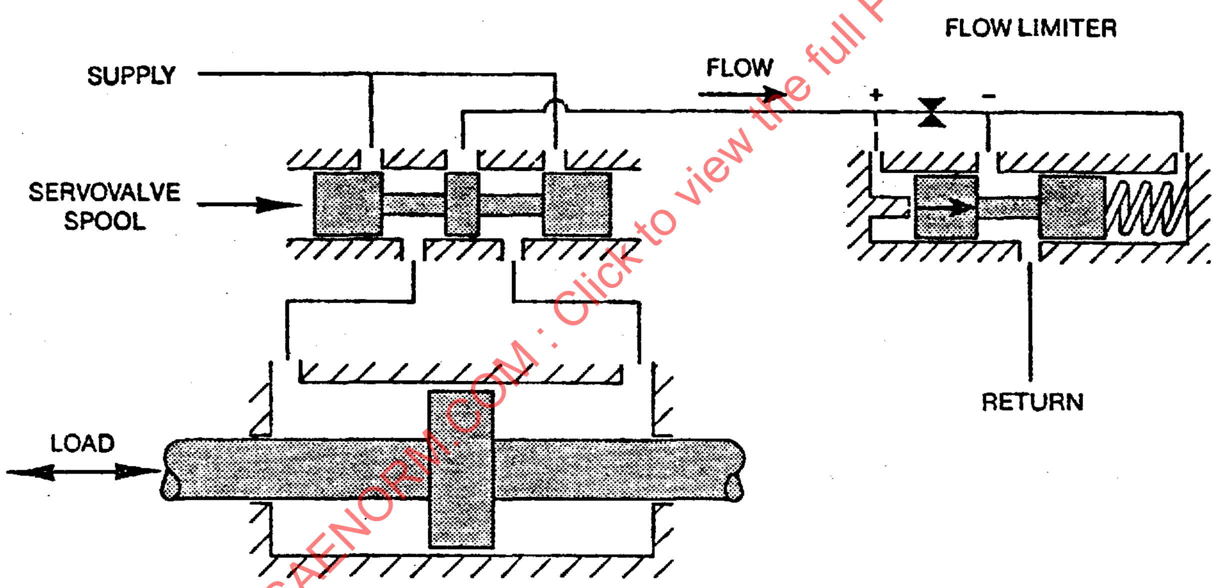
PROBLEM: SERVOVALVE SEPARATION FROM THE MANIFOLD DUE TO SERVOACTUATOR VELOCITY LIMITING

ISSUE:

1. During the countdown for the initial launch of the Gemini two-man space vehicle, the primary servovalve of the dual redundant TVC actuator blew off the manifold.
2. The actuator flow limiter used to restrict excessive TVC 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:



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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

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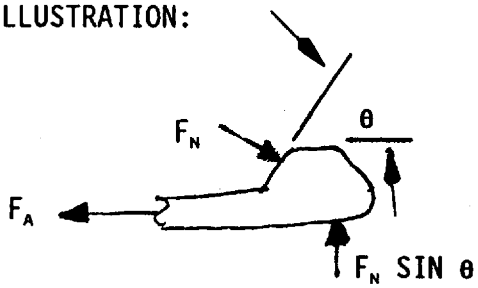
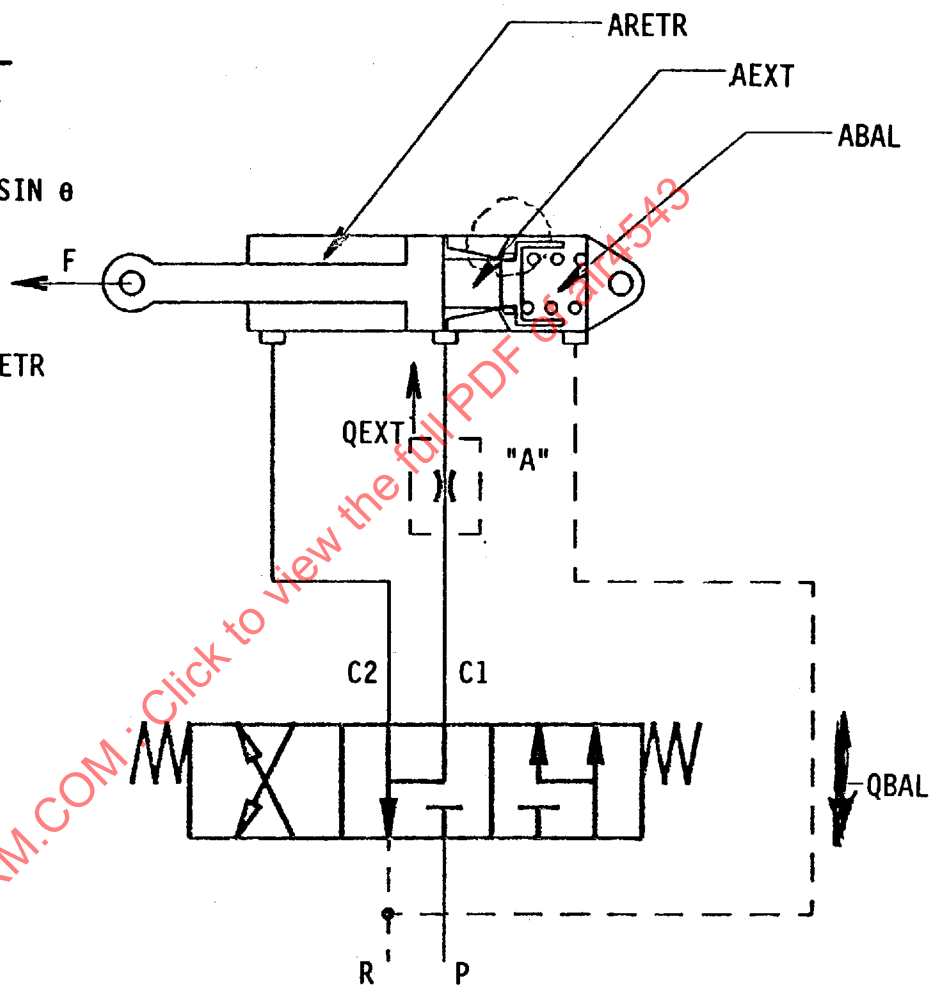
PROBLEM: GALLING OF THE CYLINDER LOCK MECHANISM WHEN UNLOCKED UNDER HIGH LOADS

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

ILLUSTRATION:



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



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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

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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).

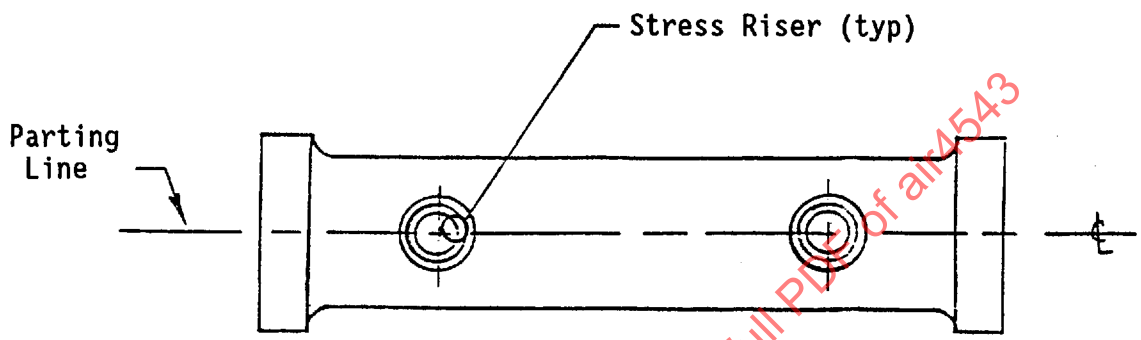

ILLUSTRATION 1

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

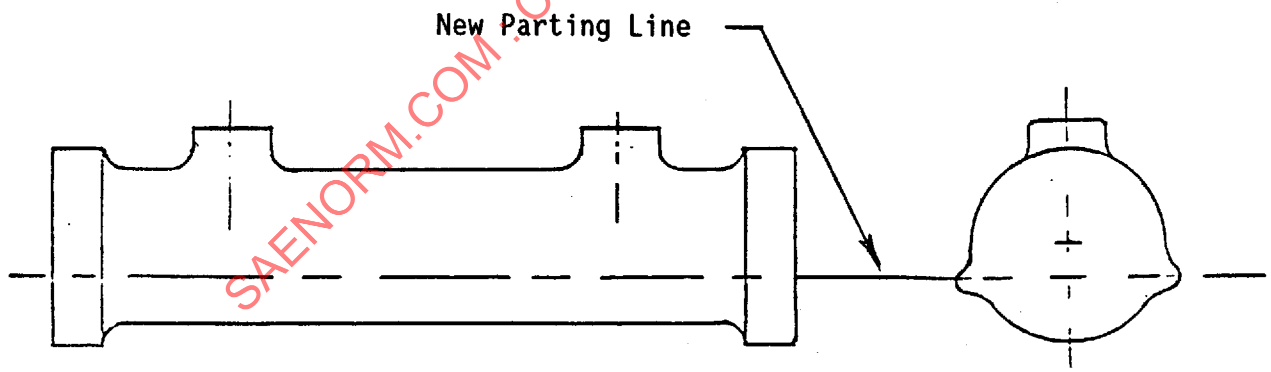

ILLUSTRATION 2

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

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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.

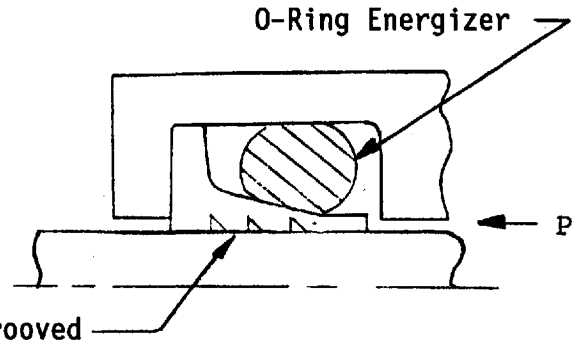

ILLUSTRATION 1

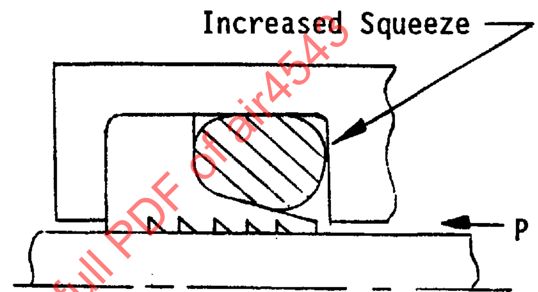

ILLUSTRATION 2

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

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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.

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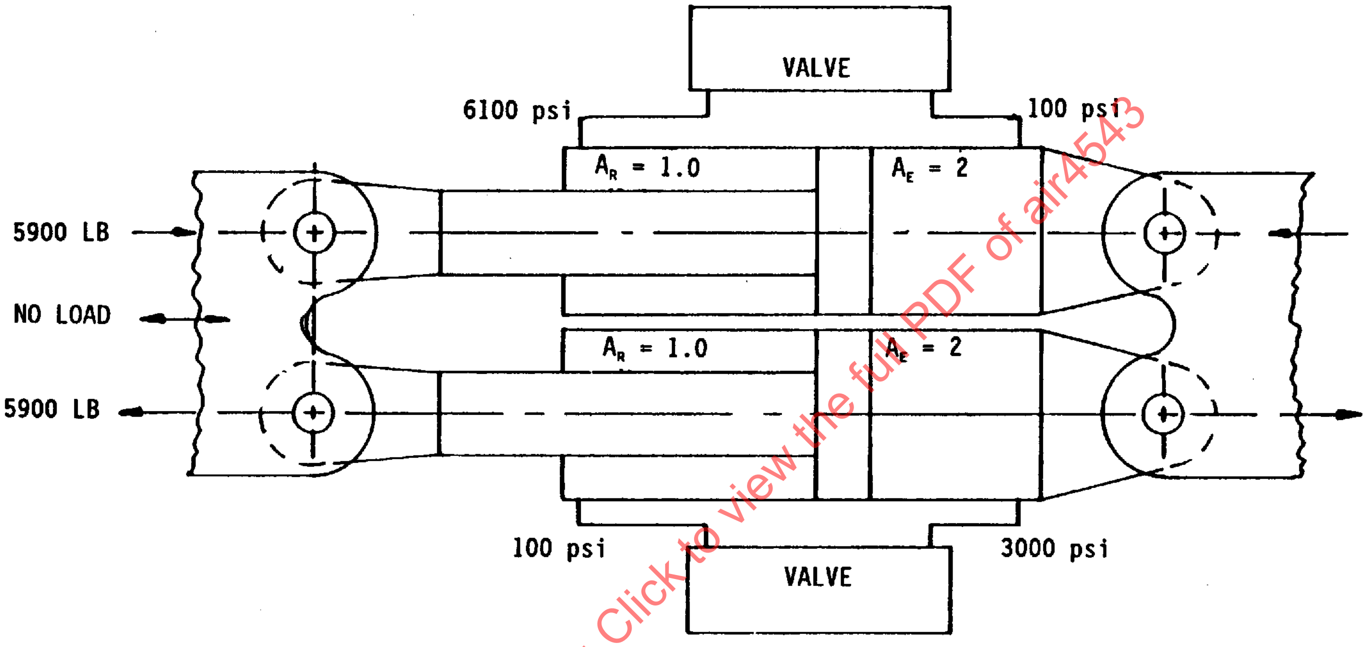
PROBLEM: FATIGUE DAMAGE TO LOADED STRUCTURE AND CYLINDERS

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

ILLUSTRATION:



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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

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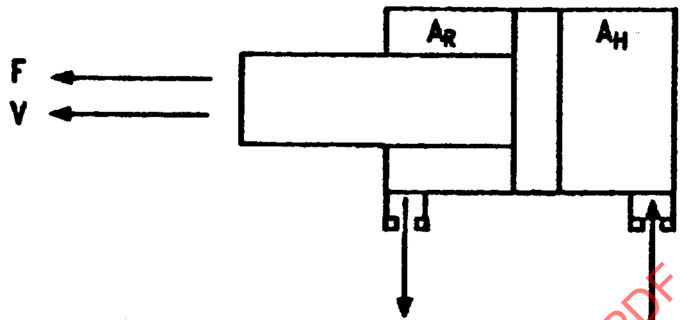
PROBLEM: VELOCITY CONTROL OF EXTENDING ACTUATOR

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

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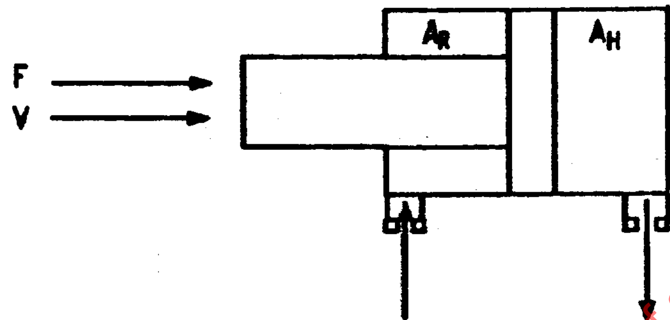
PROBLEM: VELOCITY CONTROL OF RETRACTING ACTUATOR

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

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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

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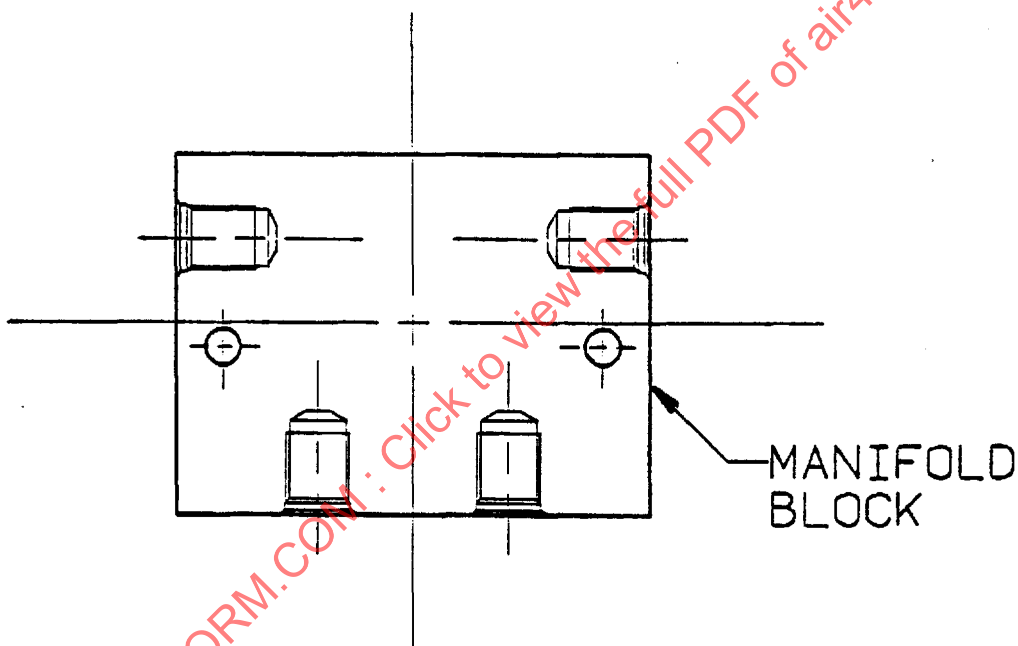
PROBLEM: AILERON FAILED TO OPERATE

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

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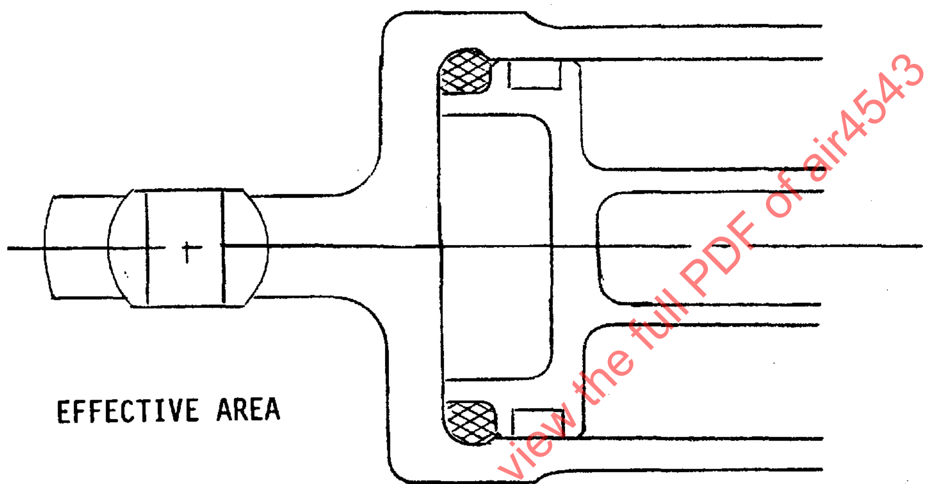
PROBLEM: FULLY RETRACTED ACTUATOR FAILS TO EXTEND AGAINST LOAD

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:

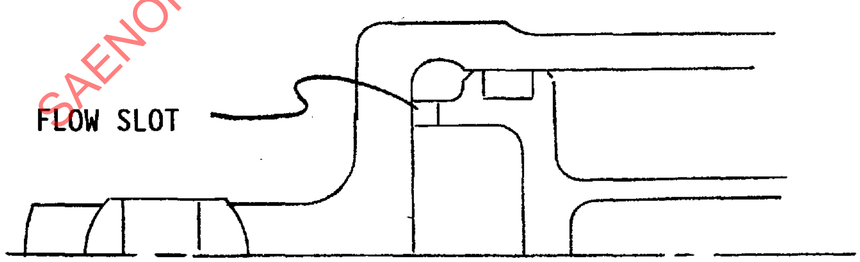


FIGURE 3.2.2.10 - Fully Retracted Actuator Fails to Extend Against Load

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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

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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

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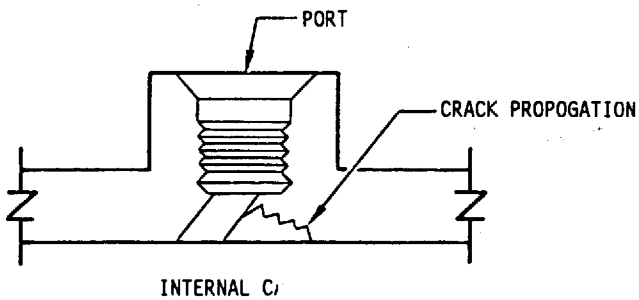
PROBLEM: ALUMINUM MANIFOLD FATIGUE

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)

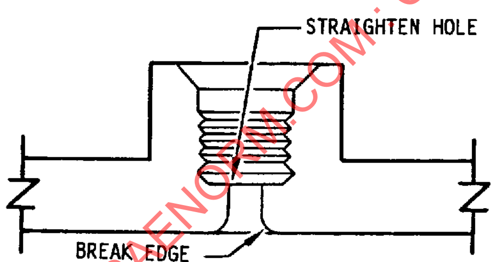


ILLUSTRATION 1

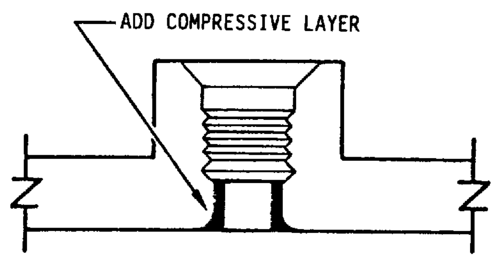


ILLUSTRATION 2

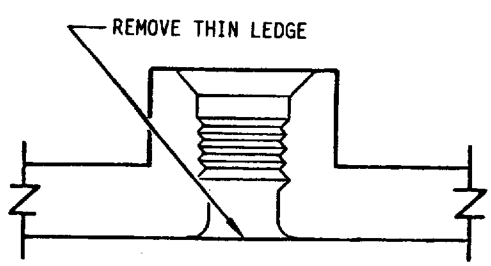


ILLUSTRATION 3

FIGURE 3.2.2.13 - Aluminum Manifold Fatigue

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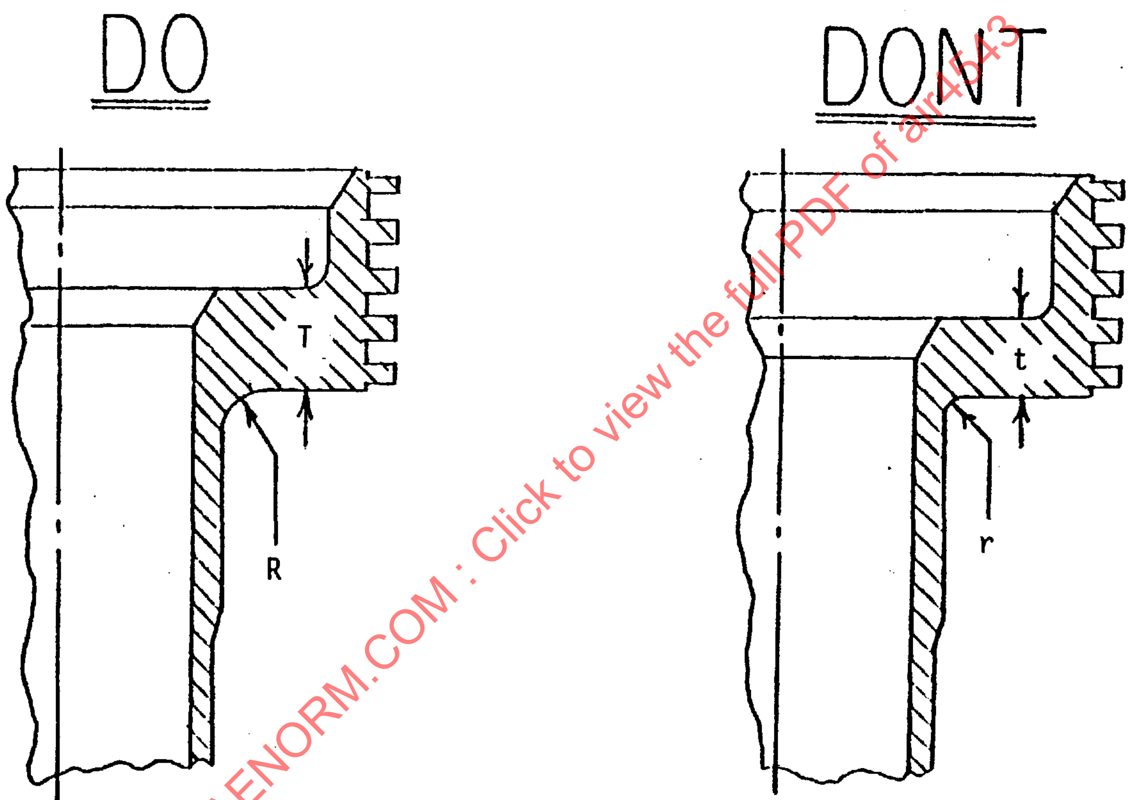
PROBLEM: PISTON HEAD SEPARATION

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

ILLUSTRATION:



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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

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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 3.2.3.8.

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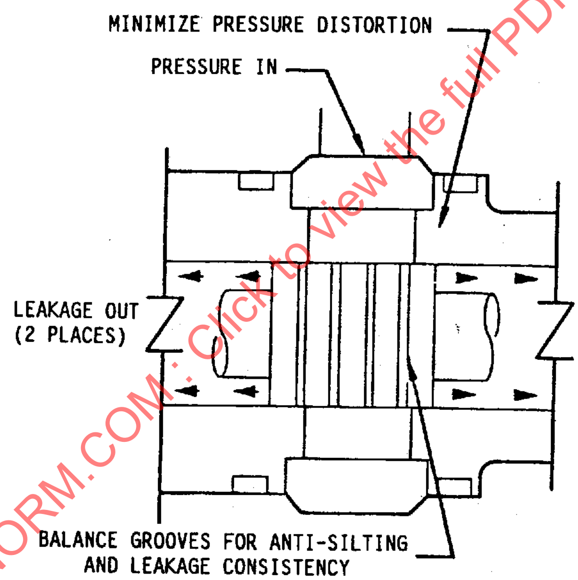
PROBLEM: VALVE REJECTIONS DUE TO SUPPLIER/CONTRACTOR FLUID FILTRATION VARIANCES

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:



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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

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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.

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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

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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

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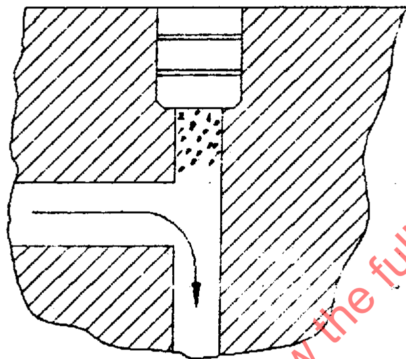
PROBLEM: ENTRAPPED CONTAMINATION

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:

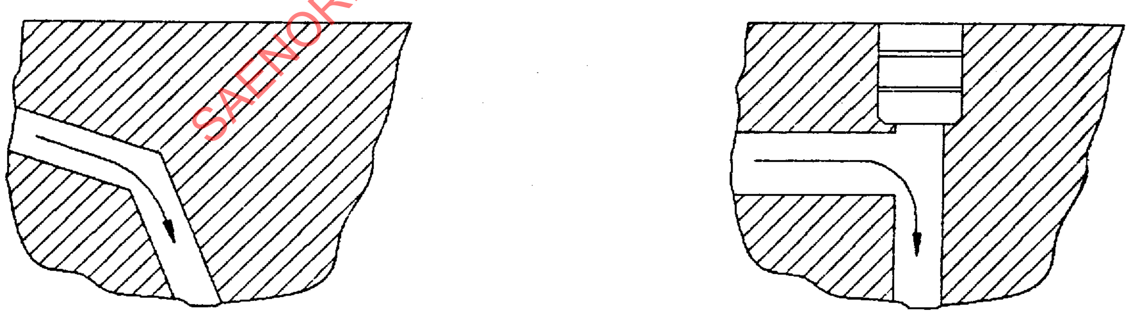


FIGURE 3.2.3.4 - Entrapped Contamination

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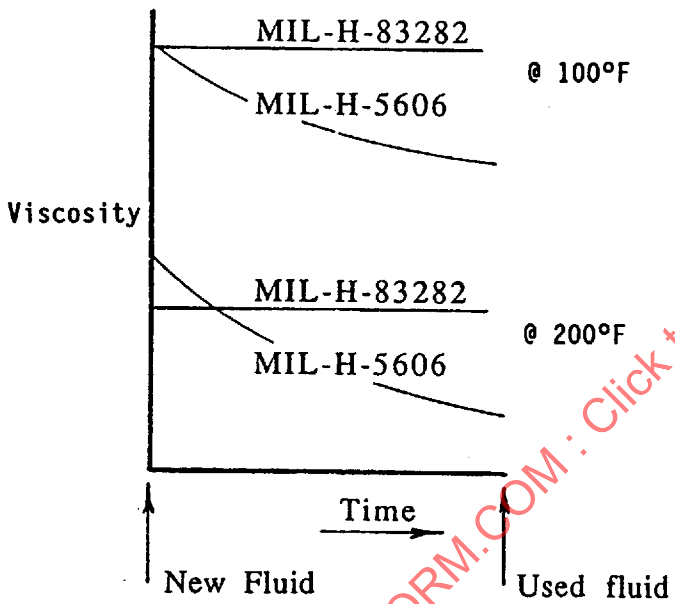
PROBLEM: VISCOSITY EFFECTS ON PERFORMANCE

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.

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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

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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

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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

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PROBLEM: INADEQUATE COMPONENT LIFE AT ELEVATED TEMPERATURES

ISSUE: Incompatibility of fluid with higher temperatures - MIL-H-46170.

SOLUTION: Do not use preservative fluid as an operational fluid.

FIGURE 3.2.3.8 - Inadequate Component Life at Elevated Temperatures

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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 3.2.4.10.

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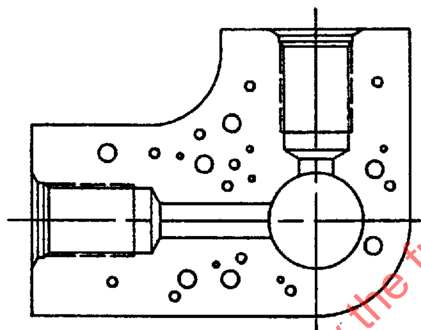
PROBLEM: CASTING FATIGUE

ISSUE: Casting "porosity" caused short fatigue life.

ILLUSTRATION:



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

ILLUSTRATION:

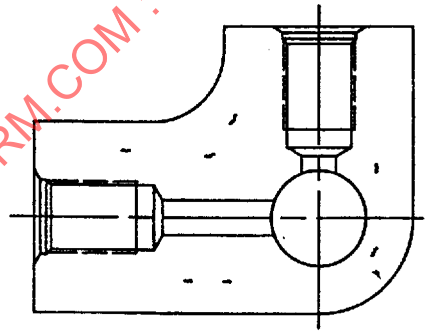


FIGURE 3.2.4.1 - Casting Fatigue

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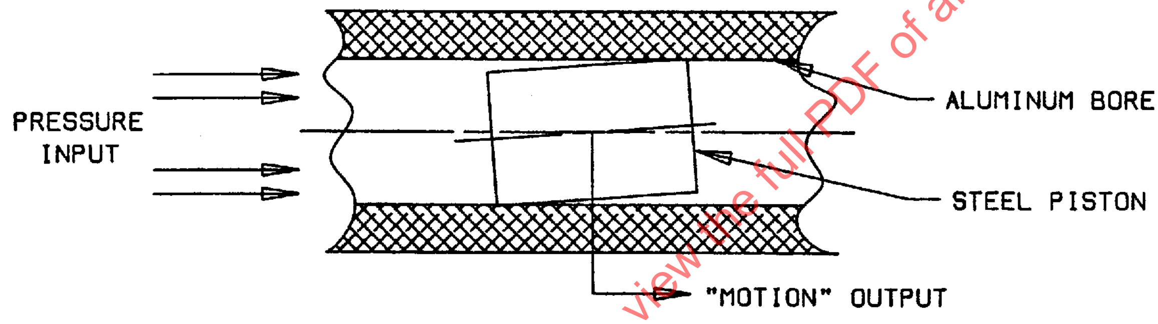
PROBLEM: SLIDING PISTON "STICKING" IN BORE

ISSUE: Steel piston "sticking" in aluminum bore.

ILLUSTRATION:



SOLUTION: Use PTFE impregnated "hard anodize" bore.

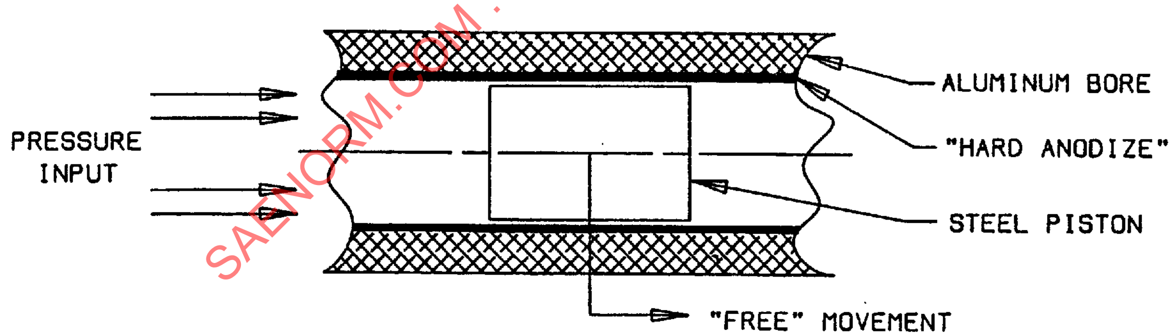


FIGURE 3.2.4.2 - Sliding Piston "Sticking" in Bore

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PROBLEM: VALVE JAM CAUSED BY RETAINED AUSTENITE

ISSUE:

1. High heat treat 440°C cres 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

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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.

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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

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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

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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.

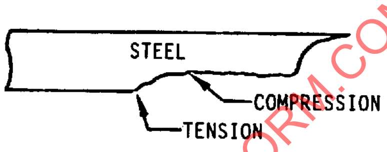

ILLUSTRATION 1

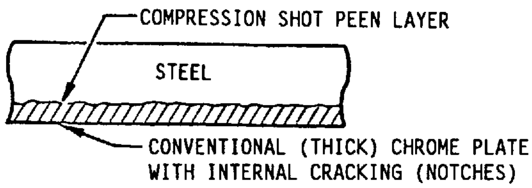

ILLUSTRATION 2

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

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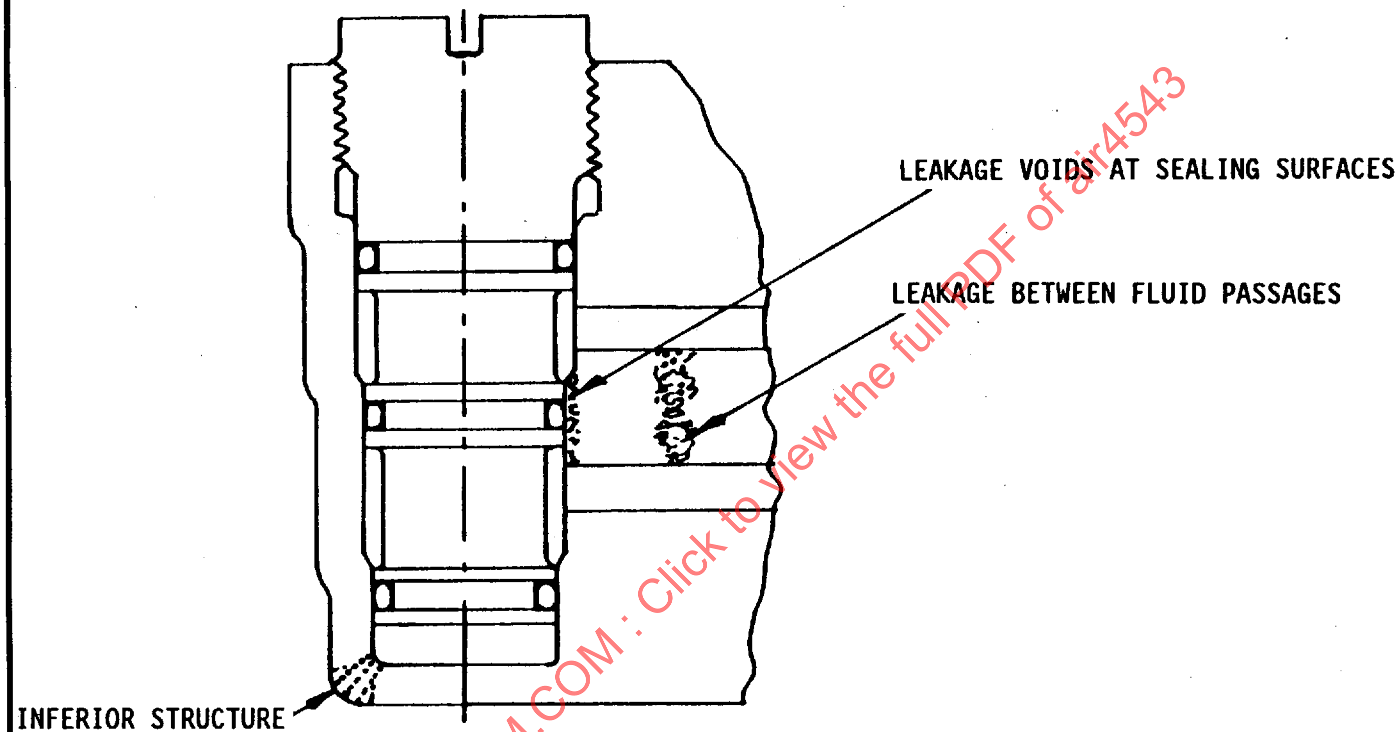
PROBLEM: LEAKAGE AND INFERIOR STRUCTURAL INTEGRITY ASSOCIATED WITH POROSITY IN CAST MANIFOLDS

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

ILLUSTRATION:



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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

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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

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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

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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-h.
   b. Drain, refill, flush, or decontaminate until acceptable levels are reached.
6. Material design
   a. Use cres steels - 440C, BG-42, etc. when possible.
7. Close or pressurize support equipment reservoirs.

FIGURE 3.2.4.10 - Internal Corrosion of Hydraulic Components

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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 3.2.5.5.

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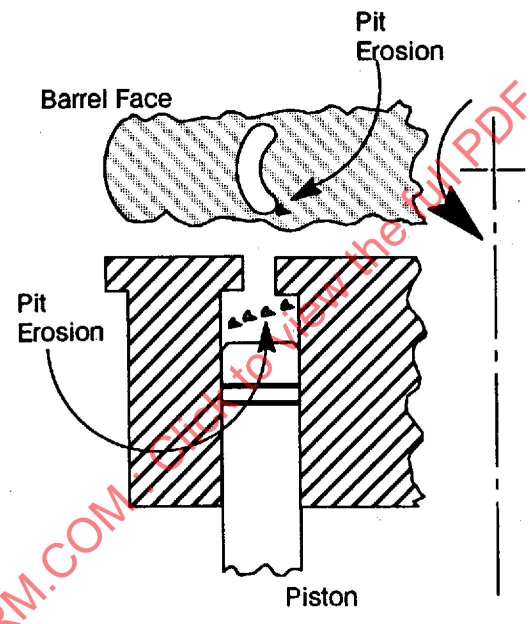
PROBLEM: PUMP CAVITATION/EROSION

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

ILLUSTRATION:



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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

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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

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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

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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

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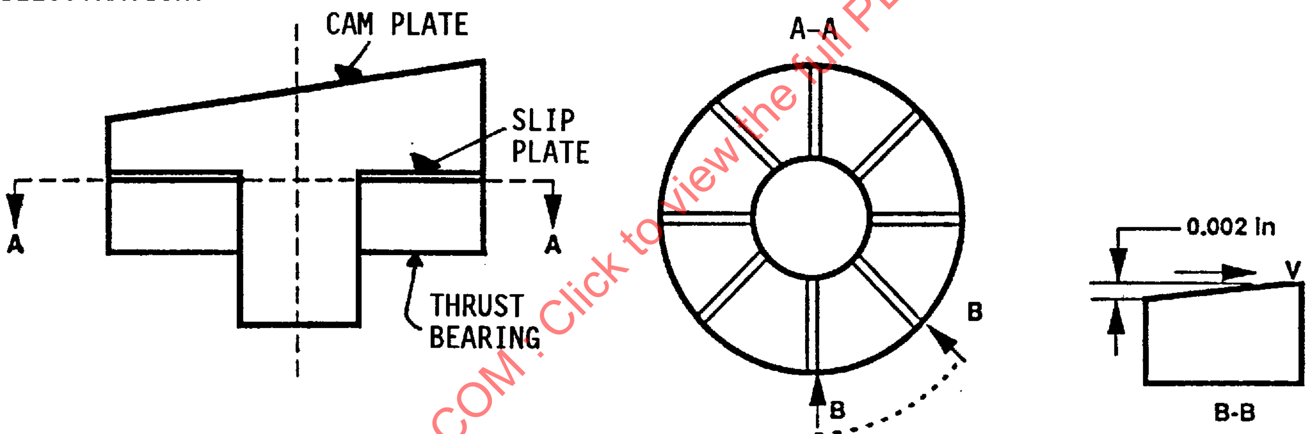
PROBLEM: EARLY PUMP FAILURES DUE TO RAPID AIR-TURBINE STARTS OF THE B-52 HYDRAULIC PACKS (TEN PER SHIPSET)

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)

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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 3.2.6.9.

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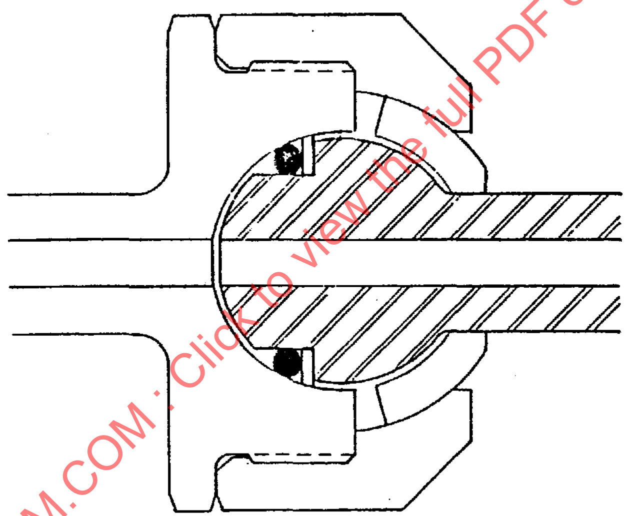
PROBLEM: SPHERICAL BALL SWIVEL LEAKAGE

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

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PROBLEM: SERVOACTUATOR PISTON SEAL BLOWBY

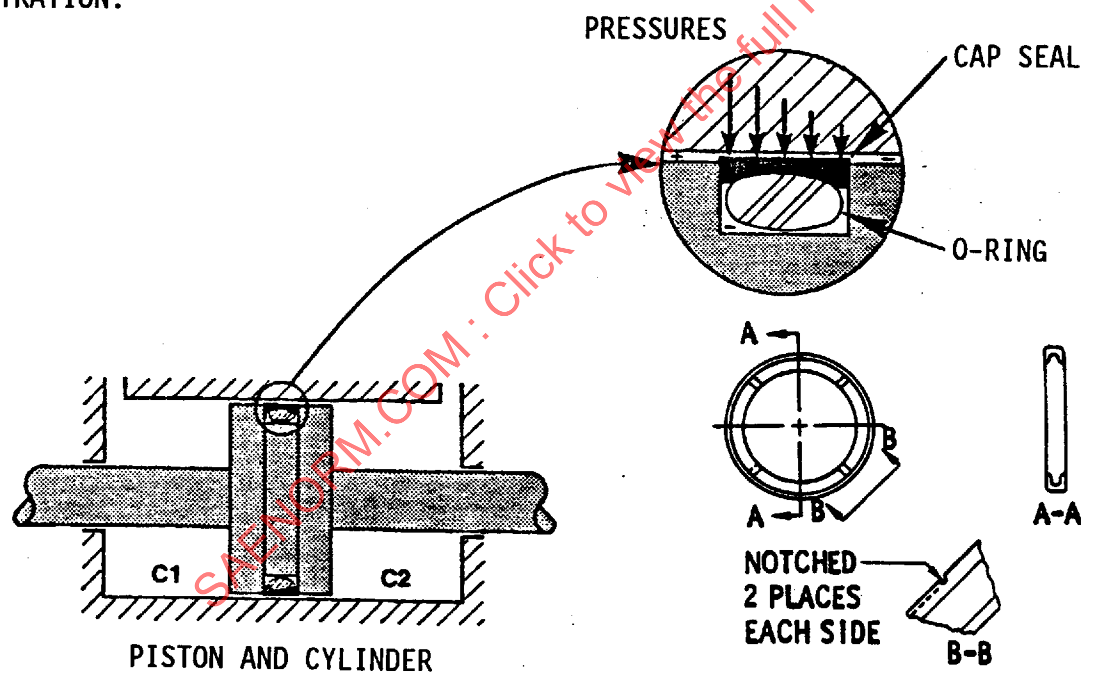
ISSUE:

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

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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

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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

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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 - TBD by the A-6 Committee.

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)

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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).

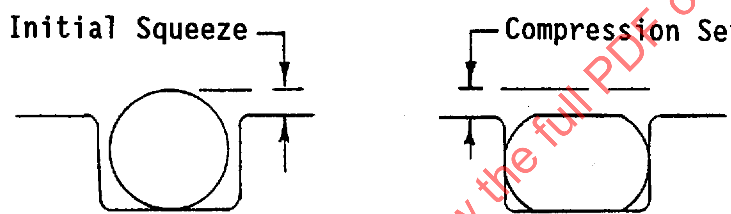

ILLUSTRATION 1

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

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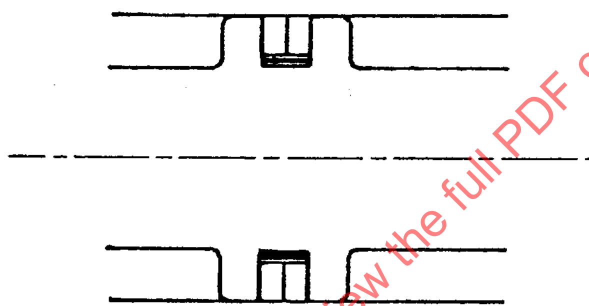
PROBLEM: SMALL ACTUATOR PISTON RING LEAKAGE BLOWBY

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:

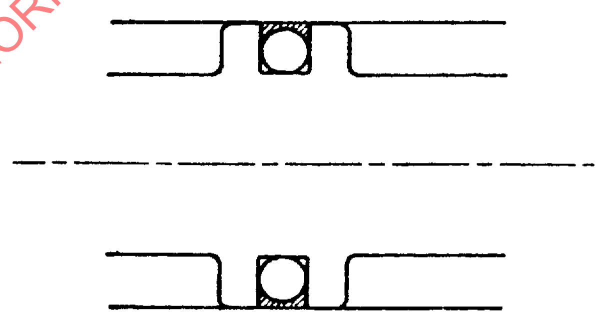


FIGURE 3.2.6.7 - Small Actuator Piston Ring Leakage Blowby

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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

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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

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3.2.7 Tubing and Fittings: This section contains lessons learned topics relating to hydraulic tubing and fittings. The lessons learned are presented in Figures 3.2.7.1 through 3.2.7.5.

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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

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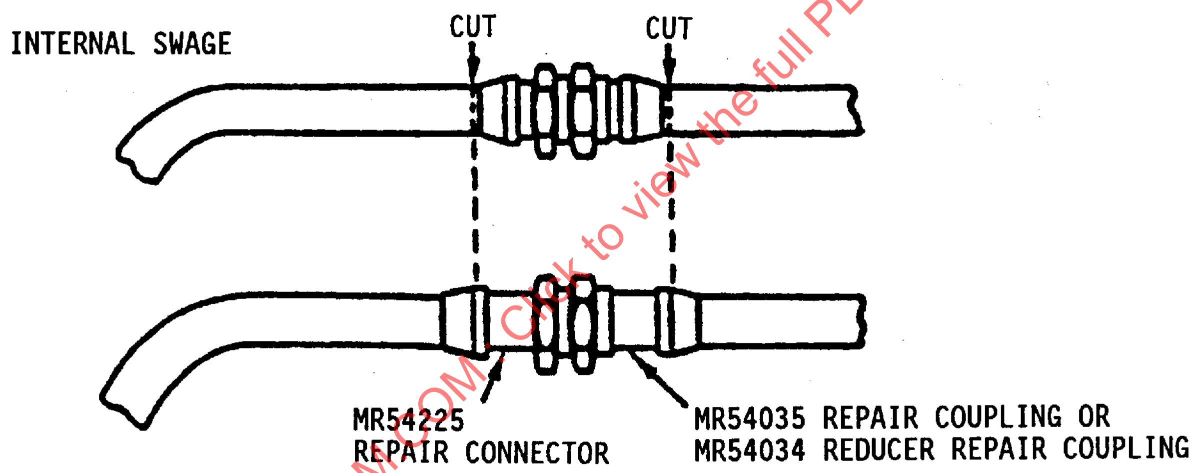
PROBLEM: LEAKING FITTINGS

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.

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FIGURE 3.2.7.2 - Leaking Fittings

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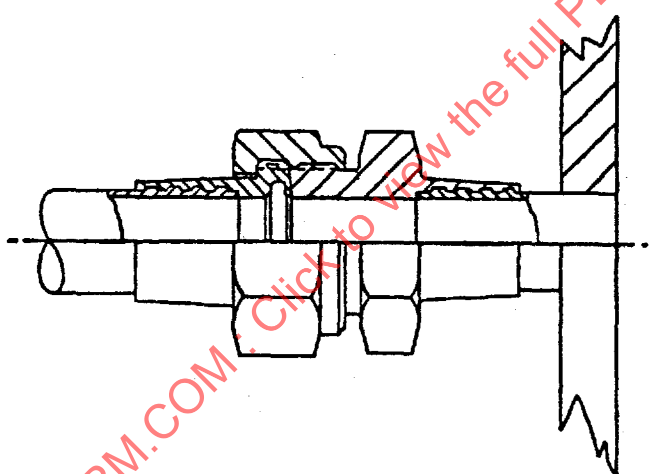
PROBLEM: MALE/FEMALE LIP SEAL CONNECTION DAMAGE

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:



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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

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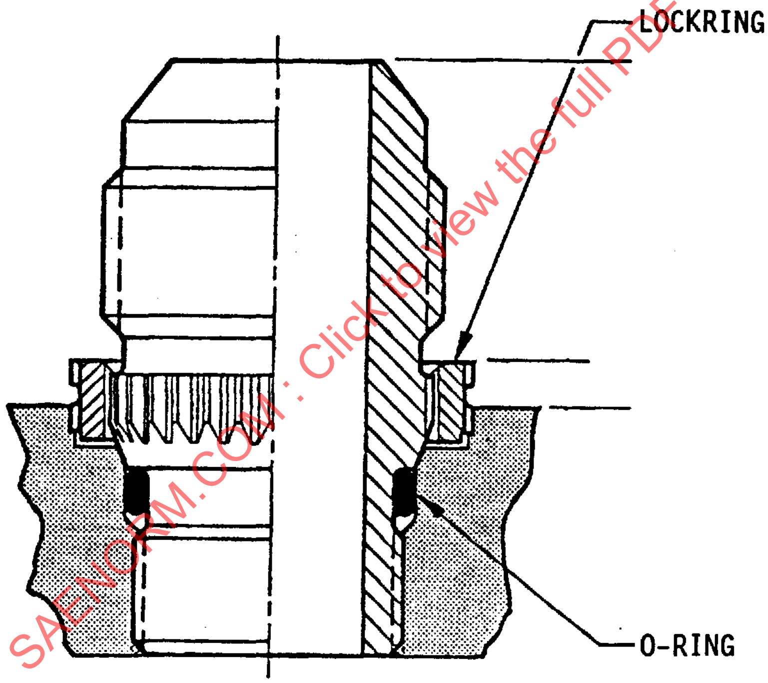
PROBLEM: DAMAGED PORTS

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

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