AEROSPACE INFORMATION REPORT

AIR1922™ REV. B

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Superseding AIR1922A

(R) Aerospace - System Integration Factors That Affect Hydraulic Pump Life

RATIONALE

Revision B of this document reflects various technical and editorial corrections and improvements that were identified during its 5-year review.

AIR1922B has been reaffirmed to comply with the SAE Five-Year Review policy.

FOREWORD

The information provided in this Aerospace Information Report (AIR) is compiled from various sources and is based on the experience and judgment of engineering and operating personnel of aircraft hydraulic components and systems.

This document is intended to be informative when considering hydraulic pump life and performance since there are no hard and fast rules for determining the optimum combination or arrangement of all the components in each hydraulic system.

However, the criteria to be followed and the operating parameters to be utilized when designing a hydraulic system remains with the designer.

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1. SCOPE

This SAE Aerospace Information Report presents the following factors that affect hydraulic pump life and performance:

a. The need to supply hydraulic fluid at the correct pressure and quality to the pump inlet port

b. Considerations for the pump output

c. Factors to be considered for the pump case drain lines

d. The mounting of the hydraulic pump

e. Hydraulic fluid properties, including cleanliness

1.1 Field of Application

This document applies to pumps operating constantly or for long periods at quasi-constant speed such as engine driven pumps or within narrow speed ranges such as electric motor driven pumps.

Wide speed range and intermittent duty cycle pumps, such as those used in Electrohydrostatic Actuators, are not covered by this document.

2. REFERENCES

2.1 Applicable Documents

The following publications form a part of this document to the extent specified herein. The latest issue of SAE publications shall apply. The applicable issue of other publications shall be the issue in effect on the date of the purchase order. In the event of conflict between the text of this document and references cited herein, the text of this document takes precedence. Nothing in this document, however, supersedes applicable laws and regulations unless a specific exemption has been obtained.

NOTE: Appendix A provides a list of documents that provide additional hydraulic system design information. It should be noted that some documents may be difficult to obtain.

2.1.1 SAE Publications

Available from SAE International, 400 Commonwealth Drive, Warrendale, PA 15096-0001, Tel: 877-606-7323 (inside USA and Canada) or +1 724-776-4970 (outside USA), www.sae.org.

AS595	Aerospace – Civil Type Variable Delivery, Pressure Compensated, Hydraulic Pump
AIR810	Degradation Limits of Hydrocarbon-Based Hydraulic Fluids, MIL-PRF-5606, MIL-PRF-83282, MIL-PRF-87257 Used in Hydraulic Test Stands
AS1241	Fire Resistant Phosphate Ester Hydraulic Fluid for Aircraft
AIR1362	Aerospace Hydraulic Fluids Physical Properties
AS4059	Aerospace Fluid Power – Contamination Classification for Hydraulic Fluids
ARP4205	Aerospace Fluid Power - Hydraulic Filter Elements - Method for Evaluating Dynamic Efficiency with Cyclic Flow
AIR4543	Aerospace Hydraulics and Actuation Lessons Learned
AIR5277	Aerospace Fluid Power - Waste Reduction Practices for Used Phosphate Ester Aviation Hydraulic Fluid

ARP5376 Methods, Locations and Criteria for System Sampling and Measuring the Solid Particle Contamination of Hydraulic Fluids

AIR5829 Air in Aircraft Hydraulic Systems

AS19692 Pumps, Hydraulic, Variable Flow, General Specification For

2.1.2 ISO Publications

Available from American National Standards Institute, 25 West 43rd Street, 4th Floor, New York, NY 10036, Tel: 212-642-4900, www.ansi.org.

or if using the Switzerland address:

Available from International Organization for Standardization, ISO Central Secretariat, 1, ch. de la Voie-Creuse, CP 56, CH-1211 Geneva 20, Switzerland, Tel: +41 22 749 01 11, www.iso.org.

ISO16889 Hydraulic Fluid Power - Filters - Multi-Pass Method For Evaluating Filtration Performance of a Filter Element

2.1.3 RTCA Publications

Available from RTCA, Inc., 1150 18th Street, NW, Suite 910, Washington, DC 20036, Tel: 202-833-9339, www.rtca.org.

RTCA/DO-160 Environmental Conditions and Test Procedures for Airborne Equipment

2.1.4 U.S. Government Publications

Copies of these documents are available online at http://quicksearch.dla.mil.

MIL-STD-810 Environmental Test Methods and Engineering Guidelines

MIL-F-8815 Filter and Filter Elements, Fluid Pressure, Hydraulic Line, 15 Micron Absolute and 5 Micron Absolute, Type II Systems, General Specification For

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

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

2.2 Related Publications

The following publications are provided for information purposes only and are not a required part of this SAE Aerospace document.

2.2.1 SAE Publications

Available from SAE International, 400 Commonwealth Drive, Warrendale, PA 15096-0001, Tel: 877-606-7323 (inside USA and Canada) or +1 724-776-4970 (outside USA), www.sae.org.

ARP4752 Aerospace - Design and Installation of Commercial Transport Aircraft Hydraulic Systems

ARP4925 Aerospace - Design and Installation of Commercial Transport Helicopter Hydraulic System

AS5440 Hydraulic Systems, Military Aircraft, Design and Installation Requirements for

2.3 Definitions

DISSOLVED AIR: This is air that is physically in solution in the hydraulic fluid.

ENTRAINED AIR: This is air in the form of bubbles that are typically less than 0.04 inch (1 mm) in diameter that are dispersed in the hydraulic fluid.

FREE AIR: This is a pocket of air that is trapped in part of a hydraulic system.

NOTE: Refer to AIR5829 for more information on these different forms of air in hydraulic systems.

HYDRAULIC PUMP STABILITY: This is the freedom from persistent or quasi-persistent oscillation of the delivery control mechanism.

OUTGASSING PRESSURE: This is the pressure below which dissolved gas begins to separate from the fluid and becomes free or entrained.

RESPONSE TIME: This is the time required for the change of displacement of the pump when subjected to a specific discharge circuit transient.

NOTE: On most pump models it is not possible to measure the pump displacement. Instead, a test can be carefully designed wherein it is possible to deduce the change in displacement of the pump by the inspection of the pump discharge pressure. This measurement is called “Response Time” as it is a dynamic characteristic of a pump operating in a specified discharge circuit. It is not strictly an attribute of the pump. In such a test, the response time is the time interval between the instant when an increase (or decrease) in discharge pressure change initiates; and the subsequent instant when the discharge pressure reaches its first maximum (or minimum) value.

SATURATION PRESSURE: This is the static pressure below which there occurs a net migration of molecules (normally atmospheric) to the gaseous state (outgassing).

NOTE: The steady-state saturation pressure of hydraulic fluid at rest in an open container is the local atmospheric pressure. The saturation pressure in a system may not be homogeneous and may change over time.

VAPOR PRESSURE: This is the pressure below which the fluid changes phase and forms vapor bubbles.

3. PUMP INLET LINE

3.1 General

Any positive displacement pump designed to flow a liquid requires some minimum pressure at its inlet port to produce satisfactory performance. The magnitude of this minimum pressure (expressed in terms of absolute pressure such as psia (kPa abs)) is a function of the pump's design, shaft speed, duty cycle, and the viscosity of the liquid being pumped.

If the pump is mounted inside an open reservoir of liquid, the inlet pressure will be atmospheric pressure plus the "head" of liquid above the pump. If such an installation does not provide the necessary inlet pressure, the reservoir could be closed and pressurized by means to the pressure required by the pump for the particular set of operating conditions.

NOTE: If hydraulic fluid is supplied to the pump by a suction line in the reservoir, then there should be an analysis of the pressure drop (see 3.6.4) and fluid acceleration (see 3.6.2) to check that there is sufficient pressure available to avoid cavitation of the pump.

For numerous reasons, it is often impractical to mount the pump inside a reservoir, thus the need for an inlet line to connect the source of liquid with the pump inlet port. Furthermore, in practice the medium is not a pure liquid but a mixture of fluids, as there is always some air mixed in it. To the extent that the inlet line affects the net inlet pressure and quality of the fluid available at the pump, it has a significant effect on the pump's performance and life. (The term “quality” here refers to the form of air mixed in the liquid.)

The primary factors to be considered so as to ensure that the inlet line is optimized for supplying fluid to the pump are:

- its cross-section;
- length, routing;
- restrictions;
- air tightness.

3.2 Inlet Pressure

Despite the many improvements made to hydraulic fluids (e.g., lubricity, shear strength, viscosity stability, etc.), one physical characteristic cannot be changed, and that is the complete lack of tensile strength of the fluid. It is, therefore, impossible to move a quantity of liquid, larger than a drop, by pulling on it and to move a fluid it has to be pushed, that is, propelled by pressure behind it – even to the inlet of a pump.

In order to obtain a satisfactory performance and life from a positive displacement pump, its internal pumping elements should remain filled with fluid, ideally as close as possible to a pure liquid. To achieve this, not only should free air be avoided, but also the fluid environment has to remain above the fluid's outgassing pressure and especially above its vapor pressure. For the fluids of interest, even at their maximum system operating temperature, the vapor pressure is typically below 1.0 psia (6.9 kPa abs), which may sound too low to be of practical concern. However, this low pressure can easily occur at the valving surfaces inside a pump operating at high speeds or during flow transients in long inlet lines.

NOTE: Airframe inertial accelerations can also act on the fluid mass and that the momentum of flowing fluid are also factors.

Between the pump inlet port and the actual internal pumping elements, there are, of necessity, fluid passages of discrete dimensions. In addition, there is a valving mechanism to separate the inlet area from the high pressure discharge area. These generate a restriction to fluid flow, hence a pressure drop. Consequently, to maintain close to saturation pressure in the pumping elements, the pressure at the inlet port should be higher. The resulting pressure for practical purposes is designated as the "Critical Inlet Pressure of the Pump."

The magnitude of the critical inlet pressure will vary as a function of pump design, operating conditions and fluid characteristics. It should be noted that at low temperatures (typically between 32 to 64 °F (0 to 16 °C)), the fluid flow tends to become laminar, thus increasing the critical inlet pressure due to fluid viscosity effects within the pump.

Typical critical inlet pressure (P_{crit}) requirements are shown in Figure 1 for various pump sizes. The critical inlet pressure for a specific pump should be readily available from the pump manufacturers.

Any operation below this critical inlet pressure will cause pump cavitation with the subsequent reduction in outlet flow, increased pressure ripple, and will result in severe damage to the pump. The amount of damage depends on how much the inlet pressure was below the critical value, and for how long. While periods of only a few milliseconds will cause only limited damage, the destructiveness is cumulative and can manifest itself as serious damage in less than a hundred hours.

It is also important to remember that the measured pressure at the pump inlet may appear adequate even though the fluid contains substantial amounts of undissolved air. In such cases, pump damage will still occur.

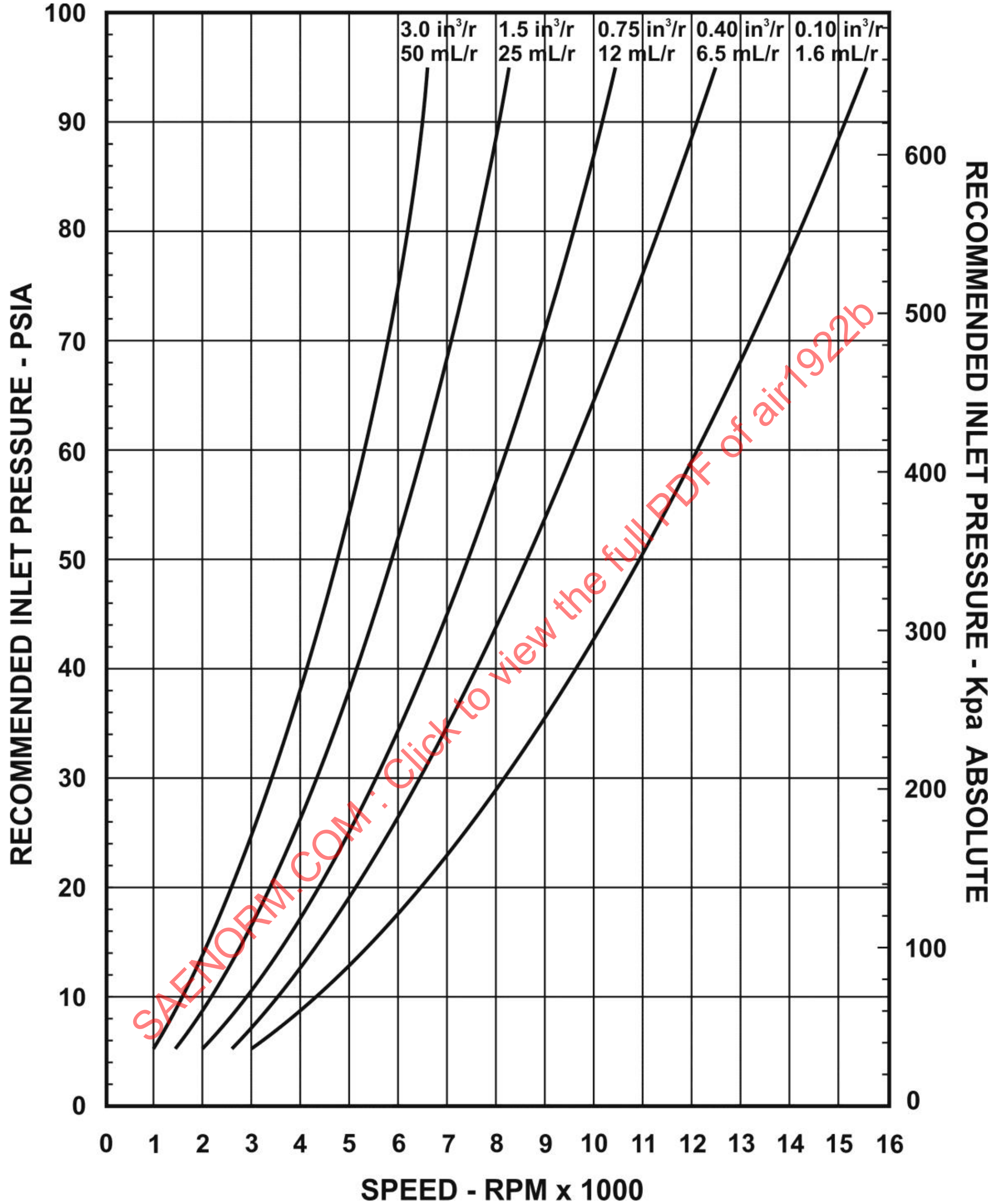

Figure 1 - Typical critical inlet pressure values for inline pumps

NOTES:

1. Consult the hydraulic pump supplier for the definitive critical inlet pressure for a specific pump application.

2. The inlet pressures shown in Figure 1 are for pumps operating with MIL-PRF-83282, and MIL-PRF-87257 hydraulic fluids at nominal fluid temperatures. An increase of 30% is recommended for operation with AS1241 hydraulic fluids to allow for their increased density compared to the MIL-PRF fluids.

3.3 Cavitation

If the inlet fluid contains entrained (undissolved) gas, typically air, or if the inlet pressure of the fluid is allowed to drop below its outgassing pressure so that dissolved air comes out of solution or worse yet, below its vapor pressure, there will be bubbles. After these enter the pumping elements and are then compressed to pump discharge pressure, they will collapse violently and generate a shockwave. In the case of piston pumps, this violent release of localized energy will manifest itself as cavitation erosion in such areas as:

- cylinder blocks/cylinder barrels;
- valving plates/port plates;
- shoes/slippers;
- shoe bearing/wear plates.

The occurrence of cavitation is clearly noticeable by its characteristic noise and line vibration. However, in the case of pressure transients, the noise and vibration may be masked by other factors. Damage to the pumping elements as a result of cavitation consists of small particles of material being removed from their surfaces and subsequently carried by the fluid through the discharge and case drain ports into the system. Since there is material removed, the damage is cumulative.

The material removal can cause changes to pump timing and hydrostatic balance, which in turn can cause premature wear of the sliding surfaces. Additionally, the wear debris in suspension may score other hydrodynamic sliding surfaces thereby creating more debris. Most of the contaminants will then circulate through the pump bearings precipitating their premature failure.

It is therefore important that pump cavitation is avoided at all times, including during transient conditions. The design of the system and components should ensure that the required inlet pressure on the fluid at the pump is always maintained.

3.4 Air in Hydraulic Fluids

The hydraulic system can contain air in three forms:

a. dissolved air;

b. entrained air;

c. free air.

The first two were commented on as they affect cavitation, and will be cited again in relation to inlet line configuration.

Extensive documents including AIR5829 have been written on air in hydraulic fluids, including experience with quantitative amounts of dissolved and entrained air.

Free air in a hydraulic system can also cause extensive damage to pumps as well as to other components. It does so primarily by interrupting the lubricating fluid film in critical areas, thereby allowing metal-to-metal contact.

Aside from pump damage, free air (often manifested as foam) can have serious operational effects such as:

- system instability;
- increased pressure ripple;
- the loss of pump prime;
- the inability of the pump to maintain flow and pressure;
- damaging the hydraulic fluid over time because when free air is compressed to design operating pressure, will also create a very tiny, but extremely hot, pocket of gas.

The presence of free air is often due to inadequate bleeding of the system; however, poorly thought-out design details can make adequate bleeding next to impossible.

System design details that will help avoid serious air problems include:

a. Ensure that the pump case drain port is located in the upward direction.

b. Avoid non-uniformity in line size and sharp bends in the pump inlet line.

c. As hoses generally cause a larger pressure drop than tubing, they should be used with caution.

d. Design tubing routes to prevent air, water, and particulate entrapment and avoid vertical loops in the inlet line as this increases the risk of having air traps which make pump priming and system bleeding very difficult and sometimes impossible during adverse conditions.

e. Ensure that hydraulic controls elevated above the reservoir can be adequately bled during ground maintenance activities.

f. Install bleed valves at high points in the hydraulic system to remove entrapped air.

NOTE: Bleed valves are normally installed where there is an air trap combined with low pressures and low flow rates.

g. Minimize the number of fittings in pump inlet lines. While these may not leak hydraulic fluid, they can ingest air.

h. With bootstrap reservoirs:

1. Provide low piston friction to reduce the vacuum effect in the inlet line as the hydraulic system cools down.
2. Provide sufficient auxiliary force (i.e., spring, gas pressure, etc.) to retain the reservoir piston against the fluid when the hydraulic system is depressurized.
3. Provide automatic bleed valves permanently mounted at the highest point of the reservoir to bleed free air.
4. Provide an in-system air eliminator to reduce dissolved air to below saturation conditions to further reduce the potential of free air.
5. Assure components do not include restrictors that cause air formation as discussed in AIR4543 and AIR5829.

i. With air-pressurized reservoirs:

1. Use an air to oil separator in the reservoir in order to minimize the amount of air in the hydraulic system.

   NOTE: If a separator is used, then the bootstrap reservoir requirements (h), 3, 4, and 5 apply.

2. The use of unseparated, air-pressurized reservoirs can significantly increase the dissolved air content of the system which can lead to out-gassing in the pump suction line. In this arrangement, it may be advantageous to include an integral inlet boost stage in the pump (see 3.5.1).

3.5 Boost Pumps

3.5.1 Integral Inlet Boost Stage in the Pump

When the inlet line is compromised, or reservoir pressure too low, the resulting inlet pressure may be too low for the positive displacement pump. In that case, the pump manufacturer may incorporate an integral inlet boost section, typically an impeller, into the pump.

The pressure boost capabilities of typical, well designed centrifugal impellers are in the 20 to 40 psi (138 to 276 kPa) range, depending on their size and operating speed, as well as the flow rate, fluid and temperature. When all these parameters are in their optimal combination, the critical inlet pressure of the pump combination can be reduced to the 5 to 10 psia (34.5 to 69 kPa abs) range. Effective performance without cavitation below approximately 5 psia (34.5 kPa abs) inlet pressure may require the addition of inducers or injectors, in addition to an impeller.

Integral boost mechanisms in the pump can be very effective and have the advantage of dissolving some of the entrained air, but they will add size, weight, and cost to the pump. Their design is very critical and maximum effectiveness is in a relatively narrow performance band in terms of speed and temperature, as compared to that of the positive displacement pump. In many situations, the total system would be better served by improving the inlet line (reducing its pressure drop) than by tolerating an inferior inlet line with the more complex two-stage or even three-stage pump combinations.

3.5.2 Alternatives

The following should be considered if it is not feasible or desirable to introduce an integral inlet boost section into the pump:

a. Incorporate a centrifugal boost pump near the reservoir produces one of the best inlet conditions for the main pump as it minimizes the side effects of the air in the fluid.

b. Install a charged accumulator in close proximity to the pump inlet port if a poor inlet condition occurs only during transients.

c. To allow or specify a slower pump response when high flows are required from large displacement pumps with marginal inlet pressure.

d. Fittings and quick disconnects installed close to or in the pump inlet port are to have a minimal pressure drop. (This is to prevent air coming out of solution at the pump inlet).

3.6 Inlet Line - Sizing and Configuration

The design of the pump inlet line, reservoir location, and reservoir pressurization are major system elements affecting hydraulic pump life. Once the basic needs of pump inlet pressure and the inevitable consequences of air and cavitation are understood, attention to the components in the pump inlet line becomes more meaningful.

At the reservoir end, sufficient pressure should be available to push the fluid through the inlet line such that, at the pump inlet port, the fluid pressure is still above the pump's critical inlet pressure under the worst commonly encountered conditions.

To put these conditions in perspective, it should be remembered that the response requirements of many hydraulic systems demand that a variable volume pump responds from its zero flow "dead headed" condition to full flow output in 50 to 200 ms. While most pumps have little difficulty in providing such response, it should be remembered that "you cannot get out what you do not put in". Therefore, the inlet pressurization should be such that the entire mass of fluid in the inlet line will also be accelerated in 50 to 200 ms and still remain above the critical inlet pressure at the pump inlet port.

It follows then that reservoir pressure should be determined by the following elements:

$$P_{resv} = P_{crit} + \Delta P_{acc} + \Delta P_{com} + \Delta_{Plin} + \Delta_{Psta} \quad \text{(Eq. 1)}$$

where:

P_{resv} = reservoir pressure

P_{crit} = pump critical inlet pressure

ΔP_{acc} = pressure to accelerate fluid mass in pump response time

ΔP_{com} = pressure loss of inlet line components at maximum flow

ΔP_{lin} = frictional pressure loss in inlet line and hose assembly at maximum flow

ΔP_{sta} = static pressure head

NOTE: Appendix B provides an example of calculating a required reservoir pressure that incorporates all the elements listed above.

3.6.1 Pump Critical Inlet Pressure

This is discussed in 3.2.

3.6.2 Transient or Acceleration Pressure

The transient or acceleration pressure (ΔP_{acc}) is that pressure necessary to accelerate the fluid mass in the inlet line to the maximum required flow within the response time of the pump. The pressure can be determined from the basic equation for acceleration force (F = ma) as follows:

$$m = \rho AL \ \text{(fluid density x volume)} \quad \text{(Eq. 2)}$$

$$a = \frac{dv}{dt} \ \text{(time rate of change of velocity)} \quad \text{(Eq. 3)}$$

$$F = \rho AL \text{ x } \frac{dv}{dt} = \rho AL \text{ x } \frac{\frac{dQ}{A}}{dt} = \rho L \times \frac{dQ}{dt} \quad \text{(Eq. 4)}$$

The acceleration pressure needed to produce this force is:

$$\Delta P_{acc} = \frac{F}{A} = \frac{\rho L}{\pi / 4 \times D^2} \times \frac{dQ}{dt} \qquad \text{(Eq. 5)}$$

When converted to a form that utilizes the commonly used engineering units, and assuming a linear pump response, the foregoing formula appears as follows:

$$\Delta P_{acc} = 88.1 \times 10^{-6} \ \frac{\rho L}{D^2} \times \frac{dQ}{dt} \ \text{(English units)} \qquad \text{(Eq. 6)}$$

$$\Delta P_{acc} = 21.2 \ \frac{\rho L}{D^2} \times \frac{dQ}{dt} \ \text{(metric units)}$$

where:

ΔP_{acc} = pressure required to accelerate, psi (kPa)

dQ = maximum flow change, gpm (L/min)

dt = pump response (seconds), s

L = length of tubing, feet (m)

D = tube inside diameter, inch (mm)

ρ = mass density of the fluid, lb_m/ft^3 (kg/L)

A = tube inside cross-section area, in^2 (mm^2)

3.6.2.1 Allowance for Fluid Acceleration Non-Uniformity

If the non-uniformity of the fluid acceleration due to wave effects is taken into account, then the transient pressure term ΔP_{acc}, is oscillatory, with its average value given by Equation 5, but with its maximum value given by:

$$\Delta P_{acc} = ((\rho \ c) \times (\Delta Q/A)) \qquad \text{(Eq. 7)}$$

If dQ/dt is constant over the time $2\Delta t$, then ΔQ is equal to $2\Delta t.dQ/dt$ or 2 (L/c) dQ/dt, so that substituting its value in Equation 7, one gets:

$$\text{The maximum value of } \Delta P_{acc} = 2 \ \rho \ L \ (dQ/dt)/A) \qquad \text{(Eq. 8)}$$

When converted to a form that utilizes the commonly used engineering units, and assuming a linear pump response, the foregoing formula appears as follows:

$$\Delta P_{acc} = 176.2 \times 10^{-6} \ \frac{\rho L}{D^2} \ \frac{dQ}{dt} \ \text{(English units)} \qquad \text{(Eq. 9)}$$

$$\Delta P_{acc} = 42.4 \ \frac{\rho L}{D^2} \times \frac{dQ}{dt} \ \text{(metric units)}$$

where:

ρ = mass density of the fluid, lb_m/ft^3 (kg/L)

c = wave speed in fluid = $(\beta/\rho)^{0.5}$

β = adiabatic bulk modulus, psi (kPa)

ΔQ = flow change at pump during time $2\Delta t$, in^3/s (mm^3/s)

A = tube inside cross-section area, in^2 (mm^2)

L = length of tubing, feet (m)

D = tube inside diameter, inch (mm)

Δt = time for the pressure wave to travel distance L from pump to the reservoir = L/c

NOTE: Care should be taken concerning the value of the adiabatic bulk modulus that is to be used in the analysis. This is because an allowance for the effect of air and the deformation of tube walls under pressure (especially hoses) needs to be taken into account. A value that is approximately 0.7 of the published value has been used to cater for these effects.

It should be noted that the derivation of Equation 6B assumes that the fluid pressure stays high enough for the air in fluid to stay in solution and for no fluid turning from liquid into vapor. Also, if part of the inlet line is a tube and part of it is a hose, then the equations for the maximum value of ΔP_{acc}, become more complex and the use of a dynamic analysis program may be required.

3.6.3 Component Flow Losses

Component flow losses (P_{com}) are those generated by such typical items as a shutoff valve, check valves, filters, and fittings. All of these should be avoided where possible, and certainly minimized in their effect on the pressure drop in the inlet line. The pressure drop at rated flow for each of these components is obtained from their manufacturers and treated cumulatively for a total component pressure loss value.

A less obvious effect of inlet component restriction and the resultant pressure drop is the possibility of outgassing the fluid; that is, dissolved air coming out of solution. This can be thought of as a local phenomenon. Provided there is enough pressure left and sufficient distance between the component and the pump inlet, the air can go back into solution downstream prior to reaching the pump. It is, therefore, desirable to keep all inlet line components, including quick disconnect couplings, as far away as possible from the pump to give the fluid a chance to recover from the outgassing phenomenon.

A special note of caution is given for systems with an inlet line pressure below atmospheric pressure; some fittings will seal pressure adequately from the inside, but not necessarily seal as well from the outside. In such cases, air will leak into the inlet line.

3.6.4 Line Friction Loss

The next element in the sum of inlet pressure considerations is the fluid friction loss (ΔP_{lin}) in the line (including hoses) itself. The pressure drop in a line for maximum steady state flow can be calculated by the standard formulas used for turbulent or laminar flow conditions.

For most airborne hydraulic power transmission systems, the flow rates and line sizes will be such that turbulent flow will exist during normal operation. During off-design operation at very low power, laminar flow might exist; but the pressure drop for such a case would be trivial. In the case of hydraulic starting systems for jet engines, however, the entire duty cycle may be completed before the system temperature is raised appreciably above its original level. For such systems, laminar flow may be present at large flow rates, and should be considered in the system analysis.

For laminar and turbulent pipe flow, the following expressions may be used:

a. Reynolds Number (R)

$$R = 3163 \frac{Q}{\nu D} \text{ (English)} \qquad \text{(Eq. 18)}$$

$$R = 21220 \frac{Q}{\nu D} \text{ (metric)}$$

b. Friction Factor (English and metric)

1. Laminar flow

$$f_{lam} = 64/R \qquad \text{(Eq. 19)}$$

2. Turbulent flow

$$f_{turb} = 0.316/R^{.25} \qquad \text{(Eq. 20)}$$

c. Pressure Drop per Line Length

$$\frac{\Delta P}{L} = 2.16 \times 10^{-4} f \rho \frac{Q^2}{D^5} \text{ (English)} \qquad \text{(Eq. 21)}$$

$$\frac{\Delta P}{L} = 2.25 \times 10^5 f \rho \frac{Q^2}{D^5} \text{ (Metric)}$$

where:

Q = volume flow, gpm (L/min)

D = tube inside diameter, inch (mm)

ν = fluid viscosity (kinematic), centistokes (cSt)

ρ = mass density of the fluid, lb_m/ft^3 (kg/L)

ΔP = pressure drop, lbf/in^2 (kPa)

L = length of tubing, feet (m)

V = flow velocity, in/s (m/s)

f = friction factor

NOTES:

1. When considering the pressure drop for flows that could be laminar or transient, an approach that could be used when using computer spreadsheets is to assume that for R<1187, assume the flow is laminar and use Equation 19; otherwise assume the flow is turbulent and use Equation 20. This ensures continuity between the laminar and turbulent equations (as no hysteresis considered in the transition) and provides a potential over-estimation of the pressure drop, thereby introducing some conservatism in the analysis results.

2. In laminar flow conditions, the pressure drop should be increased to include the transition length (the distance to get a fully established laminar flow from the entrance of the suction line). For suction lines whose length is shorter than 50 diameters), this effect can be important.

3. Turbulent flow - for pressurized lines, increase the basic pressure loss by 3% per 1000 psi pressure

4. Laminar flow - for pressurized lines, increase the basic pressure loss by 15% per 1000 psi pressure.

It is recommended that the maximum pump flow at maximum application speed (not the maximum demand on the system) be used to calculate the steady state line loss. Line losses should be determined at the lowest temperature at which full system operation is required. This temperature is often -20 °F (-29 °C) for military and commercial aircraft applications. Fast response aircraft or missile systems may require full system operation at lower temperatures. Normally, an aircraft system is allowed time to warm to the -20 °F (-29 °C) operating temperature prior to requiring large flow demands from the pump.

3.6.5 Static Pressure Head

If the pump height is more than a few inches or centimeters above that of the reservoir, the static pressure head (ΔP_{sta}) should be added to the required reservoir pressure.

$$\Delta P_{sta} = 0.0007\ \rho\ h_1\ \text{(psi)} \qquad \text{(Eq. 10a)}$$

where:

ρ = density of the fluid in lb_m m/ft^3

h_1 = height difference in feet

or

$$\Delta P_{sta} = 9.8\ \rho\ h_1\ \text{(kPa)} \qquad \text{(Eq. 10b)}$$

where:

ρ = density of the fluid in kg/L

h_1 = height difference in meters

4. PUMP OUTLET LINE

Factors in the pump outlet (pressure) line that affect pump life include:

a. pressure pulsations;

b. pump-system response;

c. using a check valve in the pump outlet line;

d. parallel pump installation.

4.1 Pressure Pulsations

4.1.1 General

The pulsations in the output flow of a piston-type hydraulic pump are transmitted throughout the pressure side of the hydraulic system. The dynamic resistance (impedance) of the hydraulic system fluid translates the flow pulsations into pressure pulsations, also known as pressure ripple.

Unsteady flow is produced by a positive-displacement pump due to two factors:

a. The kinematic flow ripple arises due to the summing of the sinusoidal output flows of the pumping elements. For an odd number of pistons (n), the percentage of flow ripple due to kinematic effect can be estimated as $125/n^2$ (for example 11 pistons equate 1.03%). The frequency of the kinematic flow ripple for pumps with an odd number of pistons is 2 times the piston pass frequency.

   The piston pass frequency is equal to the product of the drive speed and the pumping elements.

   Example: A nine piston pump rotating at 5000 rpm has a piston pass frequency of 750 Hz.

$$\frac{9\ \text{pistons} \times 5000\text{rpm}}{60\text{s/min}} = 750\text{Hz} \tag{Eq. 22}$$

b. The other much larger source of flow ripple is due to the compressibility of the fluid. The opening of the pumping element that is transitioning from low pressure to high is delayed in order to raise the chamber pressure to near the system pressure. This is called the pump timing. This delay, together with any mismatch in system pressure and pump chamber pressure when the pumping element is open, causes flow oscillation. Unfortunately, this ripple is a very transient effect that cannot be calculated without resorting to dynamic simulations of the pump and the system, although it can be a significant contributor to the overall system pressure ripple.

In addition to the piston pass frequency as determined above, significant pulsation amplitudes are often observed at the first, second, and third harmonics as well, and these should also be considered in the design of the system installation. Hydraulic flow/pressure pulsations excite high frequency mechanical motion in lines and components. When the frequency (speed) and harmonics of the acoustic source (pump) is equal to a natural frequency of the hydraulic fluid column, a resonance condition occurs. Pressure pulsations at resonance may be very high, for example, ±1000 psi (6900 kPa) in a 3000 psi (20 700 kPa) system. These resonant conditions are potentially very destructive to the system hardware as excessive motion causes stresses that may result in failure of lines, components and/or mounting hardware. In extreme cases, the pressure vessel (lines or components) may reach fatigue failure limits in a few minutes. Therefore, changes to the system and/or pump may be required to reduce pressure pulsations at system resonant frequencies when they occur at primary operating conditions (e.g., idle, cruise, etc.) but not necessarily at transient conditions.

Consideration should be given to the pump's operation outside the nominal operating conditions of speed, pressure, or temperature ranges. The pump and system may be compatible under the normal operating conditions but affect resonances under other conditions that produce high levels of pressure ripple. These may be of short duration during the start-up phase that it is of no serious concern but any quasi-persistent condition should be treated with care, as the pulsation frequencies involved are high. A high number of cycles can be accumulated over the life of an aircraft.

Pressure pulsations in the outlet line can also have an effect on pump life as high frequency resonant pulses reflected back to the pump can cause the pump compensator mechanism to oscillate at the high frequency, with resultant rapid wear of the pump parts. Test data has illustrated that lower pressure pulsations result in lower pulses being reflected back to the pump from the system. It has also been demonstrated in some systems that higher pressure pulsations reduce pump life as well as the life of other components. The failure of components downstream of the pump can also result in reduced pump life.

Pressure pulsations also occur in the pump inlet/return system. Normally, the inlet side of the system is not a problem because of the low pressure and low energy level at which it operates. However, thin wall tubing, high installation stresses, and an adverse resonance condition could produce failures in the inlet side of the system.

The coupling of hydraulic to mechanical resonances is a complex phenomenon, and is a function of line size and material, configuration (routing), and installation constraints. The total stresses in lines are a combination of hoop stress from internal pressure and bending stress due to installation and induced vibration. Large pump outlet and inlet lines having an outside diameter (OD) of 1.0 in (25 mm) and above are particularly vulnerable to high installation stress. The total combined line stress should be low enough to provide infinite fatigue life.

4.1.2 Acceptable Level of Pressure Pulsations

An acceptable level of pressure pulsation in one system may not be acceptable in another system due to differences in mechanical response. High frequency line motion induced by pressure pulsations cannot be controlled by normal line support techniques such as clamps. Clamps should be designed to withstand the line vibration without wearing out the cushion between the clamp and line or causing chafing through the line. Pressure pulsation requirements are often specified to be <10% and <5% (peak-to-peak) of rated pressure for military and commercial aircraft respectively.

The following relates central system pulsation level to potential problems.

a. >1100 psi (>7586 kPa) (peak-to-peak): Rapid failure of pump discharge line due to pressure and vibration stresses. Possible failures of mounting structure and internal functions of central components

b. >600 psi (>4138 kPa) (peak-to-peak): Line clamp cushion wear out, line failure due to clamp chafing, poor clamp life, frequent inspections required, discharge line check valve wear out

The first category (>1100 psi (>7586 kPa) (peak-to-peak)) is a potential safety of flight situation. The second category (>600 psi (>4139 kPa) (peak-to-peak)) is one of nuisance level problems that probably surfaces only after a considerable amount of operational flight experience. The best approach is to verify the acceptable line stresses on the systems laboratory simulator and the first flight aircraft to preclude failures of the first category. Changing the line length from the pump outlet port to the filter or other simple plumbing changes may be identified to relocate resonances away from continuous operating speeds.

If stress levels due to pressure pulsations are not acceptable, wide-band attenuators or hoses for mechanical decoupling should be considered. Hoses may provide mechanical decoupling which can eliminate clamp/line wear out problems. However, one should accept the penalties when using hoses instead of tubing, such as limited life, added weight and cost, and larger diameters.

It should be noted that away from the high acoustic energy of the pumps, gearboxes, and engines, the vibration environments are relatively benign. Good line support and adequate clearances between lines, and lines and structure, precludes significant vibration related problems.

4.1.3 Means to Reduce Pressure Pulsations

Techniques that significantly reduce system pressure pulsations over the operating speed range of the pump include attenuation components such as:

- Adding a flow-through volume in the pump outlet port - this can generate anti-phase pulsations that will damp out most of the pulsations generated by the pump.

- Adding a stand-off volume (such as a teed-off line) downstream of the pump outlet port - this can also generate anti-phase pulsations, but at only at a specific pump speed.

  NOTE: The attenuating means should be located as close to the source (pump) as possible for maximum protection of the downstream system as well the pump itself. If attenuators are to be installed at some downstream distance from the pump, the hydraulic line between the pump and attenuator should be checked to verify that no resonant conditions exist.

- Adding a filter close to the outlet port - the filter element absorbs all the pressure pulsations and provides a ripple free output.

- Adding an accumulator to the central system which can damp out any resonances in the pressure line.

- Making some detail changes inside the pump to fine tune the pump to the system.

Restrictors may also be used to attenuate pulsations in a dead-ended line such as that terminated by a static pressure transmitter. When a transmitter is located at the end of a line, a small restrictor should be placed at the acoustic source (i.e., the "T" where the dead-end line begins). This may protect the entire line and transmitter from significant acoustical energy. A secondary benefit is attenuation of system transient pressure spikes that may cause erroneous cockpit indications on a poorly damped pressure gage.

A check valve may be used to "acoustically" isolate a dead-end ground service line that leads to the system pressure supply. Again, the check valve should be placed at the downstream end of the service line adjacent to the main line. This will isolate the service line from pressure pulsations generated by the onboard system pump during normal aircraft operation.

4.2 Hydraulic Pump Response Characteristics

Each variable displacement hydraulic pump is a highly under-damped servomechanism with a transient response that is dependent on many parameters. System and pump characteristics that affect response (time rate of pressure change (dp/dt)) are:

a. fluid bulk modulus;

b. fluid volume under compression;

c. line characteristic impedance;

d. system load characteristics;

e. pump speed;

f. pump leakage;

g. pump flow gain;

h. pump/compensator gain;

i. initial steady state pressure and flow.

These factors also play a critical part in determining pump stability. When system flow demands change, corresponding transient pressure changes will also occur. These perturbations in line pressure are sensed by the pump pressure compensator that then tries to compensate for the changes in pressure by adjusting the flow.

The change in flow depends on how far the line pressure is away from a reference value set in the pump. The speed of response is defined as the time required for the pump control to maintain pressure in either direction (zero flow to full flow or full flow to zero flow demand) in the pressure compensated range.

Because of the many parameters that affect pump response and stability, the hydraulic system designer should adequately specify the circuit and the performance required. The pump performance should be such that any pressure overshoot is limited when pump demand is reduced from full flow to zero flow. Figure 2 presents a typical transient pressure – time characteristic as the pump responds to the system flow demand from full flow to zero flow. Under these conditions, the system relief valve limits the pressure overshoot should the pump compensator fail. With sudden flow demands, such as zero flow to full flow, the pump should respond to limit system pressure reduction to a minimum value. Figure 3 presents a typical transient pressure – time characteristic as the pump responds to the system flow demand from zero flow to full flow. The pump should also provide 90% of steady state full flow pressure within a specific time period. Note that the design of the inlet line also plays a key role in this characteristic.

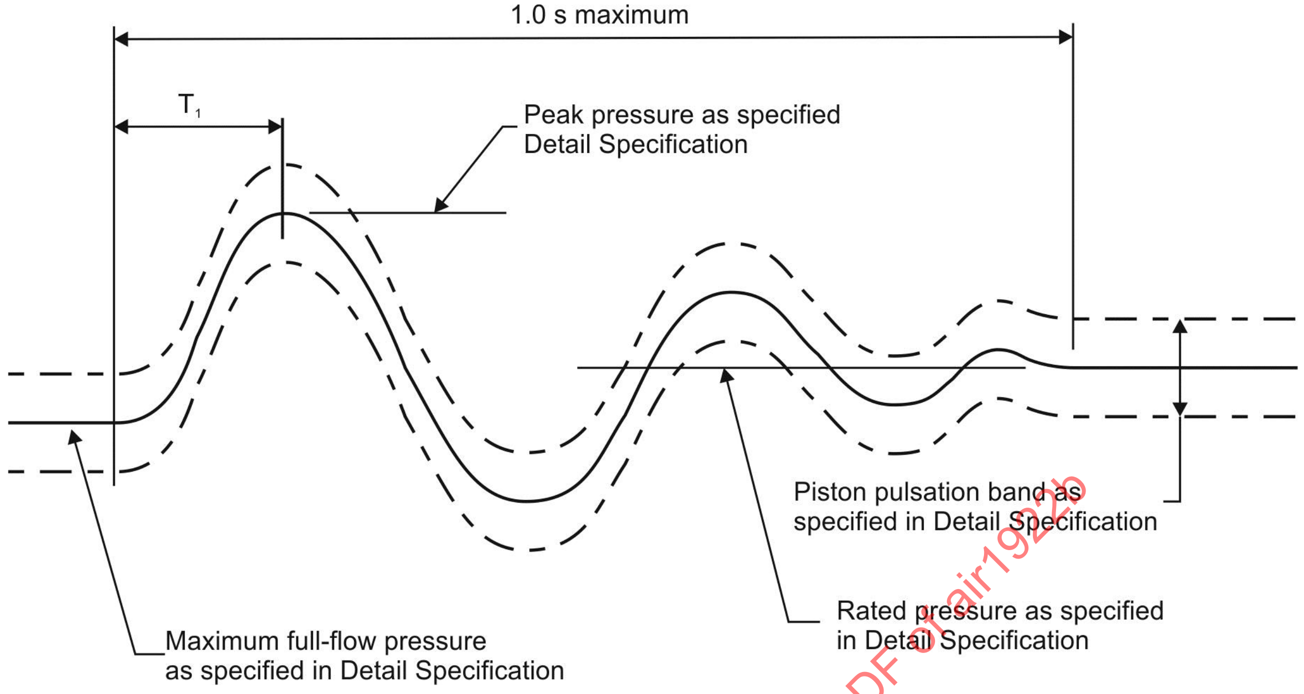

Figure 2 - Typical pressure transient full flow to zero flow

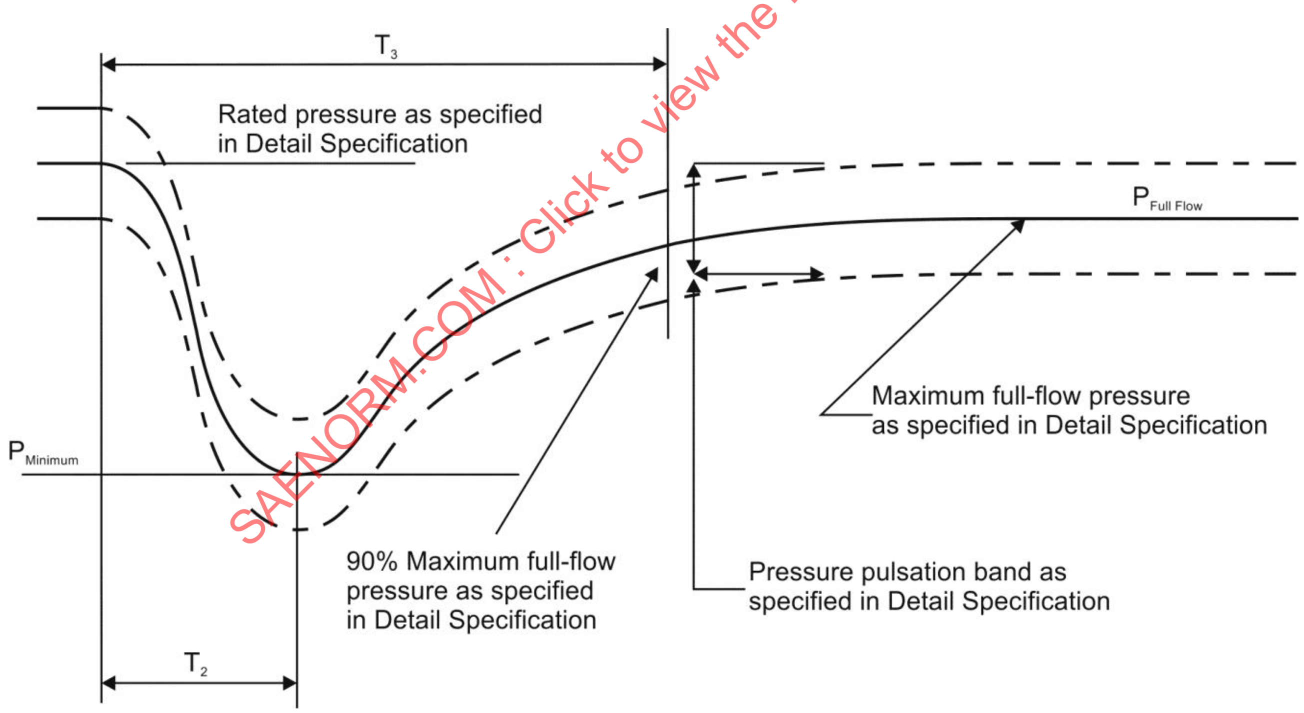

Figure 3 - Typical pressure transient zero flow to full flow

More information on pump response characteristics is contained in AS595 and AS19692.

4.3 Check Valve in Pump Outlet Line

The use of a check valve in the pump outlet line is to prevent back flow that can cause back drive motoring of the pump (such as when external hydraulic power supply ground support equipment is pressurizing the hydraulic system while the engine-driven pump is not rotating) and protect the pump from adverse system pressure spikes.

The location of a check valve in the outlet line can greatly affect the resonant characteristics of the pump and the system. This is particularly significant in systems that have zero flow conditions. During the transition from a flow to a zero flow condition the high transient pressure produced by the pump passes over the check valve.

When the pump regulates its output pressure back to nominal the check valve prevents back flow from the system, trapping this higher transient pressure in the system, which has two effects:

a. The system pressure indication is too high.

b. The pump is exposed to a very small system volume up to the check valve which may create stability problems.

Even without the transient pressure effect, a check valve that is located very close to the pump may integrate the normal pressure ripple of the pump until the pressure recorded downstream of the check valve reaches the level of the maximum peak of the pressure ripple.

4.4 Parallel Pump Installations

In parallel pump installations, means should be provided to have a master/slave relationship between the two pumps in order to prevent any interactions with each other, and hence eliminate any problems of pump instability.

One way to operate two parallel pumps in this master/slave relationship is to install a higher cracking pressure check valve in the outlet line of one of the pumps. For small flow demands, the pump with the lower cracking pressure check valve will supply the flow; however when a large flow demand occurs, the system pressure will reduce which then enables the pump with the higher cracking pressure check valve to also supply fluid.

The check valve should be designed with adequate damping to preclude poppet instability around the cracking point.

5. PUMP CASE DRAIN LINE

5.1 General

The function of the pump case drain line is primarily the removal of internal leakage from the pump. In so doing, the case drain line provides a means of rejecting heat from the unit, and offers a means of monitoring pump performance. Factors to be considered in the design of the case drain lines include the normal hydraulic tubing practices that are observed for all hydraulic system lines in addition to the guidelines and factors which follow.

5.1.1 Internal Case Drain Line

Some pumps may not have an external case drain line. Rather, the pump case is connected internally to the low pressure suction port. This configuration unit may only be used in applications in which the pump is always delivering some output flow to the system high pressure line. The system duty cycle flows or quiescent flow leakage should be sufficient to reject the heat generated by the pump.

NOTES:

1. Variable delivery pumps with internal drains may be operated at zero flow for short periods of time only (i.e., to set/adjust pressure controls).

2. A fixed displacement pump may safely incorporate an internal drain.

5.2 Case Drain Pressure

The pump envelope is directly affected by the internal case pressure level, whereby higher pressures require correspondingly higher design strength in the housing, interfacing joints, and fasteners. The effect of case pressure on the static seals and drive shaft seal members of the pump should be considered and the impact on the unit sealing integrity evaluated

The case pressure imparts an axial thrust on the drive shaft equal to the product of the shaft seal differential area (area exposed to case pressure minus atmospheric pressure) times the case pressure level. This load is transmitted through the main shaft bearing thereby affecting the life of that bearing.

Most hydraulic pump cases are designed to withstand operating pressures up to 500 psi (3450 kPa) and a minimum ultimate (burst) pressure of 1250 psi (8625 kPa). The normal operating level is usually much lower than the level specified for structural integrity.

Back pressure (i.e., the differential pressure between case pressure and pump inlet pressure) has a direct effect on the balance and loading of the pump rotating group. An axial force is exerted on each pumping piston during the suction stroke by a positive case drain back pressure. The piston return mechanism should be designed to withstand this force in addition to the inertia and friction forces inherent to the design.

High back pressure can also cause the pump internal leakage to be "short circuited" back to pump inlet through the piston shoes (slippers) during the suction stroke rather than being vented out the case drain line. This results in a loss of pump heat rejection capability that may cause overheating during low outlet flow operation and the subsequent deterioration of the fluid and pump internal parts.

Ideally, the case drain pressure and inlet pressure should be comparable. The effect of extended operation with high back pressures should be fully understood in the design and use of the pump especially in the low outlet flow modes.

In some designs, an integral relief valve between the pump case and the low pressure inlet port is incorporated. The primary purpose of the relief valve is to relieve a steady-state case over-pressurization in order to prevent the rupture of the case housing.

5.2.1 Case Drain Scavenge Pumps

The use of a scavenge pump in the case of the pump is advantageous in systems with potentially persistent high case drain line pressures. It prevents the high case drain line pressures being applied to the rotating group of the pump and assures adequate case drain flow under all conditions to provide optimum cooling of the pump and flow through the system heat exchanger if one is located in the case drain line of the pump.

Care should be exercised in the pump design either to avoid applying the pressure produced by the scavenge pump to the shaft seal of the pump or to design the seal and shaft bearing to accept these pressures and loads.

5.3 Case Drain Line - Direct to Reservoir and Return Line

Case drain lines should be sized to pass the maximum determined case drain flow at pressures at or below the established allowable case pressure for the pump. Factors affecting sizing of the line include the line length to the point of return at the reservoir, and the components in the line (i.e., filter, cooler, valves, etc.). All conditions and modes of pump operation should be considered when establishing the maximum case drain flow. Flow spikes due to the movement and venting of the pump controls during transient operation should also be determined.

A case drain line with an in-line filter and cooler and connected directly to the system reservoir provides the simplest system design and effectively isolates the case drain. Design consideration, such as long line lengths, etc., may dictate connecting the case drain line to the return line or some other low pressure line in the system. The case should be protected from system pressure spikes or back flow by proper valving, check valves, etc., if this potential exists

The system designer should specify system characteristics and that pump testing should be performed in systems incorporating realistic case drain characteristics. The case drain line characteristics should be provided on a plot of flow versus pressure drop between the pump case drain port and the reservoir, or the point of return.

The case drain line often provides the optimum location for the system cooler due to the higher temperature levels and total fluid flow normally associated with this line. Adequate flow for cooling should be maintained to assure proper cooling at all conditions and a filter upstream of the cooler should be utilized to prevent contamination of the cooler which could result in higher case pressures and loss of cooling capacity.

5.4 Case Drain Flow Change

The case drain flow offers the best indicator of the pump's condition. The deterioration of the running/sealing surfaces in a pump will be reflected in an increase in case flow due to increased internal leakages. The detection of this increased flow allows for the early removal of a unit prior to catastrophic failure (i.e., one in which the pump is lost, system hydraulic power lost, and the system contaminated).

The differential pressure indicator incorporated on some filters provides one type of monitoring device. The indicators operate on the pressure drop across the filter element and are intended as fairly crude monitors. They do not differentiate between pressure drop due to increased flow, a dirty element, or high fluid viscosity at low temperatures.

Whatever type monitoring device is used, it is mandatory that it always be checked at the same operating conditions (pump speed, pressure, temperature, and flow). This is necessary to eliminate changes due to varying system pressure, pump speeds, and fluid temperatures.

5.5 Parallel Pump Installations

Those hydraulic systems that utilize parallel pumps where the pump case drains are connected to a common drain line should be designed to prevent fluid backflow between units as backflow may flush contaminants from the drain lines back into the pump case. This could cause a cross contamination between units or reintroduction of contaminants into the pump from which they originated. A backflow condition will occur when the pressure in the common drain line is higher than the case pressure of any one pump. This condition may be caused by a comparatively high case flow from one of several operating pumps or by a system in which one pump is not operating (i.e., the pump is on stand-by).

It is desirable that the common line be sufficiently large and unobstructed so that the pressure in the common line is always lower than the pressure in any of the individual case drain lines. In cases where the line length is not very large, it is desirable to run the individual case drain line directly to the reservoir.

The location of the junction of the drain lines close to a pump case port could also result in backflow into that unit. A check valve installed in each pump case drain line will prevent backflow into the units.

The installation of separate filters for each pump or the incorporation of one filter in the common drain line should be evaluated by the system designer. The installation of separate case drain line filters does not eliminate the need for the check valves. Separate filters will prevent cross-contamination between pumps but will not prevent backwashing of contaminants from the filter into the originating pump.

A filter in the common case drain line from multiple pumps will provide the necessary system protection. However, the failure of one of the pumps may cause high case pressure in all units if the filter element is heavily charged with contaminants. This condition could affect operation of the pumps, compromising the back-up feature of the system.

5.6 Miscellaneous

5.6.1 Pump Connections

Case drain ports are normally marked on the pump housing to prevent the improper assembly of the case drain line. Additional caution is advised to assure proper connection when connecting lines to pumps that have multiple seal drain ports in the same proximity as the case drain port.

5.6.2 Quick Disconnect Couplings

Case drain ports are usually located apart from the pump discharge and supply ports and may not be readily visible. Cautionary measures should be taken to assure that the drain line connection is not overlooked during pump installation. This is especially essential with sealing quick disconnects where the seal should unseat to complete the connection. Ensuing pump operation can result in failure due to an overpressure in the pump case and overheating of the pump.

5.6.3 Shut-off Valves

The use of shut-off valves in case drain lines should be avoided as they present the possibility of a blocked line and resulting pump failure. It is recommended that check valves be utilized if a means of isolating the pump from the system or preventing line drainage is required during pump removal.

6. MOUNTING PAD ACCESSORY DRIVE

6.1 General

There are several elements to consider when designing the drive and mounting pad for the hydraulic pump in the areas of:

- Vibration;
- drive alignment;
- pad environment;
- splines;
- torque;
- speed.

Frequently, the drive pad for the hydraulic pump is established by the engine/gearbox manufacturer prior to the selection/sizing of the hydraulic pump by the airframe manufacturer. Every effort should be made to avoid the selection of an under-designed drive pad, especially in the aforementioned areas, for the hydraulic pump.

Appendix C provides a list of drive pad standards.

6.2 Vibration

Vibration tests for modern aircraft are often performed in accordance with MIL-STD-810 Method 514 (military applications) or RTCA/DO-160 Section 8 (commercial applications) requirements. The accessory drive pad for the pump should be reviewed for stress to take the required pump weight (wet) at the extreme limits for G-loads induced during vibration/operation and for stiffness of the mounting pad.

The stiffness of the accessory drive can become a factor when cumulative G-loads are induced on the pump parts/components that are at a distance from the mounting pad. The conditions that may cause adverse G-loads should be reviewed in the actual installation as well as during the individual pump vibration test.

6.3 Pad Temperature

Extreme heat conducted to the pump from the accessory drive pad can have an adverse effect on the seals in the area of the pump drive shaft. The amount of heat transfer from the accessory drive to the pump should not cause the temperature in the local area of the pump drive to exceed the rated temperature for the elastomers used in the seals or for the system hydraulic fluid.

Elastomers lose their sealing capability when excessive heat forces out the plasticizers, resulting in leakage past the seals.

Fluid breakdown occurs when the rated temperature of the fluid is exceeded. Sometimes, hard deposits are formed which cause excessive wear of the mechanical shaft seal parts and results in external leakage.

Excessively hot drive pads can cause adverse wear of oil or grease lubricated drive splines by causing the breakdown of the lubricant.

6.3.1 Soak Back

When the pump is operating, the internal leakage assures there is a constant flow of fluid through the pump case. This transports the heat developed by the pump and that absorbed from the environment and the pad. When the engine driving it is shut down this cooling flow stops.

There may be residual heat in the engine structure that can soak back into the pump. This can cause temperatures in the pump to be higher than those in normal operation. Such effects can cause damage to elastomeric seals and fluid degradation. Some provision should be made to insulate the pump if these conditions can occur.

6.4 Pad Environment

Normally, the shaft seal cavity in the drive pad area is vented to the atmosphere. Precautions are required to avoid the entrance of contaminants, fuels, fumes, or other chemicals that can affect the shaft seal integrity or cause abnormal wear of the drive splines.

The design of the shaft seal cavity should also include provisions to prevent the ingress into the engine gearbox of any hydraulic fluid leakage from the pump shaft seal. This includes consideration of the potential for blockage of the seal cavity drain lines by deteriorated or drying hydraulic fluid.

6.5 Drive Alignment

All hydraulic pumps in aircraft installations are spline driven with some having a separate coupling shaft between the pump and accessory drive and some with the pump shaft being driven directly by the accessory drive.

The following are to be considered when determining radial and angular misalignment between the pump and accessory drive pads:

a. the tolerance stack up including eccentricity and squareness of both drive pads;

b. the deflection of the accessory drive pad due to extreme G-loads and/or vibration on the cantilevered installation of the hydraulic pump;

c. the total angular deflection of the units.

The amount of eccentricity and angular misalignment that can be tolerated is dependent upon the clearance between the spline teeth and the angular movement of the self-aligning bearing, if used, in the accessory drive.

- To Excessive angularity in the splines results in:
- spline wear;
- fretting of the contact surfaces;
- axial motion of the drive shaft(s);
- shaft bearing wear;
- shaft seal leakage;
- A vibration is introduced to the shaft under misaligned conditions that can accelerate the wear of the splines and set up resonances that may cause other parts to fail.

An area of concern in the pump is the shaft seal because eccentric rotation of the pump shaft will be reflected to the shaft seal and can be manifested as pump shaft seal leakage.

6.6 Drive Splines

In order to reduce spline wear, application of grease to the spline is most commonly recommended and utilized by maintenance personnel. However, greases are often life limited and harden in time and trap moisture causing corrosion of the metal parts.

Dry lubricant coatings on splines have been used successfully as an alternative to grease.

Wet splines are recommended as a solution to reduce spline wear. When properly sealed, the wet spline approach can provide a constant lubrication and a self cleaning effect on the spline surfaces. It is noted that the required seals are additional failure modes to be considered. Since wet splines are lubricated by the engine or gearbox lube oil, the interface for the drive head and the pump should be carefully specified. A seal may be installed on the drive shaft to retain the gearbox lubrication fluid. However, care should be taken to ensure that the seal material is compatible with both the gearbox lubrication and aircraft hydraulic system fluids.

Nonmetallic spline adapters are noted in AS595 and AS19692 as a preferred method to extend spline life and reduce repair costs in hydraulic pump installations.

Older applications experiencing life limiting problems due to spline wear can be converted to use the nonmetallic adapters. It is noted, however, the shear torque of the drive shaft may be affected either by the nonmetallic adapter or the reduced cross section of the shaft that is required to accommodate the adapter in older applications and the expected ambient temperature.

Another consideration in the design of the shear section is fatigue due to the normally occurring torque ripple. For this reason, the shear section should be designed for a minimum shear torque of no less than four times the design operating torque of the pump.

6.7 Torque

Positive displacement piston pumps have start-up torques higher than running torques. This can have an effect on shaft shear sections and should be considered in the design of the driving components such as electric motors, ram air turbines and air turbine drives.

6.8 Speed

The drive pad speed for the hydraulic pump should be coordinated with the pump manufacturer to establish the optimum speed/displacement for the application flow requirements.

Figures 4A to 4D are presented as a guide; it is important to note these nomographs may shift significantly as a function of other parameters such as:

- type of fluid;
- filtration;
- operating temperature;
- pump inlet pressure conditions;
- desired pump life and reliability.

6.8.1 Low Speed

Operation for extended periods or continuously at very low speeds (<500 rpm) can also be damaging to typical aerospace hydraulic pumps.

There are many running surfaces and rolling element bearings in typical pumps that depend on hydrodynamic films for lubrication and to support the applied loads. These films are often not fully developed at very low speeds. Continuous or long periods of operation at low speeds and high pressure may result in rapid wear and premature failure unless special design features are incorporated.

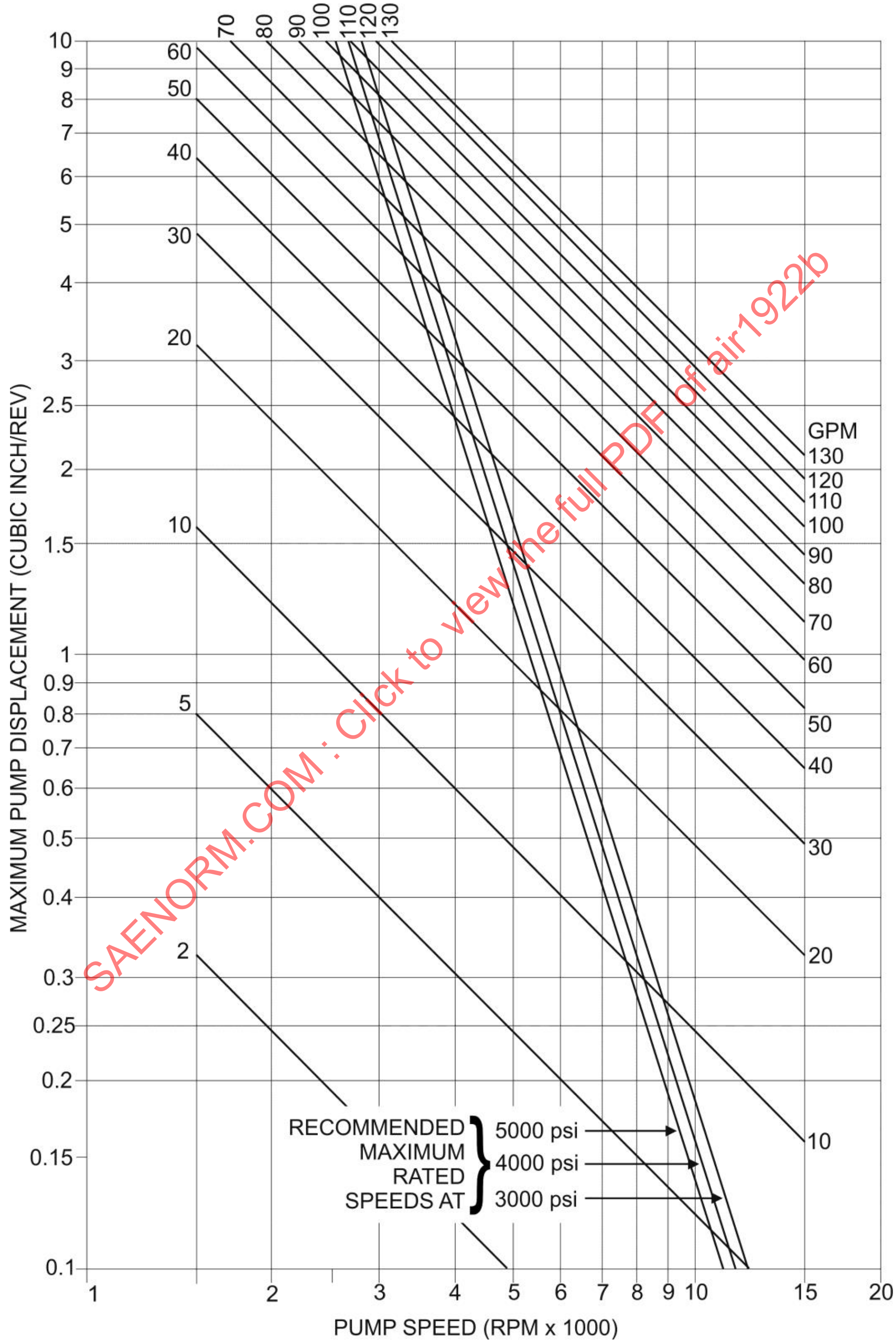

Figure 4A - Maximum pump displacement versus pump speed (English units) – Commercial aircraft applications

NOTE: Based on charts in AS595.

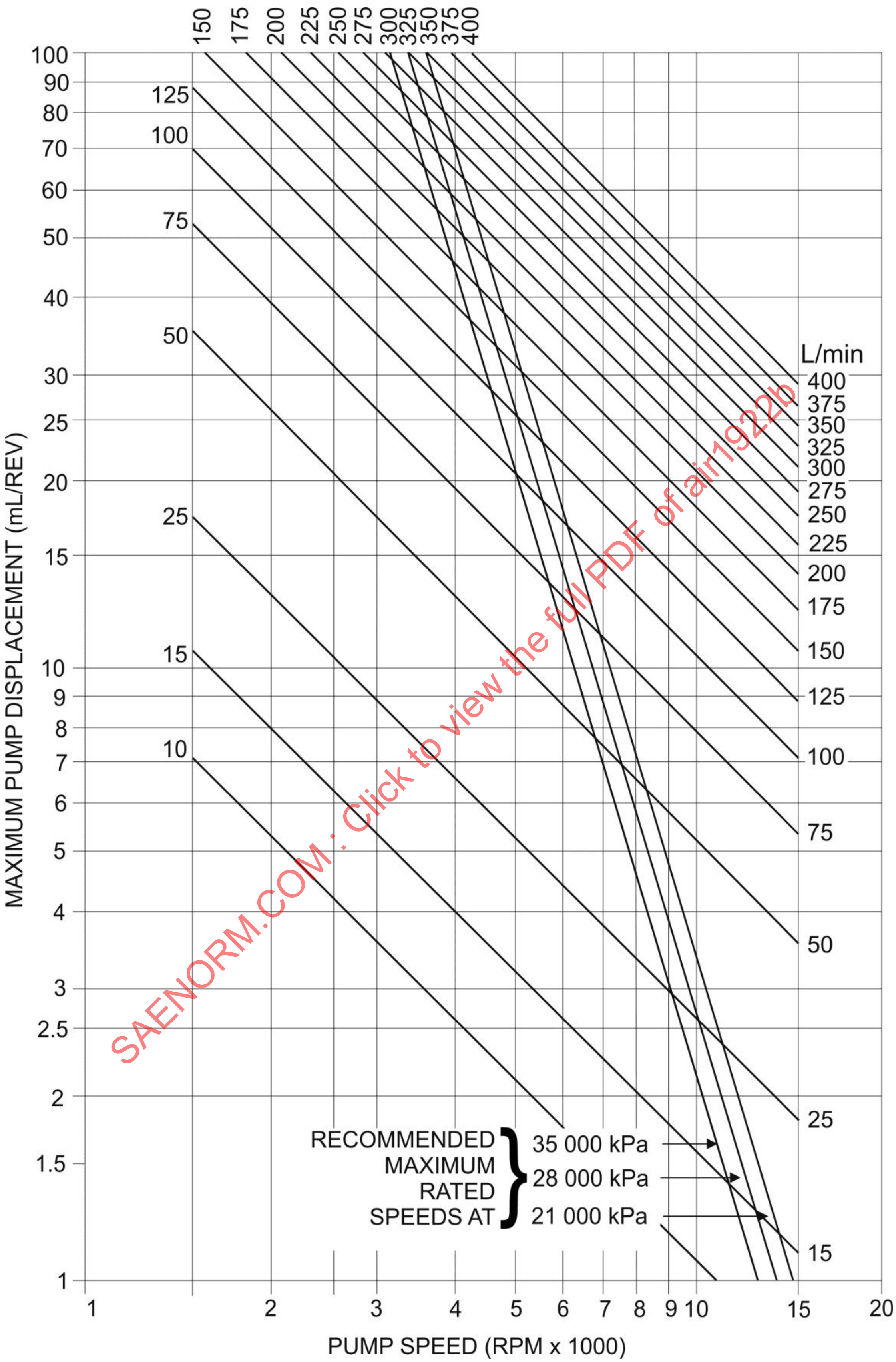

Figure 4B - Maximum pump displacement versus pump speed (metric units) – Commercial aircraft applications

NOTE: Based on charts in AS595.

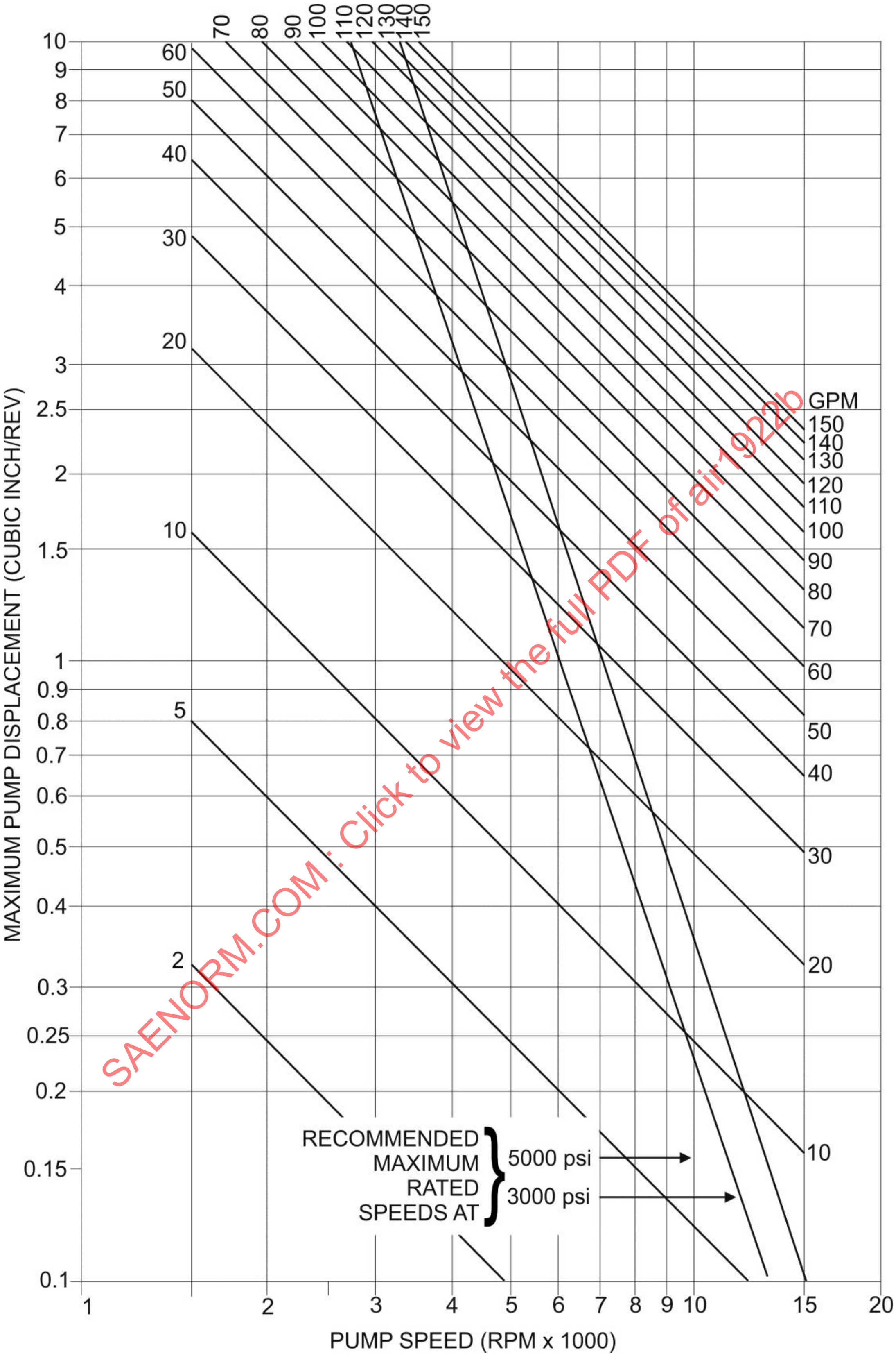

Figure 4C - Maximum pump displacement versus pump speed (imperial units) – Military aircraft applications

NOTE: Based on charts in AS19692.

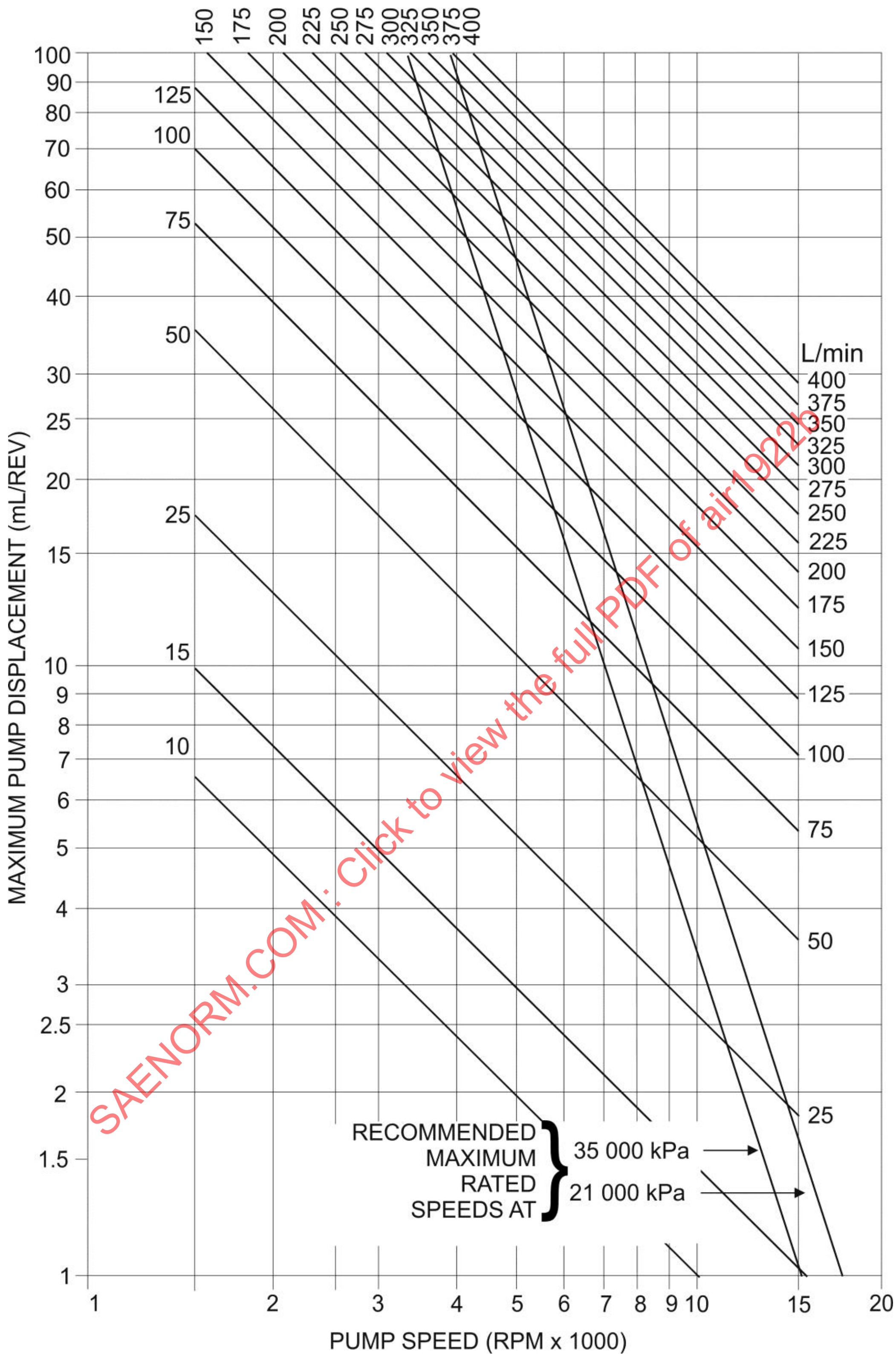

Figure 4D - Maximum pump displacement versus pump speed (metric units) – Military aircraft applications

NOTE: Based on charts in AS19692.

6.9 Miscellaneous

Quick attach detach (QAD) pads are being used extensively on drive mounted components for easier installation and removal. Experience has demonstrated QAD pads to be trouble free in service when designed to sustain the installation loads and vibration environment.

7. FLUID PROPERTIES AND CLEANLINESS

7.1 Fluid Contamination

Control of fluid contamination is necessary and a key factor in any effort to maintain and/or improve pump reliability.

The primary types of contamination which require attention in most aerospace hydraulic systems are:

- solid particles;
- water;
- free and dissolved air;
- other gases;
- chemicals such as chlorine solvents and acids.

Contamination can enter into a hydraulic system from external sources as well as being generated internally. Contamination of a system from external causes can occur when a component is replaced, new fluid is being added, ground support equipment is attached or personnel awareness is lacking.

Internally, contamination can be generated by components wear or failure, residual particles from fabrication of components, system operation at temperatures higher than rated for components and fluid, chemical reaction of fluid with other components or particulates, and formation of fluid deposits and localized hot spots that cause component wear or fluid breakdown.

Contamination control of a hydraulic system is a complex undertaking. Monitoring of the various indicators for component and system fluid condition is required. Solid particle size and amount should be controlled in a hydraulic system in order to sustain or extend pump life.

Provisions for filtering particles from the fluid should be considered in the design phase for ease of maintenance and high hydraulic system reliability.

Pump life can be extended when operating in a clean system as proven by results of pump wear tests conducted by the Franklin Institute and the U.S. Navy Air Development Center.

7.1.1 Fluid Contaminants

AS4059 is used as the standard to establish the requirements for the allowable solid particles in most commercial and military aircraft hydraulic systems.

Determining the cleanliness level for a system requires removing fluid samples from the system through sampling valves that are often designed into the system. ARP5376 provides details where the sampling valves can be located in a hydraulic system. The fluid samples can be used in various ways to evaluate the system condition and the amount of various contaminants in the fluid. This data coupled with information obtained from evaluating the case drain filter can be helpful when making a disposition for the pump.

The other contaminants in the hydraulic system that adversely affect pump life (air, water, chlorine, etc.) require monitoring by laboratory methods (see 7.3).

7.2 Filtration

Hydraulic system filtration should be carefully matched to the system components, to the operating environment and to the cleanliness level needed to obtain the desired and most cost-effective pump and system performance.

In many legacy aerospace hydraulic system applications, 5 and 15 µm absolute filters per MIL-F-8815 are used.

For new hydraulic system designs, filters whose filtration efficiency is met with cyclic flow rate conditions per ARP4205 or steady state flow rate conditions per ISO16889 at filtration ratios ≥200 should be specified.

The filters selected should not bypass fluid or energize the differential pressure indicator during cold starts that momentarily result in a higher pressure drop across the filter element. Indicators should have properly set thermal lockout devices that prevent actuation at low temperatures. Differential pressure devices that also read temperature with continuous (analog) signal outputs are available, are more accurate, allow for continuous element monitoring, and eliminate early indication due to low temperature effects on element pressure drop. There are also many other factors to be considered when choosing the efficiency rating for filters.

Filters (cleanable or non-cleanable) capable of removing 2 to 5 µm particles are recommended to be used at the system fill point(s) to ensure clean fluid is introduced into the system during servicing, etc.

7.2.1 System Filters

Filters are normally located in the following locations:

- High pressure - Either per hydraulic pump or in the central hydraulic system to ensure that clean fluid is supplied to the hydraulic services.
- Low pressure - To clean up the hydraulic fluid prior to entering the reservoir.
- Case drain - To prevent the particles generated by pump wear or failure from entering the system.

NOTES:

1. Ideally the high pressure and case drain filters should be installed as close as possible to the pump.
2. If filters are installed in the pump inlet line, other factors such as inlet line pressure drop, the impedance to fluid acceleration and aeration of the fluid should be investigated to ensure that there will not be any cavitation damage to the pump.

7.2.1.1 Case Drain Filtration

The installation of a case filter is recommended in the case drain line even though there may be a return line filter downstream. The case drain line filter may also serve as an inspection tool for monitoring the condition of the pump by observing the size (>25 µm) and amount of contaminant (especially metallic) collected on the filter and in the filter bowl. Filters are available with upstream pullout diagnostic layers to assist in identifying wear debris.

When the pump case drain line is connected to the system return line, a filter and check valve close to the case drain port will prevent the backwash of return line particles from entering the pump case.

The case drain filter size is often selected on the basis of clean element pressure drop for the expected case drain flow rate. However, in long flights or long duration of usage, a substantial volume of fluid can pass through the pump case drain port. This is to be considered when selecting the filter dirt holding capacity, desired efficiency and pressure drop. Excessive pressure drop in the case drain line may reduce pump life or damage the pump.

The effect of possibly bypassing contaminated fluid to the system may be considered. An external indicator to signal an imminent bypass condition should be included with any bypass valve.

A reasonable cleaning/replacement schedule should be established to keep the case drain filter elements in a relatively clean condition and to monitor pump wear.

7.3 Fluid Monitoring

The condition of the system hydraulic fluid has a direct impact on the hydraulic pump life and performance. Because of the influence on pump life, monitoring of the system fluid properties on a regular basis is recommended. (See Table 1 for typical in-service limits (from AIR810 and AIR5277)). The amounts of contaminants other than solid particles to be monitored and controlled are:

- free water;
- free chlorine;
- undissolved gases and air;
- acidity;
- other alien chemicals dependent on the hydraulic fluid or operational environment.

Table 1 - Suggested in-service fluid limits

Analysis	MIL-PRF-83282	MIL-PRF-87257	AS1241
Appearance	Red (clear)	Red (clear)	Purple (clear)
Moisture (%)			0.8 (max)
Acid Number (mg KOH/gm)	0.30 (max)	0.30 (max)	1.5 (max)
Viscosity 100 °F [38 °C] (Centistokes)	12 (min)	10 (min)	6.0 to 12.5
Particulate Contamination (AS4059 Class)	9	9	9
Chlorine (ppm)	200 (max)	200 (max)	200 (max)
Total Water (ppm)	350 (max)	150 (max)	

Fluid breakdown is another characteristic that will indicate that the fluid has deteriorated. This leads to loss of lubricity, large drop in viscosity, high acid number, or formation of fluid deposits such as gels or hard salt particles. The equipment for detecting fluid deterioration have improved and have resulted in a reduction in time required to perform the necessary tests and come with a variety of inspection methods to suit particular needs.

Monitoring the system fluid properties is primarily a preventive maintenance action. Some of the reasons for monitoring the fluid properties in regard to the effect on pump life are as follows:

a. Entrained (free) air in the fluid is inducted into the pump’s inlet connection, resulting in cavitation erosion on the pump parts. The compression of the air bubbles in the subsequent discharge stroke also results in higher pressure ripple in the outlet circuit

b. Free water and acid in the fluid are detrimental to many pump parts, causing corrosion and wear as well as loss of lubricity, lower viscosity, and high electrical conductivity

c. Chlorine in the hydraulic fluid acts as a reactant to form acids which causes corrosion, electrochemical erosion, and deposits