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Signal reproduction fidelity of servohydraulic testing equipment

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ABSTRACT: Servohydraulic systems are often used in pseudo-dynamic, structural, and shaking table systems to simulate earthquakes, test the seismic resistance of structures and components, and to verify analytical models. High fidelity reproduction of the excitation signal is imperative to ensure that unknown or unwanted experimental errors are not introduced into the test. Knowledge of the potential sources of system wave form infidelity will greatly enhance the researcher's ability to ensure the best test results through intelligent system design. This paper provides a technical discussion of various sources and effects of wave form distortion and how to reduce them in servohydraulic systems.

1 INTRODUCTION

Distortion analysis is a complex issue when using multiple degree-of-freedom, closed-loop electrohydraulic servo systems. There are many potential sources of distortion, and many of them are highly interdependent on each other and on various system parameters. In most cases, it is difficult to assign specific values to these sources of distortion, let alone their potential effects on test results.

The best approach to system design and specification is to eliminate and reduce these sources of distortion and to size the system so that the performance envelope lies outside areas of high distortion. When considering distortion, compromises must often be made between overall system performance and cost.

The following sections describe measurement, sources, and types of distortion. In the final section, a typical performance envelope for a medium to large seismic system is presented. The areas affected by each type of distortion are shown.

2 DISTORTION MEASUREMENT

2.1 *Distortion analysis*

There are many types of distortion and distortion measurement methods. In seismic simulation systems, one of the principal areas of concern is the frequency response of the system. For high fidelity it is important that the system response has the same frequency content as the program signal . With other types of systems, parameters such as phase lag, system roll off, peak values, energy content, resolution, etc., may be deemed to be more important system parameters.

The most common method of distortion measurement is to measure the total harmonic distortion (THD) of the <u>acceleration</u> signal. A representative group of pure fundamental frequencies (which span the frequency envelope of the system) is tested, one frequency at a time usually at two different programming levels. For each frequency tested the various harmonic components excited in the system are measured with respect to the magnitude of the fundamental program signal, Figure 1. The ratio of these components to the fundamental frequency is the total harmonic distortion as defined by Equation 1.

$$\mathrm{THD} = \frac{\sqrt{\sum (\text{harmonics})^2}}{\text{fundamental}} \qquad (1)$$

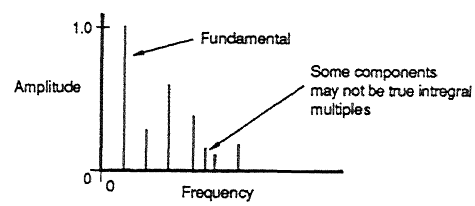

Figure 1. Frequency domain plot.

When a digital computer/analyzer is used, a fast Fourier transform can be done on the response signal. Only the significant components, typically the first three to ten, are counted.

Typical acceleration distortion values for a seismic system will be approximately 20% at 1/3 the oil column frequency, dropping off to 5% at frequencies at or above oil column. Depending on the system's mechanical resonances, this percentage will increase again at higher frequencies.

2.2 Transducers

When measuring distortion attention should be paid to the type of transducers used, their operating characteristics, mounting locations and the parameters they will measure. For example, the arcing motions of actuators, bellcranks, and linkages on some systems can introduce distortion which may not be apparent from the transducer readout.

The useful frequency range, transverse sensitivity, and the alignment of the accelerometers and mounting fixtures are important. Improper use and mounting of accelerometers can contribute significantly to distortion either directly or by contributing to control instabilities. This becomes more evident at higher frequencies.

On large seismic tables the location of the accelerometers is very important. Accelerometers should be mounted where they can measure the desired control parameter but do not pick-up local structural resonances or table deflections that would make control more difficult. Since tables cannot be perfectly rigid, it is important to realize that there will be differences and phase lags between local accelerations throughout the table.

3 HYDRAULIC CONSIDERATIONS

3.1 Servovalve Characteristics

The most important hydraulic component in terms of distortion and overall system performance is the servovalve, Figure 2. This section gives a brief overview of servovalve operation characteristics for an inertially loaded system such as a seismic system. For a more detailed analysis see Clark's paper (Clark 1983).

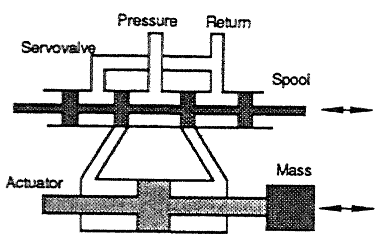

Figure 2. Functional schematic of an inertially loaded servovalve / actuator.

The servovalve is a variable orifice which, in conjunction with the external load (via the actuator piston area) and supply pressure, determines the amount of oil flow through the servovalve. If there is no external load then full pressure is available across the servovalve and maximum flow (open loop velocity) will be developed. If the actuator piston is constrained, then there will be no flow and the pressure drop across the servovalve will be zero allowing for full pressure to be developed across the piston.

Figure 3 shows a range of servovalve spool openings plotted as a function of force (piston area x pressure) and velocity (flow / piston area). The ellipse represents the plot of a simple sinusoidal inertial load (maximum force 90° out of phase with maximum velocity). Any number of ellipses are permissible provided they lie within the spool opening for this model. The ellipse shown is the worst case sizing condition which occurs at the intersection of the acceleration and velocity limits.

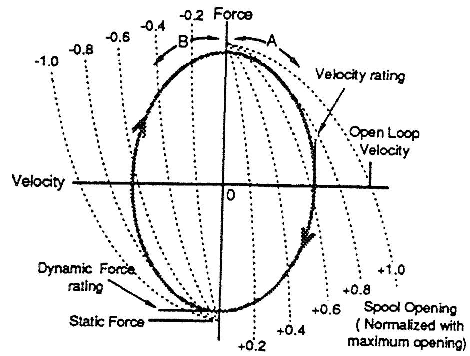

Figure 3. Force / Velocity phase plane for an inertially loaded servovalve / actuator.

It is important to notice that the servovalve limits the performance only in the first and third quadrants, and that this limitation occurs just past the peak force point in the cycle. Any hydraulic constraints will affect this part of the operating cycle first.

There is a severe non-linearity in the spool opening as the system passes through peak force. During the same interval of the cycle the spool must move much faster during A than during B. This point is often referred to as the 'turnaround' point. The discontinuity at this point can be very large and is often referred to as 'pressure switching'.

Field experience has shown that to minimize this servovalve non-linearity, and to allow for some design margin for the control system, the 'load' ellipse should be within the 80% spool openings and the peak force is limited to 95% of the available maximum. These margins are based on experience and are frequency dependent; at lower frequencies, they can be decreased.

On system performance envelopes this ellipse represents the intersection of the velocity limit and the acceleration limit. This is the most difficult point to attain and determines the sizing of most of the system components.

At frequencies higher than the oil column the same 'pressure switching' phenomena can occur at the intersection of the compressibility flow limit and acceleration limit. At these higher frequencies the distortion is dominated by mechanical resonant frequencies and 'pressure switching' is not as noticeable.

3.2 Servovalve Spool Lap

Researchers have spent much effort analyzing the effect of spool lap and metering edge shapes on signal fidelity and cavitation (Royle 1958).

A small amount of valve overlap is desired in standard grade servovalves to minimize internal leakage within the valve and to reduce the need for tight tolerances. The problem with this type of valve is that there is a dead band where a slight motion of the spool results in little change in flow. This results in a slight indent in the acceleration curve and cause distortion.

This problem has been eliminated in single servovalve applications by the careful grinding of the spool valves using the actual measured flow characteristics of the valve to set the amount of grinding.

Studies on the potential benefits of servovalve under lap show that there is no benefit and that a definite increase in distortion exists at low frequencies and high load conditions due to the increased internal leakage.

3.3 Oil Column

A system's oil column frequency is an important system parameter because it directly affects distortion and upper frequency performance. Distortion arises from the natural resonance of specimen/table mass on the oil column spring of the actuators. Typically, on most inertially loaded systems the natural frequency of the oil column is well within the operating range of the system.

Operation at the oil column frequency may result in problems with gain, but the distortion effects may be minimal. However, operation at frequencies of approximately 1/3 of the oil column will result in frequency components that can excite the oil column frequency and cause significant distortion. These frequency components are mainly due to the non-linearity of the servovalves.

Figure 4 shows a wave form with a third harmonic component.

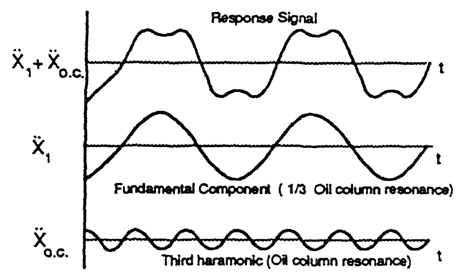

Figure 4. Third harmonic distortion due to oil column resonance.

In general, if the oil column frequency is fairly constant, this type of distortion can be minimized by careful tuning, use of valve linearization circuits, and/or computer compensation.

The problem arises when the oil column frequency changes due to changes in the effective mass of the specimen. This occurs if the specimen mass or the location of the specimen on the table changes. The oil column frequency might then be moved into an area where there is slightly greater gain resulting in increased distortion. Fortunately, in most cases the variance of typical test specimens is not that great.

This type of distortion primarily occurs at specific frequencies and tends to be greater at larger program values, although it can occur at lower values. It is one of the dominant distortion components.

3.4 Servovalve linearization

Servovalves are inherently non-linear devices in which the flow output is proportional to the square root of the pressure drop across them. Servovalve linearization is a method using analog electronics to increase the servovalve signal at higher levels to help linearize the flow characteristics of the servovalve. This has been shown (using analog computer simulation studies) to be of great benefit in regard to distortion.

Recent field experience has verified these simulation studies and resulted in a distortion reduction of one half at system frequencies at 1/3 oil column and under high load conditions.

This type of compensation technique is useful when iterative computer compensation techniques are inappropriate.

3.5 Flow limits

Distortion occurs whenever there is limit to the supply of hydraulic oil going to the servovalve. In an extreme case, where there is an absolute flow limit and the acceleration loads are relatively small, the displacement wave form will 'triangulate', with the slope of the wave form equal to the velocity limit, see Figure 5. Flow limits can arise from several sources; pump, servovalves, or accumulators.

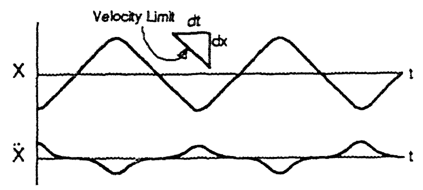

Figure 5. Triangulation of displacement wave form due to flow limits.

Flow limits cause increased distortion predominantly at the velocity limits of the performance curve. The solution is to size the system conservatively and to be aware of any system parameters which may limit peak flow or the available pressure drop across the servovalve at peak velocities.

3.6 Force limits

A system can become force limited in two ways. There may be insufficient force due to compressibility flow limits, or excessive load demands due to system or specimen resonant frequencies (See 5.4).

At higher frequencies, the compressibility flow requirements (flow required to compress the trapped oil volume within the actuator) becomes significant. Compressibility flow is proportional to frequency and the stroke capacity of the actuator. This flow requirement, in combination with the servovalve roll-off, limits the force capacity of actuators at frequencies at the high end of the performance envelope.

This compressibility flow limit in itself will not cause distortion, but will limit the systems capability to react to perturbations such as those due to cross coupling from other axes. The solution is to use higher response servovalves or multiple valves to get the required upper frequency performance.

3.7 Hydraulic piping

The ideal supply for a servovalve system would be an infinite volume, constant pressure source with zero internal impedance using a fluid which is incompressible, and has zero viscosity but good lubricity properties, infinite heat capacity (no cooling required), is homogenous (perfectly clean), and has low density (some mass is required for a servovalve to work) .

In reality, this fluid is not available. However, with careful sizing of components and good system design, the more important limitations can be met. The main issues (in regard to distortion) are pressure loss in the piping distribution, inertial pressure spikes, cross talk through the hydraulic system, oil cleanliness and dissolved gases.

Flow losses are minimized by conservative sizing of the hydraulic lines and appropriate use of accumulators. Separate pilot supplies are used to limit the fluctuations to the servovalves.

Inertial pressure spikes occur at high operating frequencies due to the inertia of the fluid itself and the increased compressibility flow requirements. These spikes can cause noise and cross-talk problems resulting in increased distortion. Appropriate line sizes and the use of close-coupled accumulators can largely eliminate these problems.

Spool friction is due primarily to silting, a phenomenon in which particles within the oil tend to accumulate in front of the spool edge, resulting in a large value of static friction. This leads to resolution problems and increased distortion at low programming levels. Clean oil and the use of dither will alleviate this.

Servovalve 'rumble' and other hydraulic noise are caused by dissolved gases coming out of solution downstream from the metering orifices within the servovalve, effecting return and drain line operating pressures. This sets a lower limit for background noise level.

4 MECHANICAL CONSIDERATIONS

4.1 Friction

Friction can arise from a number of sources within a testing system. The actuators contain a number of friction sources: seals on the actuator, rod bearings and swivels. In addition, the table may contain a number of sources (depending on the design and layout). Slide bearings, static supports and mechanical linkages are a few potential table friction sources.

Frictional loads are not large and typically account for only a few percentage points of maximum system capability. The problem occurs during turnaround where there is a reversal in the direction of motion. The oil pressure must overcome this static friction before motion can begin. This causes a discontinuity in the motion. In addition, due to the compliance within the load path, a certain amount of energy is stored within the load train. Once the system starts moving, any difference between the static and dynamic friction values results in a portion of this energy being released instantaneously. This tends to excite any mechanical resonant frequencies until the energy has been dissipated, as may be seen in Figure 6.

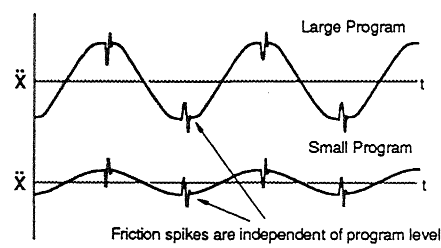

Figure 6. Effect of friction on acceleration waveform at high and low program levels.

At peak force (acceleration) operating values, the noise or distortion from friction is small. However, the magnitude of this disturbance is constant, so that as the program signal level is reduced it becomes an objectionable component. At low programming levels it can result in acceleration spikes much greater than the program signal.

Due to the high frequency content and short time duration of each pulse, friction has little effect on the THD value. Its effect on the wave form shape nevertheless can be large.

In general, this type of distortion is a percentage of maximum system capability, so that it is critical that the system is not oversized if good low level performance is required. An example of where this is frequently a problem is in the vertical axis of seismic systems. Often the vertical actuators are sized by the overturning moment (OTM) requirement . The actual vertical axis requirements may be much less, but the actuators used will be sized greater due to the OTM. This results in greater friction (as compared to a system where smaller vertical actuators were supplied).

Friction from the rod bearings may be significant enough to justify the use of hydrostatic bearings in some cases. The use of these bearings increases the cost, complexity, and pump requirements of the system. Hydrostatic bearings require longer actuators, which in some cases can make foundation design more difficult and expensive. They can also adversely affect the lateral natural frequency of the actuator due to the increased mass and length of this type of actuator.

4.2 Mechanical resonant frequencies

Within any mechanical system there are numerous resonant frequencies which can be excited. A good seismic system design philosophy is to keep these resonant frequencies well beyond (2 to 3 times) the maximum operating frequency. There is less chance that they will become excited and frequently the system gain is low enough at these higher frequencies that they will not be picked up and be amplified. (These resonant frequencies usually occur in modes where the system could not control them even if it had the power to do so.) In practice there are always some components of these resonant frequencies at the upper operating frequencies of the system. Figure 7 shows a signal with a higher frequency component due to a mechanical resonance.

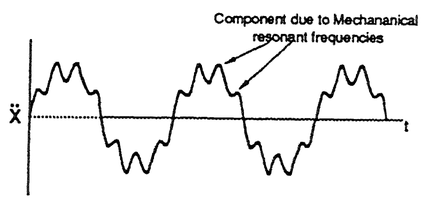

Figure 7. Distortion due to a mechanical resonant frequency

The main table modes which can cause problems are the 'warp' and 'oil canning' modes. Careful design is required to keep these modes as high as possible. The specimen configuration and mounting can aggravate these modes - especially the oil can. A change in the mounting method can extend the frequency testing range.

Local structural panel frequencies from within the table are another source of potential disturbance. The panel resonant frequencies are quite high (relative to the operating range) and are more of an acoustic nuisance. Occasionally they can interact with the servovalves and add significant high frequency components to the feedback signal.

4.3 Foundation interface

A significant source of distortion can arise from the foundation and the actuator attachment to the foundation. Attention to the foundation's design is required to avoid vibration modes which will affect the system's operation.

At higher frequencies, the compliance at the actuator attachment can prevent the actuator from inputting full force capabilities into the table. This creates a force limit at higher frequencies, limiting the response of the actuators.

With floating (isolated) foundations an appreciable relative foundation movement is possible, typically peaking at about 1 Hz. To measure true displacement relative to the 'fixed ground', additional transducers must be mounted between the fixed and floating foundations. In practice with the use of large foundation/specimen mass ratios and adequate suspension system damping, the small increase in accuracy does not justify the cost and complexity.

4.4 Backlash

Backlash is similar to friction in that in itself it does not cause serious distortion problems. It does, however, provide a mechanism by which a force or energy impulse can set off other resonant frequencies in the system. The two main sources of backlash are 1) bolted connections within the load train and, 2) the actuator swivels. Proper design and maintenance of these mechanical components are required for system fidelity.

5 MISCELLANEOUS

5.1 Electrical noise

Noise within circuit boards has been greatly reduced in recent years by improved components. The long transducer cables used on seismic systems offer an opportunity for noise pickup in the feedback loop. Through careful grounding design this can be attenuated to low levels.

Line noise in the operating environment and from within the consoles can be a major problem. Line noise typically is in the operating range of most seismic systems, making it difficult to remove or filter.

Careful attention must be paid to grounds used both for the console and pumping system. A separate instrument ground is a necessity.

5.2 Cross coupling

Cross coupling occurs whenever actuators in one axis must react loads due to motions in another axis. The most obvious is the Over Turning Moment (OTM) loads imposed on the vertical actuators due to horizontal motion. The moment arises since the horizontal actuators do not act through the center of mass of the table/specimen.

The transducers on the table must sense these unwanted motions in order for the actuators to respond.

The relative program signals in these axes may be so low so that any components from another axis will appear to be quite significant.

Control methods using feed forward compensation can help to attenuate this cross talk from other axes. Computer compensation using iterative methods can help greatly assuming the dynamic properties of the specimen table do not change too much during a test.

This is one of the more difficult areas of distortion control and new control techniques are being explored.

5.3 Static weight

If no means are provided for an external static support, then the vertical actuators will be required to provide this component. Depending on the capabilities of the system this can be a significant percentage of vertical actuator capability. This static force provides a greater pressure gain in one direction and there will be an asymmetry which will show up as distortion.

This distortion source will occur when there are high demands on pressure (i.e. large velocity, maximum force or high frequency operating conditions). With the proper use of static supports, this type of distortion is largely eliminated.

5.4 Specimen compliance

One of the more difficult types of distortion to control is that due to specimen compliance. This compliance can come from the method used to mount the specimen to the table or from within the specimen itself. The specimen's dynamic properties may be non-linear, they may change with load history, and/or have complex damping properties.

These characteristics cause a number of issues. For example, a lot of the tuning of the system is based on the assumption of a fixed coupled mass, since this determines what the oil column frequencies will be. With a complex specimen these frequencies may be load and time dependent, making optimum tuning difficult.

A resonant specimen with low damping can easily overload the capacity of a system. As a simple example, a cantilever beam with no internal damping when excited in bending at its natural frequency requires an infinite moment to restrain it and yet the actual acceleration at the base is minimal.

6 TYPICAL PERFORMANCE ENVELOPE

Figure 8 shows a typical performance plot of a medium to large seismic system. The predominant sources of distortion are listed along with the approximate regions where they occur. It is very difficult to assign specific values to the various sources. An effort has been made to rank the severity of the various sources.

The most significant are: friction sources which mainly affect low acceleration performance; oil column resonant frequencies which tend to cause problems in the mid-range; servovalve limits which result in distortion problems at the intersection of the velocity and force limits; and mechanical resonant frequencies which occur at the higher end of the frequency range of the system.

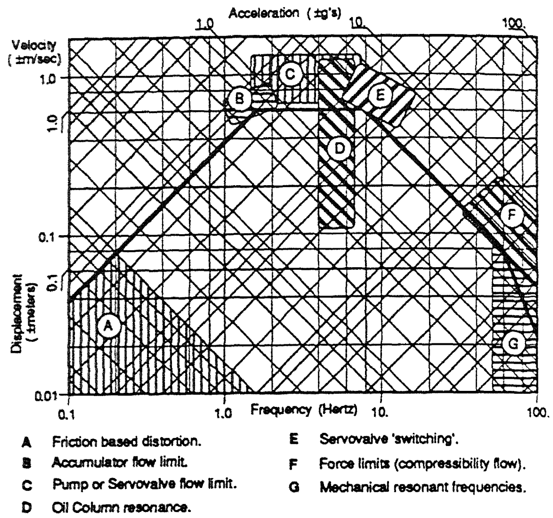

Figure 8. Performance envelope of a typical Seismic System showing regions of distortion.

7 CONCLUSION

Sources of wave form infidelity while performing experimental tests with servo-hydraulic systems have been identified and some methods of reducing distortion errors have been presented. System fidelity with large servovhydraulic systems is a complex subject. It is important when operating servohydraulic systems or when planning a test to take into account the physical realities presented in this paper to optimize test results. This paper also serves as a benchmark for future study and research in minimizing experimental errors with servo-hydraulic systems.

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