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Aerospace Hydraulic Pump Controls

RATIONALE

There is a need for a document that provides aerospace hydraulic system and pump designers the various options for controlling the operation of hydraulic pumps, thereby helping in the selection of the most suitable pump for their systems. This document satisfies this need.

TABLE OF CONTENTS

1. SCOPE

This Aerospace Information Report presents an overview of the application and control of fixed and variable displacement pumps with the emphasis on the controls most commonly used on variable displacement pumps. It describes various options to control the operation of hydraulic pumps in terms of controlling the pump output pressure and/or flow and assisting in the selection of the pump.

1.1 Purpose

The information contained herein is intended to provide application information to assist system integrators in selecting the most appropriate pump controls for a given application. It can also be used as an introduction to pump designers.

1.2 Field Of Application

The controls described are applicable to hydraulic pumps installed on commercial and military aerospace applications.

Additional details relating to design and performance characteristics of pumps in various aerospace applications are provided in the following documents:

a. AS595 provides additional information on civilian aircraft pumps

b. ISO8278 provides additional information on civilian and military aircraft pumps

c. AS19692 and ARP560 provide additional information oriented toward general military and missile pumps, respectively

d. ISO22089 and ARP1280 provide additional information for controlling power transfer units

e. AS5994 provides additional information on electric motor driven pumps

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.

2.1.1 SAE Publications

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

AS595 Pump, Hydraulic, Civil Type Aircraft and Variable Delivery

ARP560 Pumps, Missile Hydraulic

ARP1280 Application guide for fixed displacement Aircraft Power Transfer Units.

AS5994 Pump, Hydraulic, Electric-Motor-Driven, Variable Delivery

AS19692 Pumps, Hydraulic, Variable Delivery, General Specification for

2.1.2 ISO Publications

Available from American National Standards Institute, 25 West 43rd Street, New York, NY 10036-8002, Tel: 212-642-4900, www.ansi.org.

ISO8278 Hydraulic pressure compensated, variable delivery pumps-General requirements

3. OVERVIEW OF PUMP CONTROLS

3.1 Definition

Hydraulic pump controls are devices used to dynamically control the displacement of variable displacement pumps in a predictable and repeatable way. Typically, the intent is to control parameters such as pressure, flow, input starting or running torque and any combination thereof, since in some cases the system requirements may be fulfilled by combining two or more of these control schemes.

3.1.1 Aerospace Type Controls

These controls differ from Industrial and Mobile controls due to the particularities of aerospace applications that demand fast response and light weight. They are usually integrated within the pump body to increase reliability and reduce weight and external leakage. Aerospace hydraulic systems differ from industrial and mobile systems in that they are more compact, have less volume and utilize smaller line sizes thereby experiencing greater pressure drops and surges, and have smaller reservoir volumes making heat rejection more challenging. These factors make aerospace type controls somewhat different in implementation from their industrial and mobile counterparts, although in principle they are similar.

3.1.2 Classifications

Generally, aerospace pump controls can be classified into the following categories, although there are many overlapping combinations possible to fit more demanding applications:

- Pressure limiting controls
- Torque Limiting, Unloading and Blocking controls
- Thermal controls
- Adaptive controls
- Multi-function controls

The control types listed above include both hydromechanical and electrical technology, and also touch on industrial controls that may, in some instances, find application in an aerospace environment.

4. DESCRIPTION OF CONTROLS

4.1 Fixed Versus Variable Displacement Pumps

4.1.1 Introduction

Fixed displacement pumps provide flow that is directly proportional to its speed of rotation, the ratio being the displacement of the pump expressed as a volume per revolution. Variable displacement pumps are capable of varying their displacements in response to some control mechanism. Applying various control schemes to this variable displacement mechanism to achieve different control characteristics is the subject of this report. A brief introduction to fixed displacement pumps is offered to illustrate some factors that may make a fixed displacement pump more suited to a particular application as opposed to a variable displacement model.

4.1.2 Concept Description

4.1.2.1 Fixed Displacement Pump

For a given input speed, a fixed displacement pump delivers a fixed flow, as depicted by the Pump Flow line in Figure 1. As long as the system loads can use this amount of pump flow, then the operating point lies on this line, at a point corresponding to the pressure demanded by the loads.

In many instances, however, the loads attached to the pump are controlled by valving that limit their flow demand to a value less than the pump flow, as depicted by the Load Flow line. Point 1 on the chart corresponds to such a condition, where the Required Pressure includes the pressure demanded by the load plus the total pressure drop in the pressure and return lines between the pump to the load control valve. Because the load is not demanding all of the available pump flow, the pump is constrained to operate at point 2, passing the Relief Valve Flow across the relief valve at the pressure determined by the Relief Valve Setting. Consequently, area B represents the power loss in the load control valve to maintain the required flow and pressure in the load and area C represents the power loss in the relief valve to bypass the flow not needed by the load but supplied by the pump. The sum of areas B and C represent the total power loss in the system that is converted to heat, as depicted by the shaded area in Figure 1.

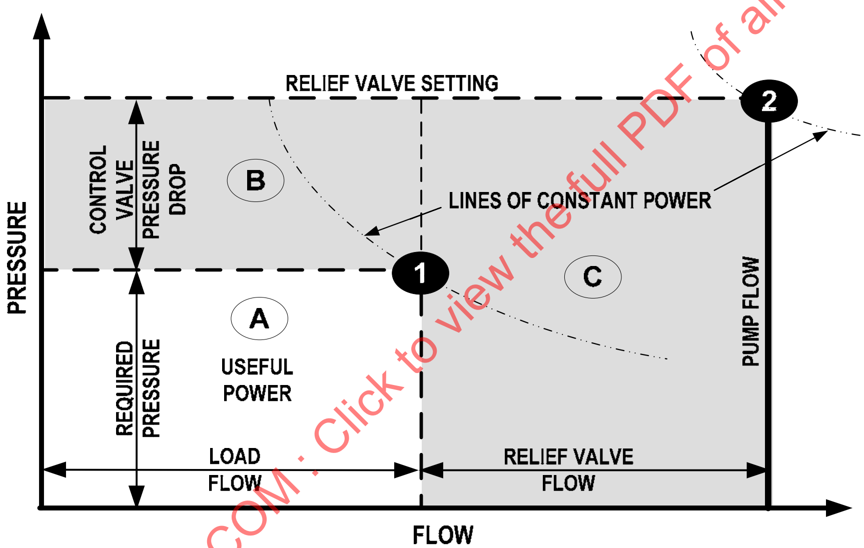

FIGURE 1 - FIXED DISPLACEMENT PUMP CHARACTERISTICS BELOW RATED FLOW

Fixed displacement pumps were used years before variable pumps were invented and were used successfully in specific applications. It is still possible to take advantage of the simple design of fixed displacement pumps by using them in applications where they can operate on the Pump Flow line below point 2, thereby not wasting power as described above.

In such a case, the operating point 1 is located on the Pump Flow line as depicted in Figure 2 and the required pressure in the load is less than the relief setting, depicted as point 2. Note that as long as the operating point 1 is below point 2 on the Pump Flow line, all of the pump flow is passing through the load at the required load pressure and none of the flow is passing through the relief valve; consequently the power loss described in Figure 1 is not present.

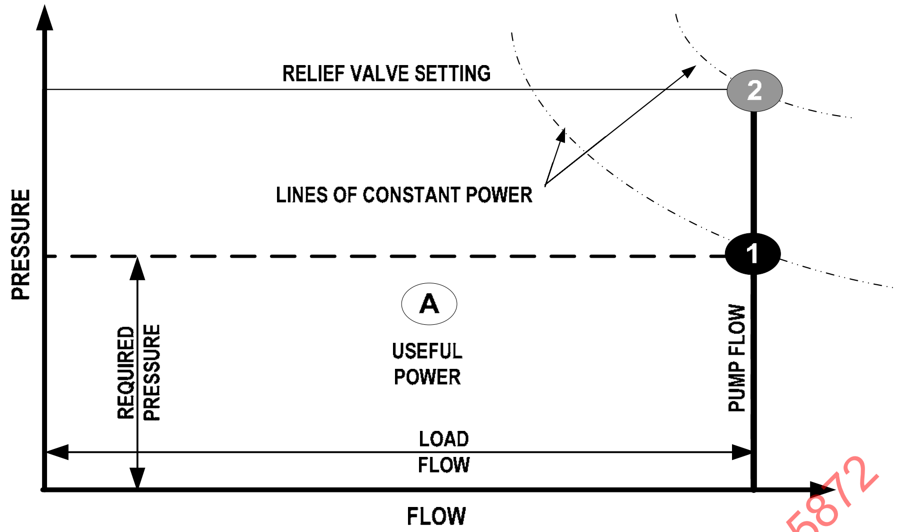

FIGURE 2 - FIXED DISPLACEMENT PUMP CHARACTERISTICS BELOW RELIEF PRESSRE

The scenario described above is illustrated by the following application examples:

a. For systems that only require moving an actuator intermittently at maximum speed, the pump can be coupled to an electric motor which can be started and stopped as required. In this case, the operating point is on the Pump Flow line below point 2, and may jump to point 2 momentarily if the load actuator bottoms, at which time an external pressure signal stops the motor.

b. For systems where the function of the pump is to charge an accumulator and hence the accumulator provides flow as needed to the system, an unloading valve (also known as a “cut-in cut-out” valve) can be provided downstream of the pump. This valve allows the pump to fill the accumulator at the pump’s full flow until the preset pressure is reached, at which time the valve shifts to a bypass position where the full pump flow is directed at low pressure back to the reservoir. A check valve integral with the unloading valve maintains pressure in the accumulator and the system downstream. As the system consumes the high pressure fluid from the accumulator, the low pressure preset is reached and the unloading valve shifts back to the accumulator charging position and the accumulator charging cycle repeats.

   In this type of application, the operating point is near the bottom of the Pump Flow line (near zero pressure) when the valve is bypassing, and higher up the line when the accumulator is charging, but is designed to remain below point 2 at all times.

   The unloading valve can be actuated either hydraulically by pilot pressure from the system or electrically by a signal from a pressure switch. This implementation is common in systems where precise regulation of system pressure is not needed, but is allowed to vary from maximum to a lower preset minimum level.

c. For electric motor driven pumps where an electric motor with sufficient dynamic bandwidth is used to vary pump speed using digital controls so as to achieve the desired pressure / flow characteristic. These types of electric motorpumps are used in the control of Electro Hydrostatic Actuators (EHA) and in system back up pump applications (for example, A380 braking and steering systems). The variable speed motor controls used in these applications are not covered in this report.

   In this type of application, the operating point moves up and down the Pump Flow line dynamically, but is designed to always remain below point 2 so as not to allow flow through the relief valve.

Fixed displacement pumps are very compact, have in fact no internal control, since all controls for the system are done outside the pump. They can achieve very high power density, power to weight ratio. They can also be integrated at higher sub-assembly level by the next level integrator; they can be located at the point of usage when driven by electric motors at variable speed.

4.1.2.2 Variable Displacement Pump

The advantage of the variable displacement pump is that it can operate at any flow, from zero to its maximum capacity. Assuming the conditions from the fixed pump example of Figure 1, the operating point 1 is again shown in Figure 3. Since the variable pump can match its output flow to the Load Flow, area C is not a power loss as it was in the fixed pump example. The only power loss in a variable displacement pump is area B, resulting from the pressure drop through the control valve. In applications involving a central hydraulic system with multiple loads operating at varying flows simultaneously, the variable displacement pump is always able to match the combined load flow demand.

Although a relief valve is also used in a variable displacement pump circuit, it is only required as a safety feature and is not normally passing fluid. The absence of flow through the relief valve reduces the heat generated in a system with a variable pump.

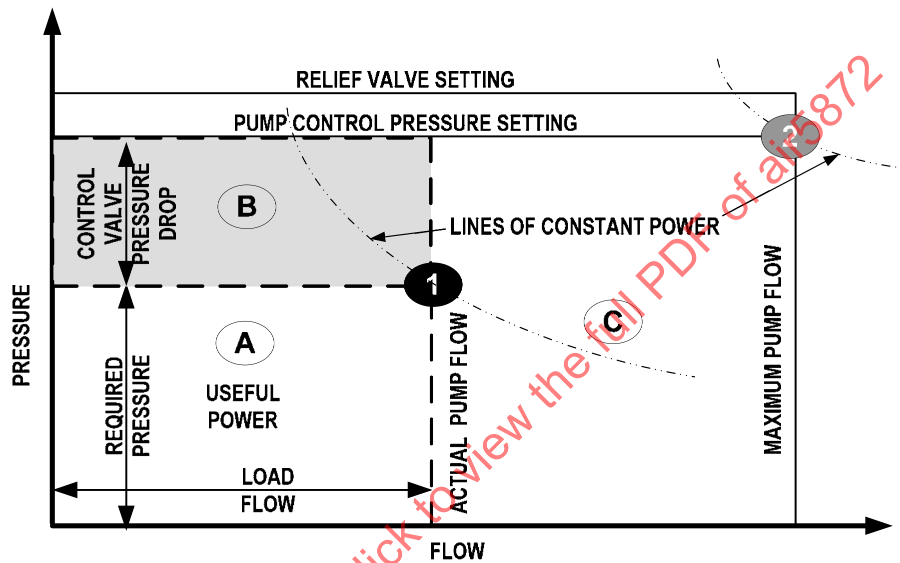

FIGURE 3 - CHARACTERISTICS OF A VARIABLE DISPLACEMENT PUMP

The control schemes described in this document are limited to those that are utilized with variable displacement pumps and illustrate ways that the pump design can be optimized and tailored to the system for which it is intended.

4.1.3 Examples and Limitations of Usage

4.1.3.1 Variable Displacement Pumps

Variable displacement pumps are typically used in central hydraulic systems characterized by multiple consumers where pump output is controlled by means other than the pump's speed. Examples of the use of variable displacement pumps include:

a. Engine driven pumps, attached to engine gearboxes or to airframe mounted accessory drives (AMAD) that are constrained to run at a speed proportional to the engine speed and provide power at near constant pressure matching the demand of the consumers

b. Electric motorpumps where the motor speed is fixed, as a function of the electric motor power supply and the inherent design of the motor. These pumps are generally used to power the same consumers are engine driven pumps for operating on the ground (without the need to run the engines) or during taxi and flight to augment the power from engine driven pumps.

c Air driven pumps, either driven by air turbines powered by the aircraft pneumatic system or by an air turbine deployed in the airstream (Ram Air Turbine), both used types for emergency power generation.

d. Systems where near-constant pressure is required throughout the flow range and digital control is not desired

4.1.3.2 Fixed Displacement Pumps

Examples where fixed displacement pumps can be used include:

a. EHA systems where the power can be in the neighborhood of 20 kW input to the pump with speeds demonstrated in excess of 20 000 rpm.

b. For starters on small jet engines as found on helicopters, the power at hand may be hydraulic power, which is usually stored in hydraulic accumulators. To recharge the accumulator, either hand pumps for very small jet engines, or electrically driven pumps, (usually DC) are used. These pumps are usually of the fixed displacement type for small systems. The only control being a pressure switch and a check valve located at the exit of the pump.

c. For cargo doors and landing gears, which operate infrequently for short durations, the same type of control is weight effective with either a DC or AC motor driving a fixed displacement pump, a micro switch is used to stop the pump at either end of travel. The rest of the circuit is comprised of cylinder(s) and a relief valve.

d. As pumping elements of Power Transfer Units (PTU's) where the pump speed is dynamically controlled by the hydraulic motor side of the PTU in response to system pressures.

4.2 Pressure Controls for Variable Displacement Pumps

4.2.1 Overview Of Pressure Controls

Pressure controls are dynamic elements installed in variable displacement pumps to control displacement by maintaining a prescribed outlet pressure. These controls are characterized by high bandwidth (natural frequency f_n > 300 Hz) and low damping (critical damping ratio $\zeta < 0.7$) in order that they can maintain tight pressure control. This may lead to stability issues in the application of these pumps to aircraft hydraulic systems. It is of utmost importance that system stability be analyzed carefully by considering pump piston ripple, transient start / stop cycles and cavitation effects during rapid increases in flow demand.

Because of the dynamic nature of pressure controls, the interdependence of the pump controls and system design have significant impact upon the fatigue life of the system components. High impulse pressure cycles are a consequence of pressure transients due to flow transients, while pressure ripple generated by the pump and transmitted throughout the system give rise to high cycle but lower amplitude impulse. Typically, pressure transients are 35% of rated pressure, while pump generated ripple is ±5% of rated. The pump pressure ripple can be as low as ±1% in some well-designed systems and as high as ±10% as allowed in older systems.

A useful parameter for analyzing transient effects is system stiffness, or impedance, which is a function of system volume, fluid bulk modulus and compliance of the system tubing and hoses. This is based on an equivalent bulk modulus, B_e, which incorporates the effects of the compliance of the tubing and hoses containing the hydraulic fluid.

According to AS19692, the standard system impedance for testing pump control response time is to be established such that the rate of pressure rise upon a step input of rated flow to zero flow is 50,000 psi/s (344 738 kPa/s).

The relationship between the rate of pressure rise resulting from a step change in flow and bulk modulus is derived directly from the definition of bulk modulus as depicted in Equation 1.

$$B_e = \frac{\dot{P}}{\dot{V}/V} \quad \text{(Eq. 1)}$$

where:

B_e = the equivalent bulk modulus of the system, taking into account the effect of the compliance of tubing, hoses and undissolved air and gas , in (force / area) units

$\dot{P}$ = rate of pressure rise in (force / area / time) units

V = system volume in (volume) units

$\dot{V}$ = Q = volumetric flow rate in (volume / time) units

The effect of air or gas in the high pressure system is only a factor if it is not in solution. It is therefore necessary to bleed all undissolved air in order to achieve the design stiffness.

4.2.2 Flat Cut-Off Pressure Compensator Control

4.2.2.1 Introduction

This control is the most commonly used for variable displacement control of pumps. It provides a nearly constant pressure at the pump output at all flows lower than the maximum flow capacity of the pump while keeping a very simple approach with a minimum of parts, thus yielding a very high reliability.

4.2.2.2 Typical Performance

The flat cut-off control is designed to provide near-constant pressure from zero flow to the maximum pump capacity as depicted in Figure 4. The reduction in flow as pressure is increased from zero to maximum is a consequence of the pump's internal leakage as pressure increases, which also includes the quiescent flow through the compensator valve. This results in a loss of about 5% of rated flow, although this is dependent upon the frame size of the pump in relation to the maximum displacement.

From the maximum flow - pressure operating point to zero flow, the compensator is designed to reduce displacement while maintaining near constant pressure. The pressure typically increases 3% or more from rated flow to zero, which is mainly a consequence of the displacement control plate spring rate.

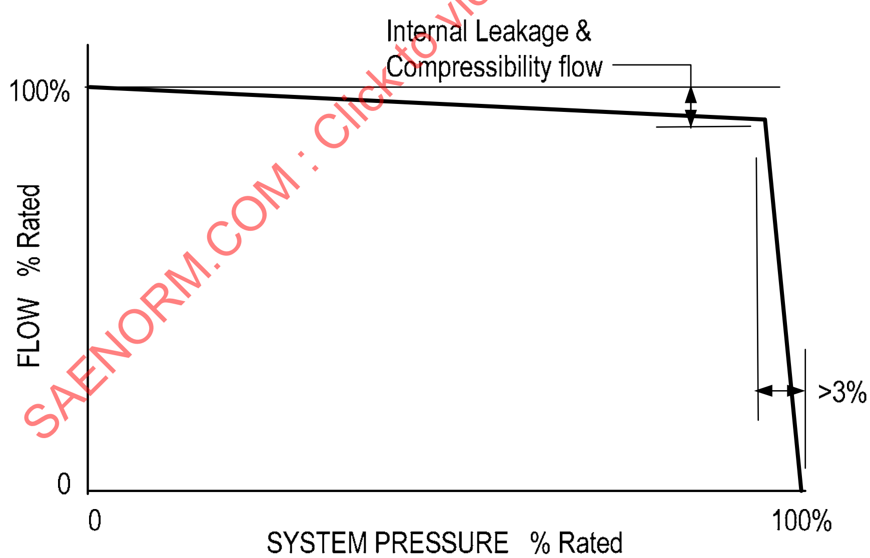

FIGURE 4 - TYPICAL PERFORMANCE CHARACTERISTIC – FLAT CUT-OFF

4.2.2.3 Concept Description

Referring to Figure 5, the Displacement Control Plate angle is directly related to the displacement of the pump and is usually positioned by a Displacement Control Actuator piston, opposed by a spring. The spring (in the absence of any other moment created by the geometry and pressure) usually returns the control plate to a position of maximum pump displacement. A smaller, opposing piston may also be part of this design and may assist or replace the spring in order to bias the control plate to a full displacement position.

A step by step description of the general function of the internal mechanism of the variable delivery pump will give a better understanding of what happens in the control circuit for a flat cut-off pressure compensated control.

As a starting point, assume that the pump is driving a load at full speed, and consequently the pump displacement control plate is at maximum angle as shown in the graphic. As the load, or resistance to flow increases (for example, due to increasing aerodynamic load on a control surface), the pressure in the Outlet port rises, and flow remains maximum until the pressure reaches the setting of the Compensator Valve Spring. Further load increase causes the system pressure to exceed the setting of the spring, and the Compensator Valve Spool is moved toward the spring, allowing flow from the outlet line to be metered to the Displacement Control Actuator. The actuator moves the Displacement Control Plate to a smaller angle, or pump displacement, resulting in reduced outlet flow. The displacement continues to decrease until the outlet pressure is maintained at the compensator setting.

In the case where the pump is at low flow and a load valve opens initiating a flow demand, the outlet pressure momentarily decreases, and the compensator valve spool is displaced in the opposite direction. This causes the compensator spool to depressurize the control actuator, causing it to retract, as the opposing control plate spring presses the control plate to increasing angle. The control plate assumes a new angle when the pump flow is just sufficient to again satisfy the flow required by the system at the compensator preset pressure. The control plate is in its new position and the compensator valve is again centered.

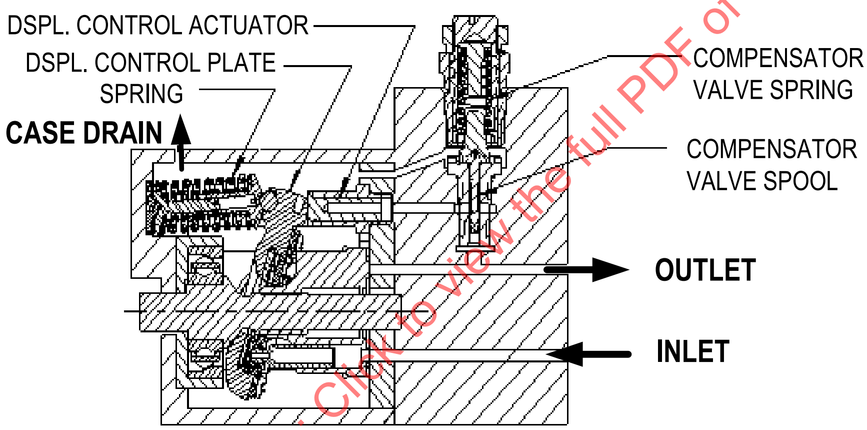

FIGURE 5 - TYPICAL CROSS-SECTION VIEW – FLAT CUT-OFF

If the pump outlet flow is completely blocked, the control plate will be moved to a position near perpendicular to the shaft, with only enough displacement to provide for the internal pump leakage flow. The system pressure will be maintained as the compensator valve is nearly centered and the pump will remain at this condition until system demands some flow.

4.2.2.4 Schematic

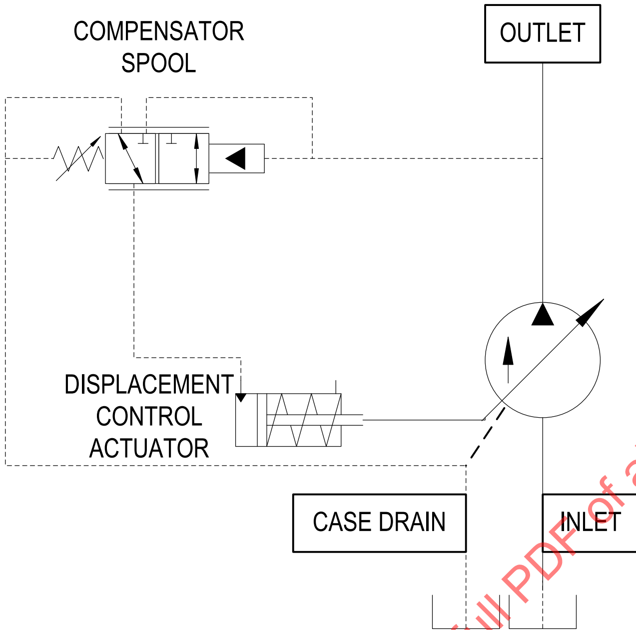

FIGURE 6 - SCHEMATIC – FLAT CUT-OFF

4.2.2.5 Limitation Of Usage

Since the pump is an integral part of the system as a control element, it is important to consider the stability of the system as a whole under all conditions of flows, pressures, speeds and temperatures.

Under special conditions, or specific systems, the stability of the pump control can be enhanced by increasing the mass of the compensator spool valve, or adding additional viscous damping within the compensator to effectively slow down the response of the spool.

It should be noted that any modification of the response of the compensator spool valve will affect the pump transient response.

It is to be noted that on some applications, it may be desirable to slow down the response of the pump in the direction of increasing flow only, when poor inlet conditions exists. The inlet line may either be too long or of insufficient size, and thus prevents the active pumping mechanism to fill properly during a increasing system flow demand resulting in transient pump cavitation, thus affecting the life of the pump.

Multiple variable delivery pumps connected in parallel may in some systems develop cross talk. One way to prevent cross-talk is to set the compensators about 2-3% apart, so that one pump begins compensating before the next. Alternately, if the same part number pump is desired in all the locations, check valves in the outlet circuit can be set to different cracking pressures to similarly separate the pump control dynamics.

4.2.3 Soft Pressure Cutoff Control

4.2.3.1 Introduction

Soft cut-off control is similar to flat cut-off described earlier, except that the cut-off starts at a lower pressure and the flow-pressure curve from full flow to cut-off has a reduced slope. Since most systems usually do not require high pressure and high flow simultaneously, the point of maximum pump power demand is less than a similar flat cut-off pump. Prime movers of hydraulic pumps limited in torque or power can be fitted with a soft cutoff control to reduce pump power demand.

The soft pressure cut-off control provides substantial advantages such as weight and size of the prime mover. If the prime mover is an electric motor, the installed electrical power available on an electrical bus is usually limited. When a pump is coupled or is integrated with an electric motor, substantial savings of cost, weight, size and power demand of the motor can be realized, with soft cut-off control.

4.2.3.2 Typical Performance

This control allows by reducing the slope of the pressure/flow curve to achieve two objectives:

- Provide maximum flow at reduced pressure
- Develop full pressure at zero flow.

This control has the overall effect of reducing demand for high power. This control is usually used on relatively small electric motorpumps installed on aircraft intended as single usage such as the landing gear control. For example, when the landing gear is deployed, it is desirable to lower it as quickly as possible, requiring high flow at reduced pressure. Near the end of travel the landing gear needs to be locked in place, requiring low flow at high pressure. In reverse, the retraction of the landing gear requires similar parameters.

figure 7 shows that the maximum pressure demanded at full flow is significantly reduced when compared to a flat cut-off compensator.

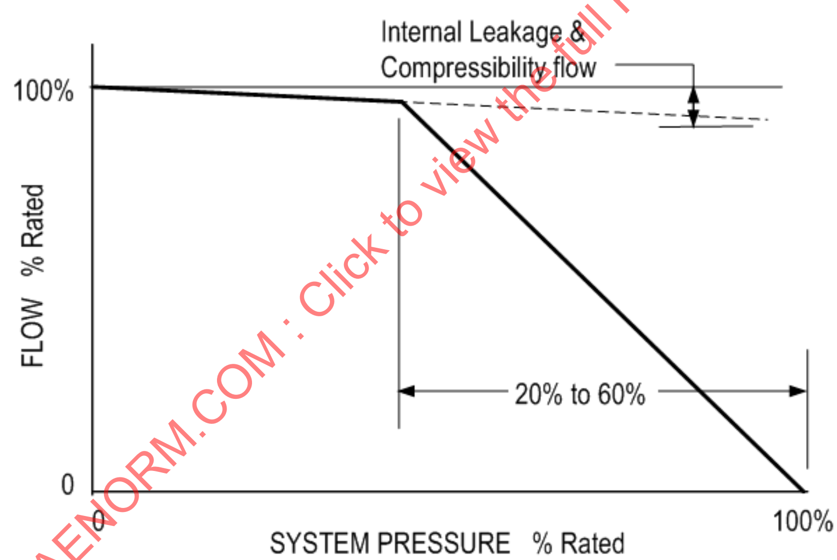

FIGURE 7 - TYPICAL PERFORMANCE CHARACTERISTIC – SOFT CUT-OFF

4.2.3.3 Concept Description

The soft pressure cut-off is achieved simply by introducing control pressure to the spring side of the flat cut-off compensator spool (in lieu of pump case pressure), as illustrated in the schematic in figure 8. In practice, an orifice is also required to maintain stability.

4.2.3.4 Schematic

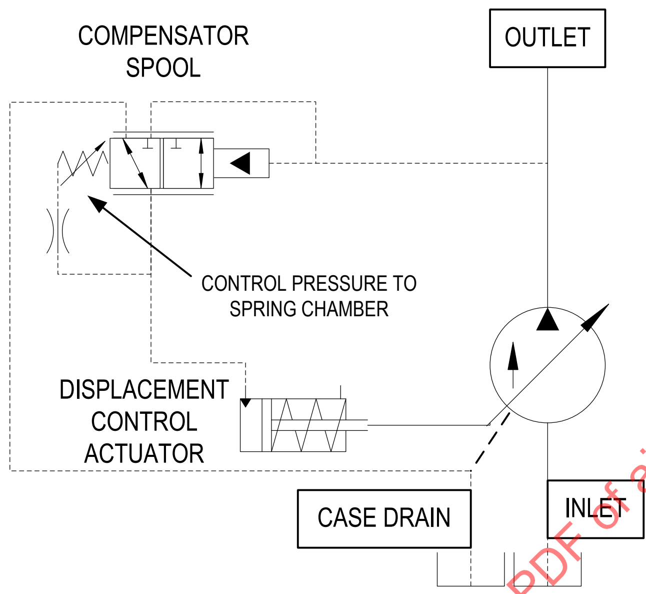

FIGURE 8 - SCHEMATIC – SOFT CUT-OFF

4.2.3.5 Limitation Of Usage

This control is primarily used when the driver of the pump is limited in power output (such as an electric motor in an aircraft, either DC or AC) as described in the previous paragraph and usually when the system is simple and does not include multiple branches. This control is not recommended for an for an engine driven pump that supplies an entire hydraulic system since it will significantly reduce the power available to critical actuators when other systems are also demanding power.

It should also be noted that this control poses stability challenges as the break point is moved to the lower levels. For most practical designs, setting the break point to 75% of full pressure is the ideal design goal.

4.2.4 Dual Range Pressure Control

4.2.4.1 Introduction

This control provides two separate controlled output pressure settings in the same pump. Selection is usually made by means of an external signal (electrical, mechanical or hydraulic usually set by an onboard computer). This control effectively introduces two independent settings to a single flat or soft cut-off compensator.

This concept is used on the F/A-18E/F pump to allow selecting either 3000 or 5000 psi operating pressure. The lower pressure is used for most flight conditions while the higher pressure is used when extreme maneuvers are performed requiring greater control surface loads. The dual pressure control prolongs system life, as the high pressure mode is only used for a fraction of the operating time. During low pressure operation, the pump extracts less power from the engine, pump and system losses are reduced, and the heat generated in the system is also reduced. A related benefit is that the pump may be designed with smaller and lighter bearings and other working components since the duty cycle is less severe.

4.2.4.2 Typical Performance

The dual pressure performance is similar to that of the flat cut-off compensator, except that it is able to function at two distinct, predetermined cut-off pressure points. This is illustrated in Figure 9.

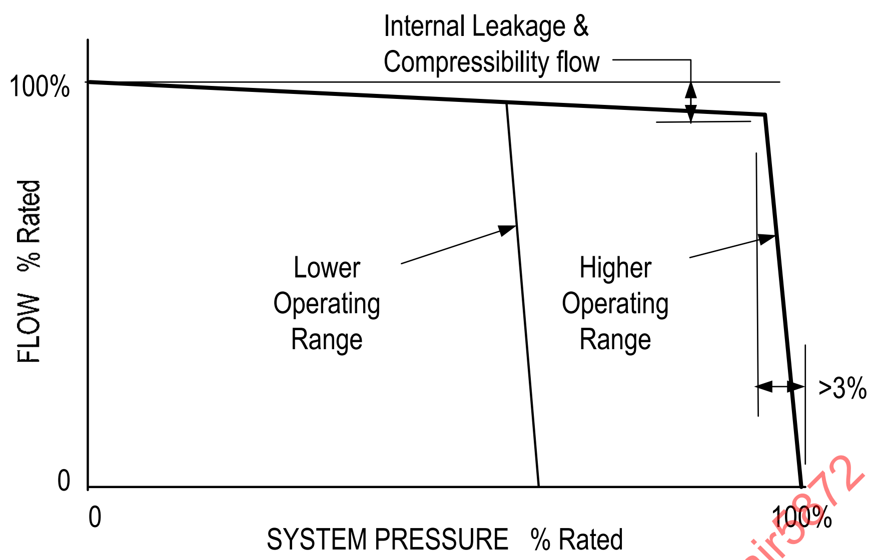

FIGURE 9 - TYPICAL PERFORMANCE CHARACTERISTIC – DUAL RANGE PRESSURE CONTROL

4.2.4.3 Concept Description

As previously explained, the pre-load on the pressure compensator spring determines the system pressure at which the pump begins regulating. Compressing the spring (increasing pre-load) causes the pump to regulate at a higher pressure, and conversely, relaxing the spring (decreasing pre-load) can be used to provide regulation at lower pressure.

As shown in Figure 10 the compensator spring is in the extended position (lower pre-load) when there is no pilot pressure at the Pilot Pressure port. This means that lower system pressure is required to overcome the spring pre-load force and the pump will regulate at its lower range.

When the pressure is applied at the Pilot Pressure port, the spring is compressed until the pilot piston stop is reached. The position of this stop corresponds to an additional spring pre-load, and thus determines the system pressure at which regulation in the higher range occurs. The pilot pressure must be sufficient to hold the piston firmly against the stop, even during transient and cyclic pressures. This pressure can be provided from an external source or it can be supplied from the pump output through a solenoid-controlled valve integrated within the pump.

A variation of this scheme uses pilot pressure that is continuously variable that can position the spring to any point from the minimum to maximum setting. The pilot pressure is typically controlled by an electrohydraulic servovalve with pressure feedback, enabling the pump to adjust system pressure by electronic control.

4.2.4.4 Schematic

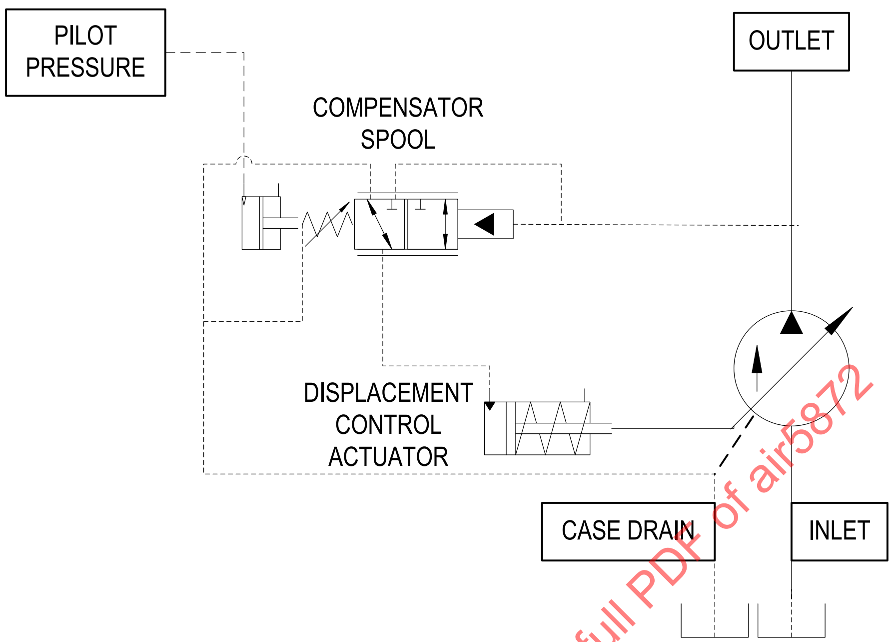

FIGURE 10 - SCHEMATIC – DUAL RANGE PRESSURE CONTROL

4.2.4.5 Limitation Of Usage

Although there is no limitation to applying this control, the system integrator should recognize that high performance pumps are optimized for pressure ripple and dynamic response in a narrow operating range of pressure and speed. Therefore, the integrator should specify whether the principal design condition is the lower or higher pressure range, then allow greater latitude in the requirements at the secondary pressure mode. The integrator should also specify the dynamic characteristic when shifting modes from low to high and high to low.

4.3 Torque Limiting, Unloading and Blocking Controls

4.3.1 Overview

Torque limiting controls are used in conjunction with pressure limiting controls to limit the input power from the pump drive. This is useful for reducing pump drag torque during engine start, or for limiting the power that can be extracted from electric motors to limit the load on the electrical system.

4.3.2 Electrical Depressurization Valve (EDV) Control

4.3.2.1 Introduction

A feature which can be added to the pressure compensated control is the electrical depressurization valve (EDV) which upon an external electrical command, places the pump into a lower output pressure mode.

4.3.2.2 Typical Performance

The typical EDV performance is depicted in Figure 11.

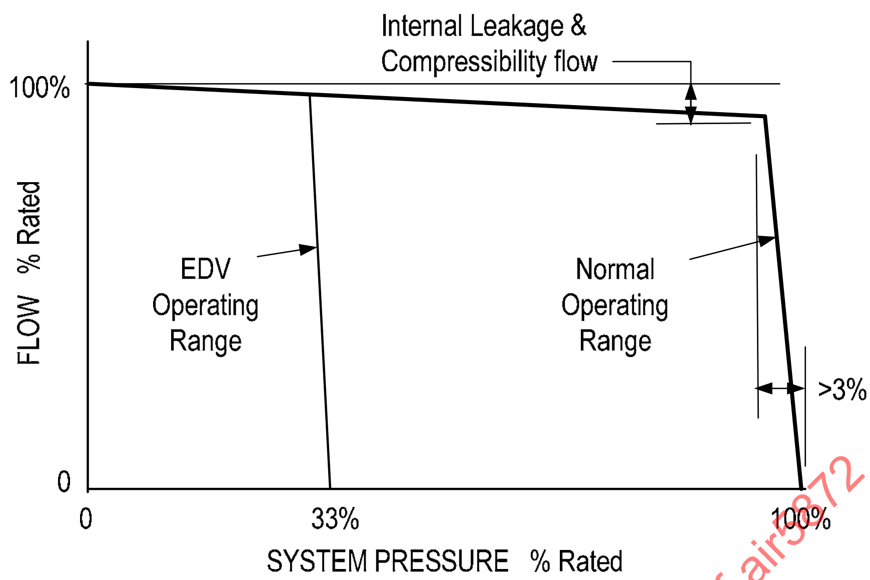

FIGURE 11 - TYPICAL PERFORMANCE - EDV PUMP CONTROL

Each pump manufacturer has some latitude in the EDV pressure design by varying the internal pump parameters to modify the pump output pressure in the EDV mode as required by the system designer. The EDV pressure ratio may be reduced to as low as 20% in case it is desired to minimize system leakages without adverse pump effects. However the internal pump losses due to friction, viscous and churning are not affected by reducing EDV pressure, so the torque demand and heat generation in the pump are not significantly reduced compared to a 33% ratio. Only the leakage losses are reduced. Because of this, should the pump be required to stay in the EDV mode for extended periods, specific provisions must be made by both the system and the pump designer to remove the heat generated from the pump. This can be done by providing sufficient case circulation inside the pump and ensuring the hydraulic system can deal with the heat load during periods of depressurized operation.

If the system has multiple pumps in the same circuit, EDV may be used to reduce the torque during cruise. In a transport aircraft system, having reached cruising altitude, the duty cycle can be satisfied with only one pump running in full pressure mode in each system. When descending, the depressurized pumps can be brought back online as higher load demand and lower pump speed is anticipated.

4.3.2.3 Concept Description

The EDV depressurizes the pump by overriding the pressure compensator, thereby reducing the pump pressure where cut-off occurs. The pressure in the depressurized condition is typically one-third normal operating pressure, which can effectively reduce pump torque demand by as much as 70%. This results from two phenomena working together: first, the pump is operating at reduced pressure which demands lower torque, and second, the quiescent leakage is reduced, thereby decreasing the pump displacement and reducing torque demand.

The EDV control is illustrated in the schematic of Figure 12. The pressure compensator is similar to the flat cut-off as previously. The EDV consists of a solenoid valve commanded externally that ports pressure to a large EDV piston, forcing the compensator spool to full deflection at typically one-third of the normal pre-set pressure.

An optional blocking valve, as shown in the schematic, may be added to the outlet port. This valve is also actuated by the solenoid pilot pressure and shuts off the pump output from the external hydraulic system. This permits the downstream system to depressurize to the base reservoir pressure, even while the pump is operating at EDV pressure.

4.3.2.4 Schematic

A typical schematic is depicted in Figure 12. It should be noted that designs may vary as each pump manufacturer has multiple designs adapted for specific applications.

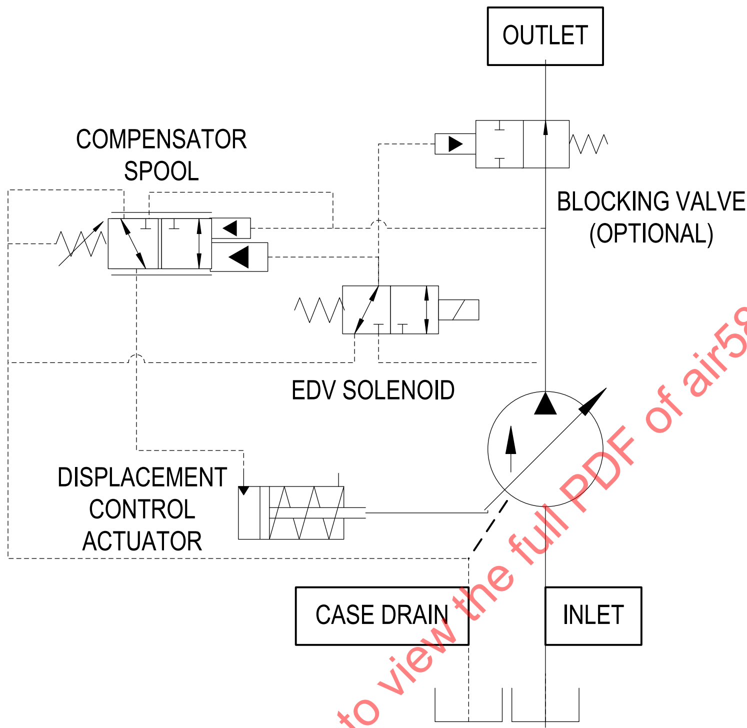

FIGURE 12 - SCHEMATIC - TYPICAL EDV CONTROL

4.3.2.5 Limitation Of Usage

When applying EDV to an engine pump, the pump's torque-speed curve in EDV mode should be reviewed to assure the engine can overcome the peak torque demand. This is particularly true for windmilling restarts, in cases where the engine has come to a full stop. The reason is that during startup, the pump begins at full displacement and reaches EDV pressure prior to destroking. The torque at this condition peaks at a level higher than the long-term steady-state value.

4.3.3 Torque (Power) Limiting

4.3.3.1 Introduction:

Most systems requiring high pressure and high flow do not necessarily require both at the same time. In fact, actuators tend to require high flow when load resistance is low, then reduced flow as resistance increases. In such cases, it is possible to provide a pump control that matches the actuator demand and also limits heat generation in the hydraulic system as well as the power extraction from the drive.

This control is well adapted to pumps driven by prime movers of limited torque such as electric motors or Ram Air Turbines (RATs) as commonly found on-board many aircraft. In the majority of cases, power limiting is used on electric driven pumps, as the electrical power is generated by aircraft alternators is of limited physical sizes and power output.

Torque limiting control is more effective than the soft cut-off because it approximates a constant torque control more closely, and is thus capable of delivering more hydraulic power for the same peak torque demand.