AEROSPACE INFORMATION REPORT	AIR1228	REV. A
	Issued 1972-11 Revised 2009-01 Reaffirmed 2014-09 Superseding AIR1228	

(R) Standard Impulse Machine Equipment and Operation

RATIONALE

AIR1228A has been reaffirmed to comply with the SAE five-year review policy.

1. SCOPE

This SAE Aerospace Information Report (AIR) establishes the specifications and descriptions of the critical components and operational guidelines for the standard hydraulic impulse machine for testing hydraulic hose assemblies, tubing, coils, fittings and similar fluid system components.

This revision to the AIR1228 provides a description of a system that meets the requirements for specifications including: AS603, AS4265, and ARP1383. This impulse system utilizes closed loop servo control with specifically generated command signal waveforms.

Data accuracy and integrity are emphasized in this revision. Knowing the uncertainty of the pressure measurement is important whether using a resonator tube system, as described in the original release of this document, or a closed-loop systems as described in this release. The accuracy of the data measurement system and consistency of the pressure waveform are fundamental to test validity, regardless of the system type. This is discussed in more detail in Section 5.

The standard impulse test system is established for the following purposes:

A. As referee in the event of conflicting data from two or more nonstandard impulse machines. Such a referee system might be built by an impartial testing activity.

B. A design guide for future test systems being built by manufacturers and users, or the upgrading of present systems.

C. A design guide for higher pressure test systems or special purpose machines being designed.

It is not the intention of this document to obsolete present resonator type machines. The specification for the resonator type machine is available in the original release of this document.

A simple block diagram of the revised system is shown in Figure 1.

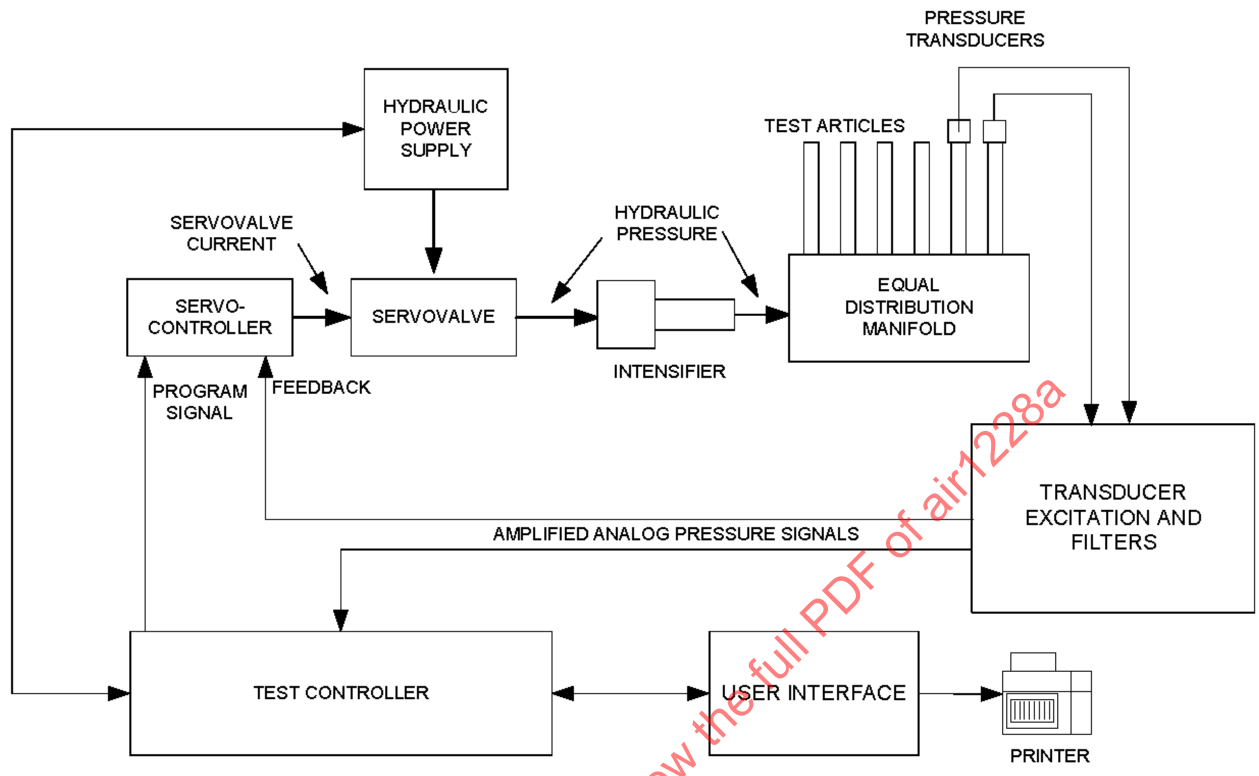


FIGURE 1 - SIMPLE BLOCK DIAGRAM OF SERVO CONTROLLED IMPULSE SYSTEM

The simplified block diagram shown in Figure 1 does not use a resonator tube as in the original release of the specification. To eliminate undesired resonances the hydraulic system is as closely coupled as possible. In this design, the pressure waveform is created in closed loop proportional control via the command signal generated by the test controller system.

If the test controller is a computer with an impulse specific software application, it is possible to control all parameters of the specified waveform, within the capabilities of the hydraulic system. These parameters include things such as rise rate, damping, secondary oscillation amplitude, etc. In a resonator tube system, it is difficult to make adjustments to these parameters.

2. REFERENCES

2.1 Applicable Documents

[1.] Keller, George R., Hydraulic System Analysis, Penton/IPC, 1969

[2.] Merritt, Herbert E., Hydraulic Control Systems, John Wiley and Sons, Inc., 1967

[3.] D'Azzo and Houpis, Linear Control System Analysis and Design, 1981

[4.] Nelson and Englund "Proposed revisions to AIR1228, AS4265, and AIR4298", SAE G-3 Conference, 3/14/2005

[5.] Mills Jr., Blake D., "The Fluid Column", American Journal of Physics, April 1960

2.2 U.S. Government Publications

Available from US Government Printing Office, 732 North Capitol Street, NW, Washington, DC 20401, 202.512.0000, http://www.gpo.gov/.

MIL-PRF-5606	Hydraulic Fluid, Petroleum Base
MIL-PRF-83282	Hydraulic Fluid, Fire Resistant, Synthetic Hydrocarbon Base
MIL-PRF-87257	Hydraulic Fluid, Fire Resistant; Low Temperature, Synthetic Hydrocarbon Base, Aircraft and Missile

2.3 Definitions

General: The impulse test system can be divided into four subsystems.

HYDRAULIC SYSTEM

The hydraulic system provides the power for generating the impulse pressure waveform. Hydraulic components include the hydraulic power supply (HPS), supply and return lines, accumulators, servovalve, intensifier, and test manifold as described in Figure 2. If the test pressure is higher than the HPS output, the system utilizes an intensifier. If the test pressure is lower than the HPS output, the system operates without an intensifier (also called straight servovalve). The system shown in Figure 2 can be switched to either mode with three manual valves. This document will use "servovalve" to represent all types of valves that can be used in a proportional manner for pressure testing.

CONTROL SYSTEM

The control system includes the test controller and the servo-controller in Figure 1. Physically this can be one integrated unit, or it can be comprised of several interconnected subsystems. This system controls the HPS, generates the pressure waveform signal in accordance with standard specifications, closes the control pressure control loop (servo-controller), and ensures pressures are within the test specification.

INSTRUMENTATION SYSTEM

The instrumentation system consists of the pressure transducers, cabling, signal conditioning, amplifiers and filters used to convert the pressure measurement into electrical signals as shown in Figure 3. The amplifiers and signal conditioning may be integral with the pressure transducers. When using computerized data acquisition, the analog-to-digital converters and associated software application are also considered part of the instrumentation system.

TEST ENCLOSURE OR ENVIRONMENTAL CHAMBER

The test enclosure provides a safety barrier for testing high pressure specimens. When required, the enclosure may be an environmental chamber capable of controlling the test temperatures of the air surrounding the test specimens (and the fluid temperatures inside the test articles) within the parameters of the test document.

3. DETAILED DESCRIPTION OF HYDRAULIC COMPONENTS

Figure 2 shows the general arrangement of hydraulic components in the impulse system. The system shown is capable of straight servovalve operation for low pressure testing and intensifier operation for higher test pressures. Operation can be switched to either mode with three manual valves. The servovalve and intensifier are shown in a 3-way configuration as described in the servovalve section.

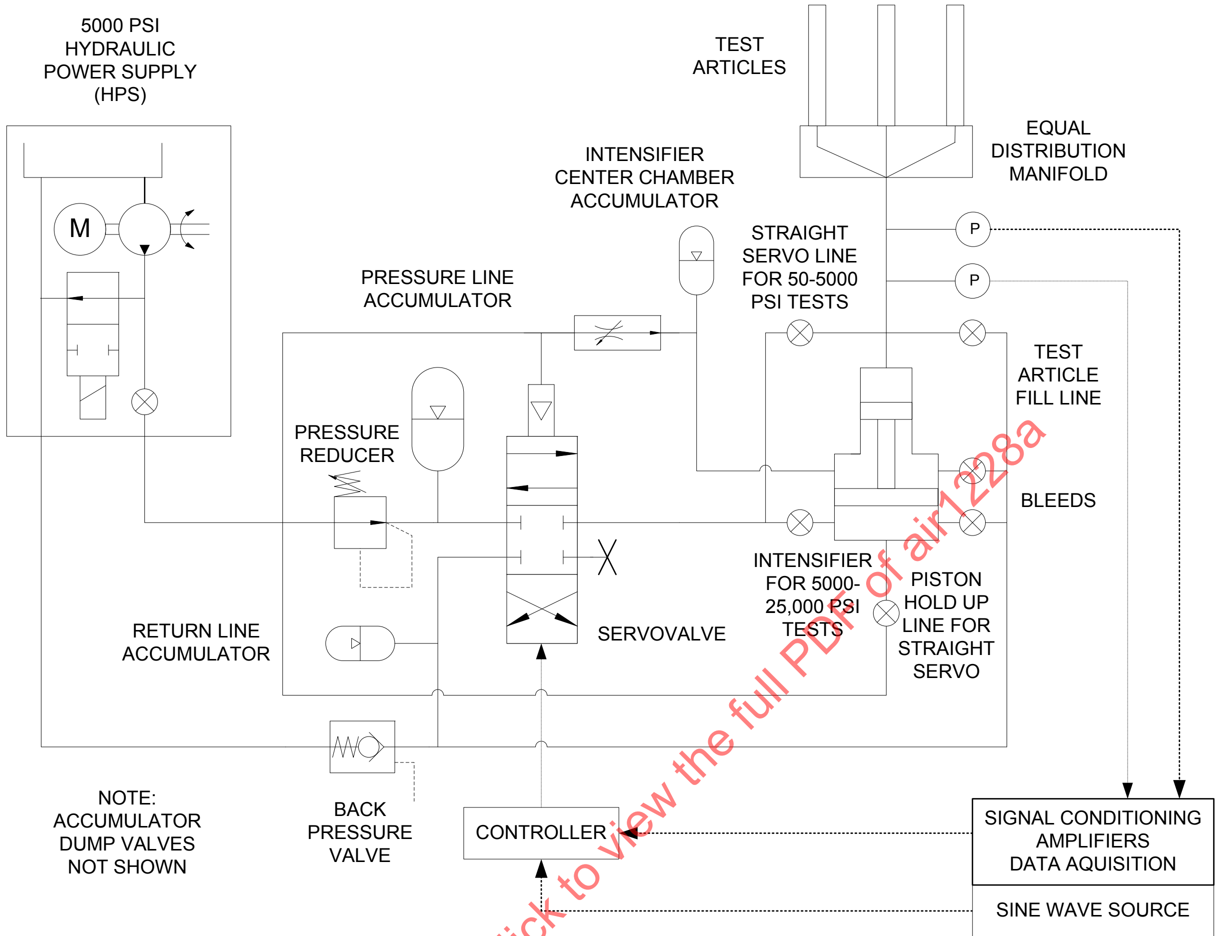


FIGURE 2 - HYDRAULIC SCHEMATIC

3.1 Hydraulic System Sizing

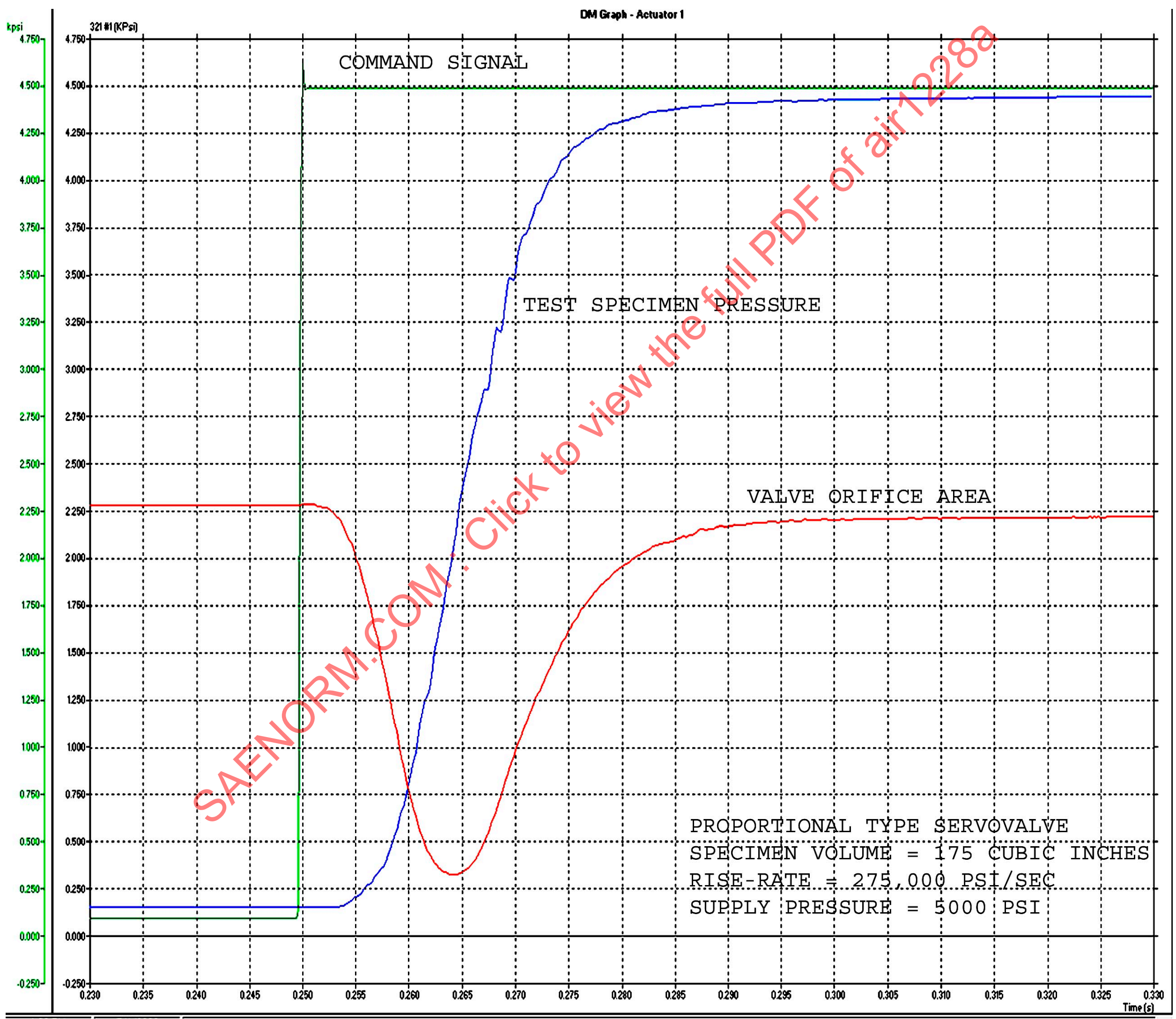
The test system size is typically determined by desired test volume, by available power for the HPS or by cost. One of these three factors typically determines the other two. In determining the test volume, or "specimen load" one must consider the quantity required by the test specification and how many specimens can be run concurrently. From an operational standpoint, it is most efficient to maximize the number of specimens run under a single condition which leads to larger test system size.

All components need to be sized relative to each other. Undersized components will limit desired performance, while oversized components cost more and may have adverse effects on performance. The system must be designed for both pressure and flow requirements

3.2 Closed Loop Control

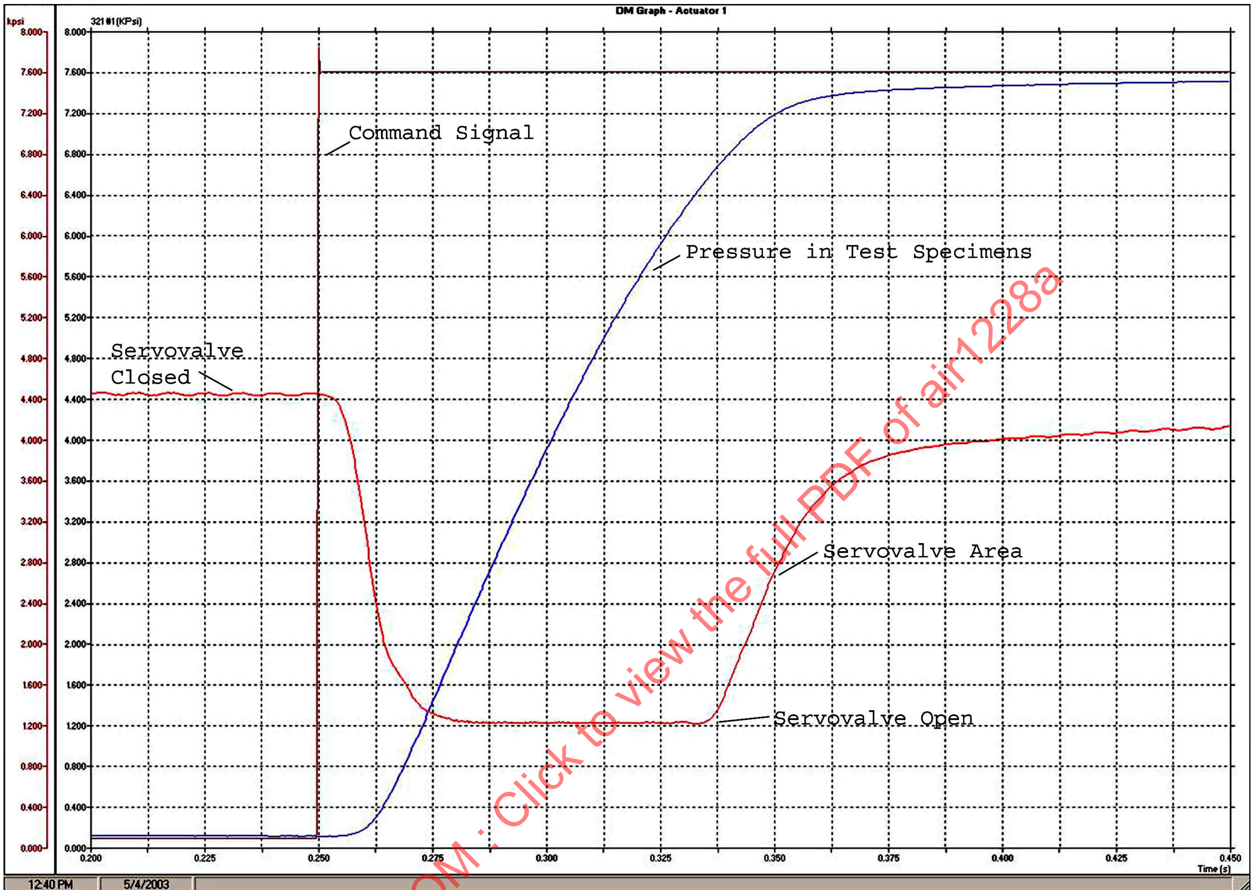
In the resonator impulse system designed for AS603 type testing, a solenoid valve is used and the supply pressure to the solenoid is the same as the desired plateau pressure. To start the waveform the solenoid valve is turned on, and goes to the fully open position until the end of the plateau pressure waveform timing and then completely closes. In a closed-loop system, a servovalve is used instead of a solenoid valve. The servovalve opening can be controlled precisely and varied in accordance with the difference between the desired pressure output and the actual pressure output.

In a closed-loop control system where the desired pressure rise times are less than twice the time it takes the servovalve to open, the orifice area when plotted versus time will resemble an inverted "V" as shown in Graph 1. In this graph the loop gain of the control system was set as high as possible without causing over-shoot of the pressure wave. The command signal is a step (square wave) 4500 psi input. The servovalve appears to open 0.0025 s after the command and the pressure begins to rise within a millisecond after that. At the start of the commanded rise, the orifice area will start increasing, and mid-way through the pressure rise it will start to close again. The supply pressure to the servovalve is usually at least 500 psi higher than the maximum desired pressure. When the desired pressure in the test volume is reached, the servovalve will once again be closed, trapping the added volume of fluid inside the test specimens. This sequence will be referred to later in this document when an example is presented with rough calculations for sizing the servovalve and related components. It is apparent from examining the valve area trace in Graph 1, that the valve dynamics play a significant role in the ability to meet pressure impulse rise times that are up to twice the time it takes to fully open the valve.



GRAPH 1 - CLOSED LOOP CONTROL WITH SHORT RISE TIMES

When the servovalve is used in a closed-loop control system where the rise times are more than twice as long as the time it takes the servovalve to open, the valve will open completely and stay in that position while the pressure increases and then it will begin to close again. In tests with this range of rise rates, the maximum flow rating of the servovalve becomes more important than the valve dynamics in determining rise rate. This is shown in Graph 2.



GRAPH 2 - CLOSED LOOP CONTROL WITH LONG RISE TIMES

Graphs 1 and 2 show that the servovalves need to be sized in terms of their maximum ratings as well as their response. This is true of other components as well. One purpose of this document is to help choose the components that will provide the specified parameters for a wide variety of test applications.

Table 1 shows an example of a 35 gpm HPS system with a variety of specimen volumes, typical component sizing and test system performance.

TABLE 1 - HYDRAULIC SYSTEM SIZING

Specimens			Test Specifications					Test System Sizing					
Qty	Type and size	Volume (cu in)	Spec	Peak Press (psi)	Temp (F)	Test Rate	Rise Rate (psi/sec)	Total HP volume (cu in)	Supply Press (psi)	SVV (qty & gpm)	SVV operation	Intensifier (HP/LP dia, ratio)	HPS Flow (gpm)
12	AS5620-16 X .051 X 25"	190	AS603	4,500	amb	70 cpm	180,000	220	5,000	(1) 60 gpm	3-way	None	7
11	AS5620-16 X .065 X 25"	150	AS603	7,620	amb	70 cpm	180,000	190	3,000	(2) 60 gpm	3-way	4 X 7, 3:1	23
4	AS5951-16 X 23" hose	55	AS603	7,620	400	70 cpm	180,000	95	3,000	(2) 60 gpm	3-way	4 X 7, 3:1	23
2	AS5620-16 X .065 X 10"	0.5	AS4265	20,000	amb	14 Hz	820,000	10	3,000	(2) 60 gpm	3-way	1.75 X 5, 8:1	35
12	AS5620-16 X .065 X 10"	3.5	AS4265	20,000	amb	10 Hz	600,000	13	3,000	(2) 60 gpm	3-way	1.75 X 5, 8:1	35

Notes:

Hydraulic oil in the system is Mil-PRF-83282
Ambient temperature condition assumes the hydraulic oil is 100 F
AS603 test articles are AS5620 S-tubes and a similar length 90 degree hose for comparison purposes
AS4265 test specimens are supported by AS4298 internal stiffeners which fill their volume
Pressure accumulator is 2.5 gallons, return and intensifier center chamber accumulators are both 1 gallon
Servo valve has a 0.016 second step response
Intensifier stroke is 2 inches. Start position is .5 inches from the bottom and the stroke during impulse is between .4 and .7 in these cases
HPS flows include approximately 2 gpm leakage flows without intensifier and 6 gpm with intensifier
AS4265 cases are maximum speed as allowed by a 75 HP 35 GPM HPS resulting in the given frequency and rise rate

The hydraulic pressure in a typical impulse system runs at two levels, the pump output pressure (supply pressure), and peak impulse pressure at the test specimen which is controlled by the servovalve. The pump will typically run at a constant value between 2000 psi (minimum pressure for servovalve pilot stage in most impulse applications) and 5000 psi. This supply pressure should not vary more than a couple hundred psi or it will affect the operation. A large variation may be eliminated with the proper accumulator. See the "accumulator section" for more information on accumulator sizing. By maintaining a separate supply pressure to the servovalve pilot stage, the supply pressure to the servovalve final stage may be adjusted for the type of component that is being tested.

The pressure at the test specimens can run at radically different ranges to give a wide spectrum of test flexibility. The pressure at the test specimens can be varied from 100 psi (straight servovalve) to 20 000 psi (with an 8:1 intensifier). With the straight servovalve configuration, the pressure down stream of the pressure reducer (shown between the servovalve and the pump in Figure 2) can run as low as 100 psi for reservoir testing (ARP1383) or fuel system component testing. With the reducer set at the 1000 to 2000 psi range, aluminum hydraulic return lines can be tested with the AS603 wave form. With the HPS running at the 2000 psi range, 1500 to 2000 psi titanium return tubes can be tested with the AS603 wave form. With the HPS running at 5000 psi, all 3000 psi components can be tested at 4500 with the AS603 wave form. With an 8:1 intensifier in the system, test pressures up to 20 000 can be achieved for the AS4265 wave form. The components used in the impulse test systems must be sized with these pressures in mind.

The flow in the system also runs at two rates, the low steady state flow due to accumulator recharging and system leakage, and the peak flow due to the pressure impulse rise rate. The pump and lines leading to the pressure accumulator and the lines from the return accumulator back to the pump reservoir all run at the low speed flows. For the 35 gpm system sized in Table 1, the peak flows can be over 200 gpm across two servovalves in a parallel configuration. Impulse test system components must be sized with these flows in mind, because it is the peak flow capability that determines the performance of the system.

3.3 Fluid

The fluid in the system must be compatible with the type of testing that is being conducted. For high temperature testing, a high flash point fluid such as MIL-PRF-83282 should be used. For lower temperature testing a less expensive petroleum based fluid such as MIL-PRF-5606 may be used. Impulse testing of commercial aviation components with O-rings typically requires the use of a phosphate ester hydraulic fluid. Many other types of fluid are used and it is sometimes necessary to use a fluid separation device (see below) when the required test fluid can not be used in the HPS.

Each fluid has its own unique bulk modulus characteristics which affect the stiffness of the system and thus the test system demand. As the bulk modulus decreases, the system requires more flow to meet the same pressure waveform profile. The fluid bulk modulus changes with both temperature and pressure. The bulk modulus can change 50% between the pressure ranges required for AS603 and AS4265. The bulk modulus can change by a factor of 3 through the range of temperatures required for hose testing. When sizing a system, the fluid specifications, temperature and pressure ranges need to be taken into consideration.

Entrapped air dramatically affects the effective bulk modulus of the fluid and thus the system demand. The flow requirement to achieve a desired rise rate is directly proportional to the inverse of the effective bulk modulus. Provisions must be made to bleed the air out of the system. This may be accomplished by flushing all the passages with high velocity flows or by orienting all the hydraulic passages so that the air rises to a bleed port at the highest point. When hydraulic fluid is un-pressurized, the air that attaches to the fluid is called entrained air. After the test system is started and warmed up, the remainder of un-bled entrapped air and the entrained air is dissolved into solution and the system becomes slightly stiffer.

The effective bulk modulus is used to calculate the stiffness, hydraulic natural frequency and pressure in a closed volume. The formula for the effective bulk modulus (from Reference [2]) in psi is shown in Equation 1:

$$\beta e = \frac{1}{\frac{1}{\beta_f} + \frac{V_g}{V_t}\left(\frac{1}{\beta_g}\right) + \sum \frac{V_c}{V_t}\left(\frac{1}{\beta_c}\right)} \quad \text{(Eq. 1)}$$

where:

β_f is the fluid bulk modulus (psi)

β_g is the bulk modulus of the entrapped gas (psi) for air (this is 1.4 * Pressure)

β_c is the component (tube, hose, etc.) bulk modulus (psi)

V_g is the volume of gas in the system (cubic inches)

V_t is the total volume of the system (cubic inches)

V_c is the volume of the component in the system (cubic inches)

In the summation of the denominator in Equation 1, the bulk modulus of each component or tubing section, β_c, is multiplied by the ratio of that parts volume to the total volume, V_c / V_t .

The bulk modulus component equation for thin wall tubes or intensifier barrels is approximated (from Reference [2]) as shown in Equation 2:

$$\beta_{tn} = T * E / ID \quad \text{(Eq. 2)}$$

where:

T = wall thickness (in)

E = modulus of elasticity (psi)

ID = inside diameter of the component (in)

The bulk modulus component equation for a hose can be approximated (from Reference [2]) as shown in Equation 3:

$$\beta_h = V_h * \Delta P / \Delta V_h \quad \text{(Eq. 3)}$$

where:

V_h = volume of the hose (in^3)

ΔP = the pressure range which will be applied to the hose (psi)

ΔV_h = hose volumetric expansion (typically supplied as volume per unit length)

3.4 Hose Bulk Modulus

Hose volumetric expansion is a non-linear relation with pressure. The understanding of this relationship may be extremely critical when designing a system for testing hoses. It is best to use the manufacturer's specifications to determine the exact value at the specific pressure where the hose is used or tested. If that data is not available, a simple test can be run by pressurizing the hose to the maximum pressure specified in the test, and bleeding off the pressure in increments into a measuring beaker. By recording the pressure increments and volume increments, the hose volume can then be plotted versus pressure and the volumetric expansion can be predicted for the specific pressure range. The bulk modulus of hoses varies significantly depending on size and pressure and is usually much lower than metal tubing.

3.5 Tubing Bulk Modulus

A 3000 psi system comprised only of metallic components and tubing, such as titanium or steel will be much stiffer than a system with hoses. A total effective bulk modulus around 100 000 psi can be used as a starting point, but this varies with temperature, pressure, test specimen compliance and the amount of entrapped air so it should be calculated for the specific application.

When determining an approximate value for effective bulk modulus it may be necessary to average the values between peak and low pressures, high and low fluid temperature, etc. The change in bulk modulus with variations in other parameters can be very significant.

3.6 Fluid Separation

There are impulse applications where specific fluids must be used to test the component properly. For example, phosphate ester fluid must be used for commercial airplane components with O-rings. Jet fuel or fuel substitute (such as, Stoddard solvent) may be required in fuel system component testing.

For testing with phosphate ester fluids, the entire impulse system can be made with phosphate ester compatible components. In that case, special attention must be paid to those seals installed inside of the industrial hydraulic components of the impulse machine. In some O-ring testing, compatibility with several different fluids is necessary and the O-ring must be tested with each fluid. In this case, the "three chamber" intensifier makes a good oil separation device. Fluid from the HPS to the bottom of the intensifier can be any mix of phosphate ester and the fluid in the top intensifier chamber up to the specimens can be the specific phosphate ester under test. The leakage across the high pressure intensifier seal will usually be insignificant so the integrity of the fluid under test will be preserved. The phosphate ester under test can then be removed and replaced with the next test fluid. If an intensifier is not necessary for the impulse machine, a 1:1 piston accumulator will accomplish the same fluid separation task. This must be sized with the principles given in the accumulator section.

For impulse testing of fuel system components with jet fuel or fuel substitute (Stoddard solvent), a hydraulic impulse system may also be used with a fluid separator. In this case, two actuators with mechanically coupled rods physically separate the hydraulic fluid from the fuel or solvent while providing visibility of leakage if the actuator seals fail. If this low pressure testing is accomplished with a multi-purpose high pressure impulse system, the actuator areas can be sized to provide a de-intensified ratio. For example, a 1:4 de-intensification ratio run at 1600 psi supply pressure will provide 400 psi to the test specimen.

3.7 Hydraulic Power Supply

The hydraulic power supply (HPS), commonly referred to as a hydraulic bench, provides the required pressure and flow of hydraulic fluid. It should be equipped with adequately sized oil filtration and cooling, high temperature and low reservoir shutoffs, current overload protection, and a fast pressure dump valve. In advanced test system designs, the hydraulic supply alerts the test failsafe system when it shuts down (in any HPS failure mode) activating all other failsafes in the system.

The hydraulic power supply needs to be sized with consideration for all system flows including flow due to peak impulse pressure at test frequency, leakage flows (for example servovalve and piloted valves which have continuous flows to tank) and recirculation flows (for test fluid temperature conditioning and intensifier recirculation).

3.8 Accumulators

There are two to three accumulators in this hydraulic system design, each required for a different purpose. These accumulators are shown in Figure 2. The supply line accumulator provides flow for the pressure impulse while the return accumulator absorbs the return line surges. The supply line accumulator provides the peak flow required for the high rise rate impulse because hydraulic power supplies typically have very slow pump response times. A third accumulator is required when a three-chamber intensifier is used in conjunction with a servovalve driven in a 3-way configuration. A higher flow rate can be realized from the servovalve by using it in a 3-way configuration where one output port is blocked and the other port is used to drive the low pressure side of the intensifier (see servovalve section). If the intensifier has three chambers, the center chamber is then driven with an accumulator. The center chamber accumulator (typically set at 400 psi) absorbs the flow in and out of the chamber with each impulse. Reference [1] provides a section for calculating the size and pre-charge of gas-oil accumulators for specific applications.

The supply pressure accumulator should be sized with consideration to pressure droop during impulse rise time. An undersized accumulator with significant droop will lower the pressure supply to the servovalve and limit system output. Return and center chamber accumulators should be sized with consideration of the rise in pressure. An undersized return accumulator will not absorb the return line surges. An undersized center chamber accumulator will increase the pressure demand on the system during impulse cycling.

All accumulators should be sized with consideration to exit port flow and accumulator speed. Accumulator flow should be sized with regard to the peak flows occurring during the impulse cycle which can be several times the pump flow capability. Bladder style accumulators have an anti-extrusion valve which restricts flow more than piston style. However, since bladder style accumulators have faster response than the piston style, they should be used when possible. Bladder accumulators are typically recommended when the application response requirement is less than 0.025 s. Since nominal impulse rise times are in this time frame, bladder accumulators are recommended. However, bladder compatibility with fluid must be considered particularly when phosphate ester based fluids are used. When piston accumulators must be used for fluid compatibility, the piston diameter should be sized to limit piston seal speed to 20 inches per second. An undersized piston will cause high piston velocities and may result in premature seal failure. Diaphragm accumulators typically have small exit ports restricting their effectiveness.

The supply pressure accumulator should have a blocking valve so, upon emergency shutdown, its pressure is isolated from the servovalve and gradually released to return (reservoir). All accumulators should have free passage to reservoir after system shutdown, so that no trapped pressure remains.

The accumulator used on the center chamber of the intensifier may require a fast dump feature in some test applications. This will prevent air from being pulled into the test articles. Some test articles with non-metallic seals are exposed directly to atmosphere. Examples include O-ring test spool and block where the O-ring leakage can be collected outside the block, and AS4265 internally sealed and supported straight tubes. In both instances, air can be drawn back into the system if the accumulator on the intensifier center chamber releases pressure slower than the supply pressure accumulator, forcing the intensifier down and pulling a vacuum on the specimens. In these instances, a fast pressure "dump" valve ported to return, on the center chamber may be required so a negative pressure does not draw air into the test articles.

3.9 Hydraulic Lines and Passages

There are two opposing design criteria which govern hydraulic line and passage sizing. Minimizing hydraulic volume between the servovalve and the test specimen reduces overall system demand and component size. However, hydraulic lines and passage diameters need to be sized large enough to minimize pressure drop. As mentioned above, the peak flows during the impulse pressure cycle rise can be several times the pump flow capacity and these flow values must be used to determine line and passage fluid velocities.

The industrial hydraulic industry recommends maximum line velocities of 30 feet per second for pressure supply lines and 10 feet per second for return lines. All lines and manifold passageways to the specimens should be evaluated for pressure drop due to under-sizing. If the line and passage lengths are minimal, higher speed can be tolerated without detrimental pressure drop. Line bends and elbows should be kept to a minimum also.

Hydraulic tubing line lengths should also be minimized to keep resonance frequencies above the operational bandwidth. For example, the initial pulse of a 3000 or 5000 psi AS603 waveform with typical rise rate is usually simulated with a ring frequency around 12 Hz. In order to reproduce this waveform, the test system natural frequency should be sufficiently higher to avoid waveform degradation. When the natural hydraulic frequency of the tubes, manifold and test volumes is close to the programmed frequency of the waveform, the oscillations will not only affect the quality of the waveform but also affect the control system stability. The most critical lines to minimize are between the servovalve and the intensifier and between the intensifier and the test article. With a straight servovalve system, the most critical lines to minimize are between the servovalve and the end of the test article. System natural frequencies should be calculated between the servovalve and the end of the test article and include the intensifier on the intensified system. It is possible to have more than one resonant mode depending on the tubing configuration. As an example there may be one resonance in the tube connecting the servovalve to the intensifier, another hydraulic resonance that includes the mass of the intensifier piston and another for the high pressure side of the intensifier with the test specimen volumes. Reference [2] has equations and examples for calculating these resonant frequencies.

The dominant resonant mode when using an intensifier includes "hydraulic springs" in series. These hydraulic springs consist of the compliant fluid columns in the low pressure side of the intensifier and the tubes connecting the low pressure side to the servovalve - and the fluid columns in the high pressure side of the intensifier, the test specimens and the passages or tubing connecting the high pressure side of the intensifier to the specimen manifold.

The compliance, K, of a hydraulic spring is equal to the effective bulk modulus times the area divided by the length as shown in Equation 4:

$$K = \beta_e \frac{A}{L} \quad \text{(Eq. 4)}$$

The units for K are force per unit length, or lb/in. The resonant frequency of a simple spring-mass system is the square root of the spring rate divided by the mass. Dividing this value by 2 pi gives the frequency in Hertz. This is shown in Equation 5:

$$F = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \quad \text{(Eq. 5)}$$

The hydraulic column in the low pressure bore of an intensifier is typically at least 500 times stiffer than the hydraulic column in the tubing that is connecting the intensifier to the servovalve. Because of this, the low frequency resonance of the intensifier system is determined mainly by the hydraulic column stiffness in the connecting tubing and test specimens.

In Equation 6 the relationship for springs in series (in parenthesis), is substituted for the K term in Equation 5:

$$F_h = \frac{1}{2\pi}\sqrt{\frac{1}{M_t}\left(\frac{1}{\frac{L_1}{\beta_{eLP}A_1}+\frac{L_{LP}}{\beta_{eLP}A_{LP}}+\frac{L_{HP}}{\beta_{eHP}A_{HP}}+\frac{L_2}{\beta_{eHP}A_2}+\frac{L_3}{\beta_{eHP}A_3}+\frac{L_4}{\beta_{eHP}A_4}}\right)} \quad \text{(Eq. 6)}$$

where:

F_h is the frequency in Hz

M_t is the total mass of the piston plus the mass of the hydraulic fluid (lb-s^2/in)

L_1 is the low pressure tubing or bore length from the servovalve to intensifier (in)

A_1 is area of the low pressure tube or bore from servovalve to intensifier (in^2)

β_{eLP} is the effective bulk modulus in the low pressure side of the intensifier (psi)

L_{LP} is the length of the intensifier low pressure chamber (in)

A_{LP} is the area of the low pressure end of the piston (in^2)

L_{HP} is the length of the intensifier high pressure chamber (in)

A_{HP} is the area of the high pressure end of the piston (in^2)

β_{eHP} is the effective bulk modulus of the high pressure fluid (psi)

L_2 is the bore (or tubing) length from the intensifier to the manifold (in)

A_2 is the bore (or tubing) area from the intensifier to manifold (in^2)

L_3 is the parallel bores (or tubing) lengths in the specimen manifold (in)

A_3 is the parallel bores (or tubing) areas in the specimen manifold (in^2)L_4 is the average length of the test specimens

A_4 is the area of the specimens (in^2)

The effective bulk modulus of the associated tubing will be higher in the high pressure side than in the low pressure side. All β_e terms are calculated using Equation 1 including the effects of fluid pressure, temperature, air, and component elasticity.

Manifold port branches and multiple specimens are hydraulic springs in parallel. In these cases, the quantity of internal cross sectional areas in parallel may be summed. In the following example, the L_3 and A_3 ports are 12 equal length parallel ports (see equidistant manifold in Figure 2). The specimen tubing length, L_4 can be the average lengths of the specimens and A_4 can be the sum of the internal cross-sectional areas.

An example with typical component sizes for computing the hydraulic natural frequency follows:

M_o = mass of hydraulic fluid = 195 in^3 * .03 = 7 lb

M_p = mass of intensifier piston and rod = 44 lb

M_t = total mass = (44 lb + 7 lb)/32(ft/s/s)/12(in/ft) = 0.13 lb-s^2/in

L_1 = 12 in (length of tube from SVV to low pressure side of intensifier)

A_1 = 1.0 in^2 (1.25 in O.D. tube from SVV to intensifier)

$\beta_{eLP,}$ = 100 000 psi (average effective bulk modulus of low pressure side)

L_{LP} = 1 in (length of low pressure intensifier bore)

A_{LP} = 38.5 in^2 (area of low pressure bore of intensifier)

L_{hHP} = 1 in (length of high pressure intensifier bore)

A_{HP} = 12.6 in^2 (area of high pressure bore of intensifier)

β_{eHP} = 160 000 (average effective bulk modulus of high pressure side)

L_2 = 4 in (length of bore from intensifier to manifold)

A_2 = 0.44 in^2 (0.75 in dia hole in manifold)

L_3 = 8 in (length of manifold port), quantity 12 in parallel

A_3 = 0.05 in^2 (0.25 in dia hole in manifold), quantity 12 in parallel

L_4 = 25 in (average length of S-tube specimens), quantity 12 in parallel

A_4 = .53 in^2 (-16 x 0.088 wall titanium tube specimens), quantity 12 in parallel

β_{eS} = 150 000 (effective bulk modulus of titanium S-tube specimens)

$$F_h = \frac{1}{2\pi}\sqrt{\frac{1}{.13}\left(\frac{1}{\frac{12}{100000 * 1.0} + \frac{1}{100000 * 38.5} + \frac{1}{160000 * 12.6} + \frac{4}{160000 * .44} + \frac{8}{160000 * .05 * 12} + \frac{25}{150000 * .53 * 12}}\right)}$$

F_h = 25 Hz

In this example, the resonating system which is composed of the mass of the intensifier piston plus the mass of the oil, is modeled as a single mass with six hydraulic column compliances in series. This is a simplification of the actual dynamics but will yield a good approximation.

The hydraulic natural frequency of a fluid column in a pressure line depends on the speed of sound through the fluid. One method of calculating this is shown in Equation 7.

$$F = \frac{c}{4L} \quad \text{(Eq. 7)}$$

where:

F is the hydraulic column resonance in Hz

L is the unobstructed length of the tubing (in)

c is the speed of sound in hydraulic fluid (in/s)

The speed of sound in fluids can be determined using the relation shown in Equation 8:

$$c = \sqrt{\frac{\beta_e}{\rho}} \quad \text{(Eq. 8)}$$

where:

β_e is the effective bulk modulus (psi) from Equation 1

ρ- fluid density ($lb\text{-}s^2/in^4$) a typical value for MIL-PRF-83282 is 0.000076

c - velocity of sound (in/s)

For a 48 in long tube this resonance is about 190 Hz for a typical 3000 psi mil oil system. This is well above the required wave form command signal frequencies and therefore is not typically problematic. There have been some cases where the tubing resonance coincided with the natural frequency of the first stage of the servovalve (around 400 to 600 Hz) and caused oscillations due to mechanical coupling. This is a rare problem but can be fixed by making the tubing run from two shorter runs of different sized tubing.

Since return lines typically operate at much lower pressures, and are long runs back to the pump reservoir, they often have low resonant frequencies which can coincide with the command signal frequencies. This can cause problems with control and create undesired large amplitude tubing oscillations in these lines. In the typical system described in this document a return line would be about 1.5 in diameter, operate at 5 psi and have a length of 48 in between the servovalve and the return line accumulator. The natural frequency of this line is calculated by Equations 5 and 6 to be around 11 Hz which is clearly in the range of excitation waveform frequencies.

To remedy this situation, a return accumulator can be installed. In addition, a return "back pressure" valve can be used to raise the pressure of the return line between the servovalve and return accumulator as shown in Figure 2. The return accumulator will absorb energy from the impulse hydraulics and the back pressure valve will raise the hydraulic natural frequency. If the return pressure is raised to 50 psi, the resonance will be 35 Hz using the calculation above. On some systems the back pressure valve and accumulator is also an effective strategy for reducing negative pressures excursions, after the "high pressure return to zero" portion of the impulse waveform.

All lines and passages should be designed to have un-blocked flow to return (reservoir), so that no trapped pressure remains after shutdown.

3.10 Servovalve

The servovalve should be sized for the peak flow requirement during the impulse rise time. This flow calculation should include factors determined from the high pressure fluid volume (including test specimens, high pressure manifold and intensifier volume), fluid type and temperature, percent of air in the fluid, test frequency, maximum targeted impulse pressure and rise rate.

The effective bulk modulus is calculated as described in Equation 1. The approximate peak flow required is calculated from Equation 9:

$$Q = \frac{\Delta P}{\Delta T}\frac{V_t}{\beta_e} \qquad \text{(Eq. 9)}$$

where:

Q is the peak flow requirement (in^3/s)

ΔP is the change in pressure (psi)

ΔT is the time interval during which the pressure change occurs

V_t is the total volume in this hydraulic section

β_e is the effective bulk modulus in this hydraulic section (psi)

Since the effective bulk modulus changes significantly with pressure it should be calculated at an average pressure value and evaluated more closely if the system design appears marginal. The effective bulk modulus is a non-linear function of temperature, pressure and other variables such as entrapped air.

With the proper accumulator design, the maximum flow through the servovalve(s) can be several times that of the maximum rated pump flow. The servovalve is the most restrictive device in the flow path and careful sizing will prevent it from limiting system output. The servovalve specifications for both flow and frequency response are important considerations for determining the pressure rise rate that can be achieved with the hydraulic system. Since nominal impulse rise times are often close to the servovalve step response time constant, frequency response is a significant design factor and should not be overlooked. If the servovalve is undersized, impulse rise rates and/or specimen load will be limited. The flow through the servovalve is non-linear with respect to the available pressure drop through the servovalve. The servovalve is very complicated but can be approximated with the orifice equation for a given valve opening. The servovalve is often modeled as a pair of orifices that operate in unison, between the supply pressure and pump return. These orifices are in series and typically have the ports of a double acting actuator or hydraulic motor connected to the output port of each orifice. For impulse testing it is only necessary to use one of the orifices. The second servovalve port is sometimes used to drive the center chamber of a three port intensifier. This is described in more detail at the end of this section.

To determine if the servovalve is correctly sized for a given test requirement Equation 9 may be used. If the servovalve is being used as a three way valve, only one of the two internal orifices is used, so the pressure drop rating can be used at half the pressure. So a servovalve that is rated at 60 gpm at 1000 psi drop, can assumed to be 60 gpm with a 500 psi drop through one orifice. This relationship is shown in Equation 10:

$$Q = K_{sv} A_o \sqrt{\Delta P} \qquad \text{(Eq. 10)}$$

where:

Q is the flow from supply pressure to the output port (in^3/s)

ΔP is the difference in pressure between supply pressure and the output port (psi)

A_o is the orifice area (in^2)

K_{sv} is a constant that varies from 70 to 100 depending on the valve, 100 is often used as a typical value. The actual value can usually be calculated from manufacturer's specifications.

The orifice area A_o varies from 0 to the maximum opening for the servovalve. The servovalve will open within a time interval which is on the same order as the time to reach maximum pressure in many of the impulse categories. Therefore, the maximum servovalve area will give a greater flow than what is actually observed in this short time interval. The pressure drop which drives the flow through the valve also decreases as the output port pressure increases, so this also creates a non-linear response which will be lower than an assumption based on maximum servovalve flow from the specifications.

Once all the constants are known, it requires a differential equation or simulation to accurately solve the flow problem. If the maximum area of the servovalve is known, it is possible to get a rough approximation by taking half the area and using a pressure drop which is mid way between the beginning and end pressure values.

Straight servovalve example:

AS603 waveform with 4500 psi peak using a two stage 60 gpm servovalve with a 5000 psi supply and 3000 psi pilot pressure.

From the manufacturer's specification for the servovalve, the area at maximum valve spool travel is 0.146 in^2 and the time to maximum travel is 0.032 s. K_{sv} is equal to 71 for this valve, by deduction from Equation 10.

The beginning pressure drop across the servovalve is 5000 and the end pressure drop is 500, so it is apparent that the flow will be non-linear with respect to time. The typical rise rate for this specification is 180 000 psi/s so the maximum time to reach the peak pressure will be about

4500(psi)/180 000 (psi/s) = 0.025 s

The servovalve will not reach the fully opened state in this time so the area at 0.025 s will be:

Area at time of peak = (0.025 (s)/0.032(s)) * 0.146 in^2 = 0.114 in^2

The typical valve dynamic is shown at the beginning of this document in Graph 1. This graph shows the valve opening (orifice area) plotted against time as well as the pressure rise trace. As shown in the graph the change in orifice area with time is not a linear function. For impulse rise rate time constants that are less than the servovalve opening time, the opening area can be "pro-rated" as is done in the above equation where the fully opened area is multiplied by ratio of rise time to valve opening time. If the rise time is more than twice the valve opening time, as shown in Graph 2, the calculation should be done in two parts so that half the orifice opening is used during the servovalve time constant and the second part uses the fully opened orifice area for the duration of the flow. This is because as shown in the graph if the rise time of the specification is less than twice the time required to open the valve fully, the valve will start closing again before it ever becomes fully open.

If the servovalve flow requirement at the rated pressure drop is 60 gpm during a time interval which is equal to the servovalve time constant, it is apparent from Graph 1 that a 60 gpm valve will not provide this flow soon enough. In many servovalve types, the spools for different flow ratings are the same diameter, but travel further to get the increased flow rating. For these valves it will not provide better response in this critical time interval to have a higher flow valve. This is an instance where two valves in parallel is an effective solution. With two valves in parallel, it is possible to get twice the flow during the critical time interval, because the valve area gradient is doubled.

Returning to the example, since the rise time is close to the valve time constant, using half the area yields a good approximation. We assume that our average area is half of the opening reached in the rise time interval or:

$0.114 \text{ in}^2 / 2 = 0.057 \text{ in}^2$

The average pressure drop is ((5000-50)+(5000-4500))/2 = 2725

The average flow will be

$Q = 71 * 0.056 * \sqrt{2725} = 208(\text{in}^3/\text{s}) = 54 \text{ gpm}$

In this example the servovalve is rated at 60 gpm with a 1000 psi pressure drop, or 500 psi drop per orifice (port). Since only one orifice is used and the pressure drop is much larger, the maximum flow will be higher than 60 gpm. The average flow in this example will be 54 gpm.

The total volume over 0.025 s will be approximately

$\Delta V = 209 (\text{in}^3/\text{s}) * 0.025(\text{s}) = 5.2 (\text{in}^3)$

If an intensifier is used, the above volume must be divided by the intensifier ratio.

ΔV is the incremental fluid volume that will be forced into the test specimens to raise the pressure.

If Equation 7 is rewritten to show the total test specimen and associate tubing volume that can be pressurized to the 4500 psi peak, the number of specimens that can be tested simultaneously with this servovalve can be determined.

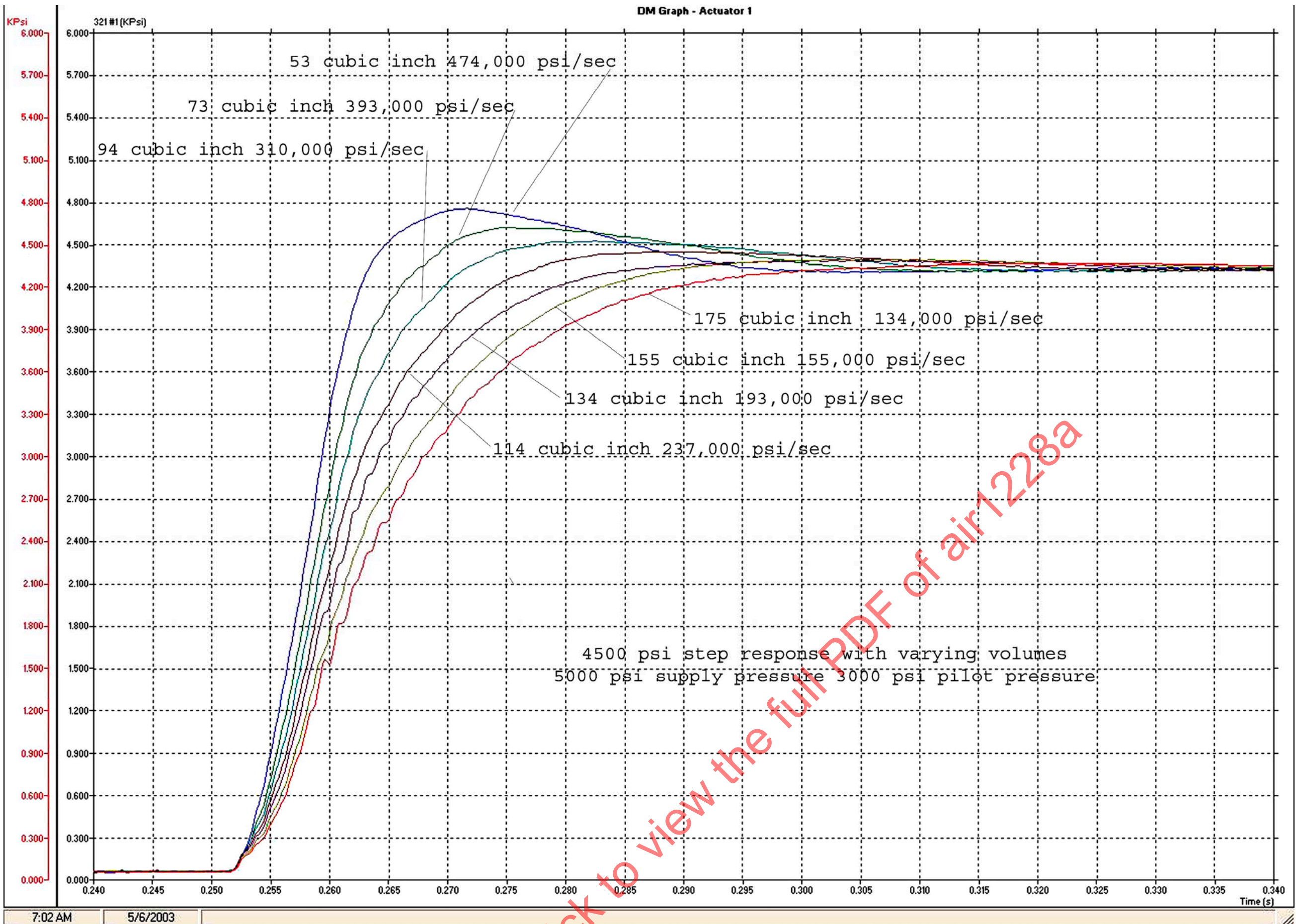
$$V_t = \frac{\Delta V}{P}\beta_e \qquad \text{(Eq. 11)}$$

$$V = (5.2 \text{ in}^3/4500 \text{ psi}) * 120\,000 \text{ psi} = 138 \text{ in}^3$$

The approximate calculation and assumptions stated above predict that this is the volume that can be pressurized to 4500 psi at a rise rate of 180 000 psi/s using one port of a 60 gpm servovalve with 5000 psi supply and a 3000 psi pilot pressure. There are many variables involved in different tests so this should be regarded as just a starting point to size the system. As Graph 3 shows, this rough approximation works out fairly closely to experimental values.

The plot sequence shown in Graph 3 provides experimental data for a system which is similar to the example used in this exercise.

As mentioned previously, it is important to do a more detailed analysis if the approximations are close to the operating margin. The approximations do not take into account many factors that will influence the flows and pressures, and the relationships are actually highly non-linear. A more detailed analysis is found in References [1] and [2].



GRAPH 3 - STEP RESPONSE FAMILY WITH CONSTANT PROGRAM AND VARYING TEST VOLUME

Servovalves can be used in parallel to achieve higher flow performance. This is often more practical than implementing a three stage valve solution or buying an expensive unknown valve that hasn't been evaluated. An efficient method for combining valves in parallel is to use a manifold block that allows two valves to be attached with port sides facing in a mirror image. In this arrangement the ports in the manifold block can be "through drilled" so that the valves can be attached with their pressure and return ports lined up. The output ports of the valves will correlate if the electrical signals to each valve are opposite. Using this manifold to match the pressure, return and control ports along with a means to reverse the spool direction of one valve (for example a reversing cable) simultaneous parallel operation is achieved with minimal effort. The dual servovalve configuration also provides a simple means of reducing system performance for small test sample volumes by merely unplugging one valve, which reduces the flow gain of the system by half. This allows the minimum gain margin to be reduced so that control of small volume tests can be achieved easier. The "parallel servovalve" method works best if the servovalve mechanical nulls are zeroed on a flow bench before installing on the test.

The servovalve may be used as a 3-way or 4-way device. In the straight servovalve system (without an intensifier), the servovalve is used as a 3-way device with one outlet port blocked. In an intensified system, the servovalve can be used as a 3-way or 4-way device. For a given size servovalve, 3-way operation will have more capacity because the intensifier center chamber flow is not restricted as it exits through the servovalve. Servovalves are typically sized with 500 psi pressure drop across each side of the valve for a total or 1000 psi drop at rated flow. In the 3-way configuration there is only 500 psi pressure drop at rated flow.

3.11 Intensifier

It is preferable to design a test system to run straight servovalve without an intensifier because it eliminates a significantly complicated component from the system. For example, all 3000 psi AS603 testing requires only 4500 psi peak pressure and can be accomplished using 5000 psi components without an intensifier. However, if an intensifier is required to meet the test pressure, care should be given to its sizing, design, and fabrication.

The intensifier is sized with consideration to both pressure and flow requirements. The maximum test pressure required and the maximum supply pressure available determine the required intensifier ratio. The ratio is selected based on the maximum test pressure required divided by the supply output pressure. To determine the output pressure of the supply system one must consider pressure accumulator droop, line loses, pressure drop though the servovalve, and the opposing pressure in the center chamber of the intensifier.

The intensifier is also sized with consideration to flow demand and seal speed. The flow demand calculation must consider the maximum test pressure, the maximum expected test specimen volume, total high pressure fluid volume (including all high pressure manifold intensifier volume), fluid type and temperature, percent of air in the fluid, test frequency, and maximum targeted impulse rise rate. This total high pressure requirement (calculated from effective bulk modulus) determines the flow output of the intensifier. Using the Equation 1 to calculate an effective bulk modulus at the higher pressure and then Equation 7 to calculate the flow requirement, it is possible to get a rough order of magnitude value for the intensifier sizing. The high pressure bore diameter is determined from the maximum industrial target seal speed of 20 inches per second. If the high pressure piston velocity is faster than this, premature seal or wear band failure is possible from excessive heat generation.

A standard high pressure piston bore size can be chosen to achieve the desired flow output and target seal speed. A standard low pressure bore size can then be chosen to achieve the desired pressure intensification ratio. The intensifier dimensions should be selected so that there are performance margins for both pressure and flow.

Total intensifier stroke should be kept to a minimum to reduce total high pressure volume (a 2 in stroke has been used successfully on systems shown in Table 1).

Intensifier piston position is a very helpful parameter for setting the initial high pressure piston position during operation and for diagnostics. One method of indicating the piston position is to add a small external rod (5/8 in diameter) on the head side (low pressure side) of the intensifier to give a visual indication of stroke. This position indicator rod will reduce the intensifier ratio slightly and needs to be considered when calculating the maximum output pressure.

There should be no piston snubbing on either end of the low pressure chamber of the intensifier. Free flowing ports in both chambers are required to prevent the corruption of the wave profile at the high and low pressure levels. It is important to verify this, because this snubbing capability is often standard for intensifiers used in other applications.

Intensifier longevity is achieved with consideration for seal speed (from the high pressure piston sizing) along with seal and wear band material, high pressure piston and low pressure bore coatings, and heat and contamination removal. Recommendations for these components include graphite impregnated Teflon seals, non-metallic piston wear band, and HVOF (high velocity oxyfuel) tungsten coated rods and barrels super-finished to 2 to 4 micro inches. Wear contamination and heat can be removed with bleed ports opposite the inlet ports on the center and bottom intensifier chambers. A simple needle valve on each of the bleed outlets can be set to exchange the fluid volumes a few times a minute. The bled fluid is then recirculated to the HPS for cooling and filtration.

Spring loaded U-cup seals have been used with good longevity for high pressure dynamic seals. A good rule of thumb for seal retention is “the fewer the pieces in the assembly, the fewer parts to fail over time.” For example, a rod bushing assembly requires additional O-rings and backups to seal the bushing. If a rod bushing is unavoidable, minimize or eliminate the exposed pressure area and size it for any exposed pressure. If O-rings are incorporated into the design, chamfer grooves are not recommended for O-ring glands. All seal glands must be fabricated to recommended sizes (or tighter) and surface finishes must be maintained.

3.12 Test Specimen Manifold

When possible, the test specimen manifold should be directly attached to the high pressure outlet port of the intensifier to minimize fluid volume and to minimize the number of sealed connections. Minimizing fluid volumes will also minimize test system hydraulic power demand. Fewer high pressure connections will result in fewer failed connections over time. The manifold should be designed to minimize total passage volume yet maintain adequate port diameter to minimize flow restrictions to the test specimens. Excessive port flow restriction can create undesirable peak pressure anomalies. When long passages are required and the diameter must be large to allow for drilling, rods can be secured in the passages to reduce volume. The passages feeding individual test specimens should be symmetrical from the pressure source so that test results are not biased by passage asymmetry (see equidistant manifold in Figure 2).

The manifold design and fabrication should eliminate stress concentrations due to acute passage intersection angles, weld plugs, and sharp edges at passage intersections. The manifold material strength should be chosen for the required pressure range and desired fatigue life. A higher strength material may be required for AS4265 testing than is used with AS603 testing of the same test articles. The manifold material should have corrosion resistance for environmental applications.

3.13 Test Specimens

Test specimens should always be attached to the test manifold in such a way that allows the air to be bled from the high pressure side of the system. Specimen end fittings should be chosen that allow the system to be readily bled. S-tube and bent hose test articles require support of the free end of the test specimens. S-tube and hose supports must not introduce any abnormal stress into the test specimens. S-tube retention must be rigid enough to prevent any “Bordon tube” effect which will over-stress the specimen. AS4265 testing of straight tubes requires an internally sealed center support per AS4298 to minimize fluid volume and to prevent fluid column buckling of the tube (see Reference [5]).

3.14 Manifold Port Configuration (static seals)

There are a variety of port seal configurations that can be used on manifolds throughout the system, each with their own pressure and temperature limitations. In descending order of performance and durability are face seals, metal-to-metal connections (manufactured by HIP/Autoclave), AS5551 ring locked boss, SAE J1926-1 threaded port, and lastly AS5202 threaded port. O-rings in a face seal gland or annular arrangement out-perform O-rings in a chamfered gland (SAE J1926-1 and AS5202). However, the steeper SAE J1926-1 chamfer performs better than the shallow AS5202 chamfer. The performance of those ports with elastomeric seals can be improved with 90 durometer urethane O-rings in most instances, (however there are temperature limitations). In the most demanding location between the intensifier and test manifold, face seals have proven to be a durable design, particularly when used with 90 durometer urethane O-rings.

Test specimens with treaded ends require some form of adaptation to the test manifold. Since the threaded connection can wear out, it is advantageous to design adapters that can be removed and replaced. With modular specimen adapters, they can be replaced as they wear, as they are damaged, or they can be changed out for different thread sizes.

3.15 Spill Containment

Well designed failsafes and spill containment will minimize release of the test fluid to the environment and reduce cleanup requirements. The HPS should be equipped with a "low reservoir level shut off" and the test controller should have error monitoring to detect a drop in pressure peaks.

In addition to these two failsafe modes, two physical barriers of containment are recommended, one surrounding the test articles and one enclosing the test system. While on rare occasion they puncture, clear plastic bags have been used successfully to contain spray from failed test specimens. Polyethylene bags (up to 55 gallon trash can size) have been used for ambient testing with mil oil. Nylon oven basting bags have been used successfully with Skydrol and at elevated temperatures. The bag is tucked into a catch pan located below the test specimen manifold. The catch pan can be equipped with a small sump and a float switch to activate the failsafe system. This system is very effective when running a straight servovalve test system, where there is no limit to the amount of fluid that can be pumped out of the failed test article (a system with an intensifier has a limited volume that can be pumped out).

The second level of containment is designed to collect the fluid if the primary test article containment system fails or if a leak occurs in the rest of the test system. A containment pan or multiple containment pans should be located below the HPS and the remainder of the control hydraulics. A test enclosure or environmental chamber can be used for secondary containment.

3.16 Test Enclosure and Temperature Chamber

A test enclosure should be designed to safely contain the occasional blown fitting along with sprayed fluid from a test system failure. A temperature chamber can provide this safety containment in addition to high and low temperature environments. The high temperature test enclosure should provide a means of evacuating the smoke and vapors from the chamber in the event of a failure. When the failsafe system is triggered from a hydraulic leak, it should be designed to return the environmental chamber temperature to ambient and activate the chamber exhaust system. Exhausting to the atmosphere requires filtration to remove the pollutants.

There are several other considerations when implementing a temperature chamber. The test enclosure or temperature chamber should provide adequate viewing of the test specimens. The hot hydraulic fluid in the test manifold must be separated from the intensifier seals. This thermal isolation helps to prevent premature intensifier high pressure seal failure. One method of achieving this is to use a heat exchanger to cool the fluid as it surges between the intensifier and the test specimens. Typically some physical separation is required between the test manifold and the intensifier to provide space for the fluid cooler. It should be noted that the longer length required for the fluid cooler will compromise the minimal line length goal designed to reduce interference from test system resonance.

Pressure transducers usually have temperature limitations or ranges, beyond which their accuracy is reduced. Typical impulse test sequences (for example AS603) have hot, cold, and ambient sections and the pressure transducer calibration must be accurate in all conditions. A simple solution is to run a sensing (small tube) line from the test manifold to a pressure transducer located outside the temperature chamber. The pressure transducer should have a 1/4 in diameter tube running outside the temperature chamber to isolate the transducer from the test temperature ranges. Although sensor tubing lines may be necessary to reduce temperature effects they also add lag to the sensor signal and shouldn't be too long. Two to four feet has been found to be adequate.

4. DETAILED DESCRIPTION OF INSTRUMENTATION AND CONTROL

4.1 Instrumentation and Control Overview

This section will address general requirements as well as some specific considerations for the instrumentation and control section of the impulse system. Figure 3 shows a generalized block diagram for the instrumentation and control system. Two pressure transducers are shown for measuring the pressure in the test specimens. Although only one pressure sensor is required for operation, two sensors which can be compared during the test can help eliminate invalid cycles. The sensors can have integral excitation and amplification or distributed components which are interconnected.

In the arrangement shown in Figure 3, there are capabilities to acquire data, provide a user interface, provide test control and scenario management, hydraulic pump control and closed loop control for the servovalve. In this arrangement the functional entities are shown in separate blocks with some grouped together. In practice, they can all be separate, or can all be grouped into one system. The system may be entirely comprised of analog electronic circuitry or it can be mostly digital electronics with software equivalents of the analog electronic functional blocks.

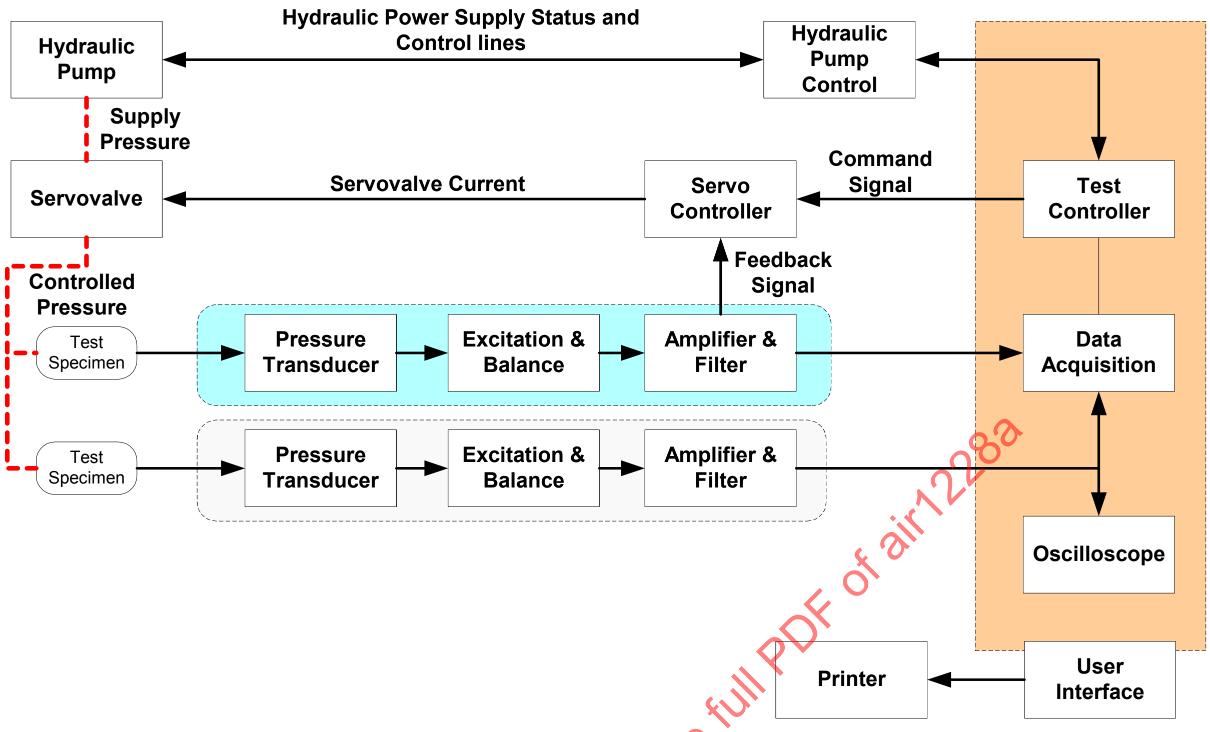


FIGURE 3 - INSTRUMENTATION AND CONTROL DETAILS

4.2 Servo-controller

4.2.1 General Considerations

For pressure control, the servo-controller requirements are not complicated. Usually proportional gain is all that is necessary. Error Integration, derivative gain, lead-lag, or other compensations are rarely required. The controller merely subtracts the feedback signal from the command signal, amplifies the result and sends either a current or voltage signal to the servovalve. Though a controller of this simplicity can be depicted with just a summing junction and a current driver, there are some additional parameters and controls that are required for functionality. Figure 4 is a basic analog electronic servo-controller diagram showing the fundamental elements necessary for pressure control.

In general, a servo-controller will have additional components for functionality and circuit safety issues which are not shown in the diagram of Figure 4. Printed circuit boards and assemblies are available inexpensively which provide all of these features. Control systems and control system terminology are described in detail in Reference 3.

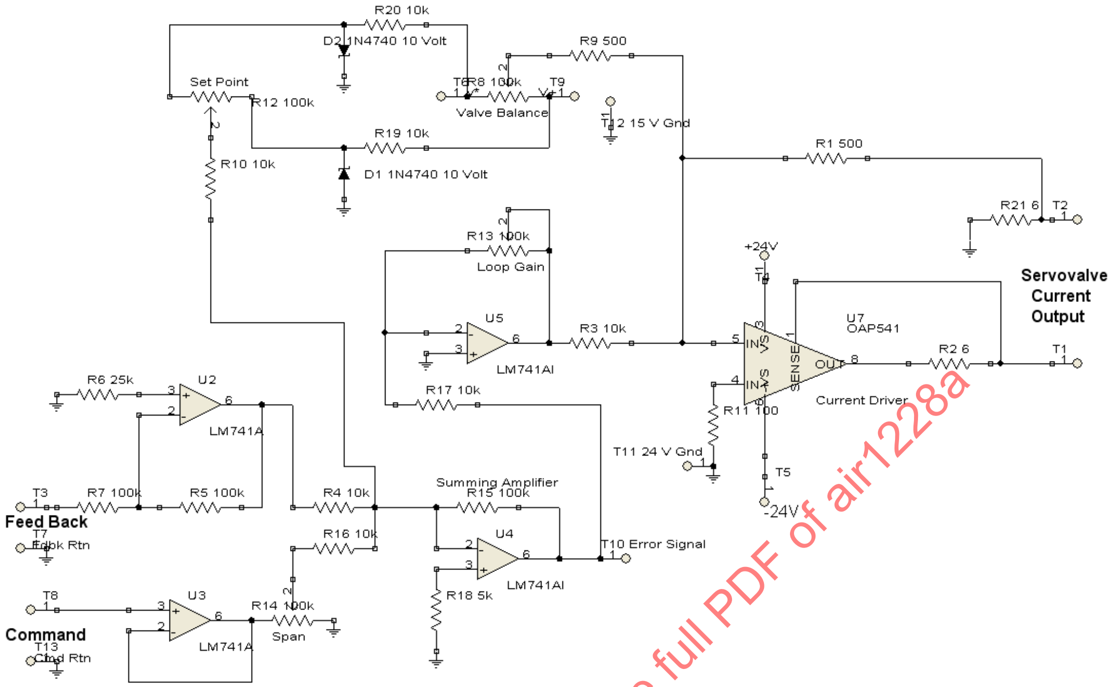


FIGURE 4 - BASIC ANALOG ELECTRONIC SERVO-CONTROLLER DIAGRAM

4.2.2 Bandwidth

The servo-controller may be analog or digital. In either case, it is necessary that the controller have sufficient bandwidth to achieve and control the rise rates and frequencies required for impulse testing. Digital controllers must operate with “real time” operating systems or executables. Non-real time operating systems may have intermittent latencies greater than 10 ms. This can cause transients in pressure control which might trigger failsafe circuitry or damage test specimens. The minimum bandwidth which is recommended for digital and analog controllers is 1000 Hz. This will provide a relatively flat response out to 500 Hz and ensure that the controller does not add more than a milli-second of lag to the loop response. This is very easy to achieve with analog controllers. The less expensive digital controllers often have extremely low bandwidths, especially if they are targeted for temperature or process control.

If the controller is part of a digital system and its bandwidth determines the acquisition rate of the data system, this will also affect the accuracy of the measurements. As an example, a sample rate of 2000 samples/second will provide a 0.1% measurement resolution for reading the magnitude of sine waves up to 42 Hz. This sample-rate will provide a 2% resolution for calculating rise rate in an AS603 impulse waveform with a 4500 psi peak at a nominal 180 000 psi/s.

4.2.3 Controller Adjustments

The controller should have a minimum of three adjustments, span, set-point and gain.

Span adjusts the amplitude of the command signal, set-point controls the command signal offset from zero and the gain adjustment refers to the amplitude setting of the proportional gain.

A separate adjustment for valve balance makes operation more logical as it allows the set-point to be correlated with the actual pressure feedback signal. It is important to have a numeric indication of proportional gain setting, set-point and command signal span so that settings may be re-established for similar test setups and after performing maintenance tasks.

4.2.4 Advanced Control Strategies

Although usually not required, it may be beneficial to employ all aspects of a PIDF controller for unusual conditions. Error integration, adaptive control and/or gain scheduling can also be utilized, for example, in unattended operation of tests where the fluid and test articles undergo significant temperature changes.

4.2.5 Failsafe

When the test controller is comprised of a computer with an application targeted for hydraulic pressure testing (see Reference [4]), the failsafe functionality can include evaluation of several parameters. The evaluation of these parameters helps ensure test repeatability and validity. The parameters which can be evaluated by the computer application may for instance include the comparison between two pressure transducers, peak and low pressure tolerance bands, rise rates, cycle count and plateau pressures. In addition it is frequently desirable to monitor the hydraulic pressure supply system, so that the command may be disabled if the pump fails.

If the test controller is not comprised of a computer application with failsafe functionality, the controller should have pressure limit failsafe detection. Error failsafe is not considered as important since there is frequently a large phase lag between command signal and response.

If the parameters are evaluated automatically during the test to ensure that they are within a tolerance band, the system can be designed so that only valid cycles are counted and the test is stopped if any limits are exceeded. Choosing the tolerance band to be smaller than the specification range, in proportion to the uncertainty of measurement will ensure that the test meets the specification.

4.3 Pressure Transducers, Amplifiers and Excitation

It is preferable to have two pressure transducers on the test so that the readings can be compared. Although both transducers might fail, it is not likely that they will both fail at the same time and manner. Either manual or automatic comparison of the transducers readings will help ensure that the readings are accurate. The pressure transducers must be fatigue rated. Some transducers are only rated for a few hundred thousand cycles. Even fatigue rated transducers will eventually fail. It is not unusual for a diaphragm type transducer, with a strain gage bridge, to develop as much as a 25% offset during an impulse test. Some inexpensive transducers and amplifiers will have poor common mode rejection and have excessive noise on the output signal. High impedance signal returns and inadequate shielding will also add noise to the signal. It is much better to correct the sources of the induced noise, than to attempt to filter it out. Heavy filtering will add a lag to the pressure control loop and create problems with controllability.

4.3.1 Pressure Transducers with Integrated Excitation and Amplification

Advantages to using pressure transducers with integrated electronics:

a. It is possible to design the transducer for longer fatigue life with the amplifier integrated with the pressure sensor.

b. With integrated electronics in the pressure transducer, the susceptibility to common mode problems from EMI can be reduced.

c. The overall instrumentation system size and cost is reduced.

d. The transducers and amplifiers are certified as a unit so that calibration costs are reduced.

Disadvantages to having the excitation and amplification integral with the transducer:

a. Most electronics have a limited temperature range in which they can maintain their accuracy tolerance. This limited temperature range is often much narrower than the temperature range of testing, prohibiting accuracy or even operation at the test temperature extremes.

b. Test temperature ranges which exceed the transducer electronics accuracy range require placing the transducer outside the environmental chamber. The transducer is then connected to the specimen manifold with a long tube. As mentioned in the hydraulics section, the pressure lag created by the long tube affects the controllability of the pressure waveform.

c. Transducers with integrated electronics are typically less accurate than a separate transducer with high quality instrumentation amplifier and signal conditioning.

d. Signal filtering, if available, is usually not adjustable in transducers with integral amplifiers.

e. The transducer with integrated amplifier may be subject to excessive vibration in impulse testing so the vibration rating of the sensor should be evaluated.

f. Transducers with integrated electronics are usually larger and heavier which limits the locations where they may be installed.

4.3.2 External Transducer Excitation

Pressure transducers capable of measuring static pressures that do not have integral amplifiers usually require an external voltage input to create a measurement. These pressure transducers typically utilize the Whetstone bridge strain gage configurations. The "bridge" type transducers require excitation that is regulated and accurate. Noise on the excitation lines will show up in the signal lines and might contribute to common mode offset. If the excitation voltage level drifts after the calibration has been completed, the readings may be outside the accuracy tolerance. Separate wires to carry the voltage reference signals from the transducer back to the excitation supply should be used to compensate for the voltage drop along the cable. When computing the uncertainty of measurement, the variations in the excitation voltage are an important part of the computation. High end signal conditioning uses separate power supplies for each channel. This separate supply enables the use of Wagner grounds and provides better channel to channel isolation.

4.3.3 External Transducer Amplifiers

High-end external transducer amplifiers are usually designed to be very stable and accurate. These amplifiers typically have differential high impedance inputs, outstanding common mode immunity, low noise, and excellent gain accuracy and stability. In addition, most are equipped with selectable filter frequencies and multiple buffered outputs. The accuracy specifications of many of these amplifiers are valid from DC to frequencies in the tens of kilohertz. These amplifiers are relatively expensive and are usually used in a chassis arrangement which provides power, cooling and connectivity. Periodic calibration of all parameters that affect the accuracy of the amplifiers should be performed to ensure that they are certified.

4.4 Analog/Digital (A/D) Converter and Multi-function I/O Cards

In digital servo-controllers, computer based test controllers and acquisition systems, an analog to digital converter subsystem is employed to convert the analog pressure signals to a digital form. Over the years these cards and systems have become less expensive and more accurate. Inexpensive components and subsystems, however, may have inadequate bandwidth and resolution to meet the requirements of the test specifications. A product which provides 2000 samples per second per channel with 16 bit resolution is more than adequate, assuming that all the remaining accuracy tolerances meet the testing requirements. It is possible to acquire a device with an A/D that is very accurate, but does not provide the measurement accuracy stated in the test specifications. For instance, in a 4500 psi peak AS603 test, the resolution of an 8 bit A/D is not adequate to determine if the pressure signal is within specification. Likewise a sample rate of 50 samples per second would provide a measurement which has more than 10% error for the peak, and would not be able to determine the rise rate at all. With analog to digital converters it is usually necessary to have anti-aliasing filtering (see filtering).