FES/BarDyne Technology Transfer Publication
#9

What Time Do You Have?

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Abstract

Hydraulic systems are currently used in a wide variety of industrial applications. In a great number of these cases the response time of the system is a critical parameter. The response time of a hydraulic system is the synergistic results of the response times of all of the components used in the system. Therefore, most component manufacturers will provide information relative to the responsiveness of their components. Unfortunately, this response information is not consistent nor is it compatible from manufacturer to manufacturer. The advent of computer-aided design and analysis software has forced the system designers to be much more aware of the system response characteristics and therefore the component response information. The question "What time do you have?" is directed toward the problems associated with the responsiveness information currently included in many component manufacturers catalog.

Introduction

In the process of designing any type of engineered system, it is becoming more and more likely that a computer-aided design analysis will be performed. The simplest kind of analysis which can be applied is the steady-state evaluation. This type of analysis has been used for a great many years to determine the parameters which must be used in the system in order to accomplish certain design goals. For example, the steady-state speed of the rod in a hydraulic cylinder can be found by dividing the flow rate into the cylinder by the area of the cylinder. Therefore, to determine the flow needed to achieve a given cylinder velocity in a hydraulic system, a cylinder size can be selected and the steady-state speed can then be calculated. However, it is impossible to use the steady-state analysis to evaluate the dynamic characteristics of the system.

In a mechanical system, forces and velocities are usually dynamic in nature. That is, when determining the force and velocity exhibited at one end of a mechanical lever pin in the center, the time functions of the input forces and velocities must be used along with the elastic characteristics of the lever. When such a dynamic analysis is performed it can be seen that the output force does not equal the input force in the case of the mechanical lever. If the input force is rapidly applied the lever will first bend and then the output will

Presented at the Chicago, Illinois **International Fluid Power Exposition and Technical Conference** in April, 1996.

begin. The energy stored in the lever due to the bending action will eventually cause the output to over shoot the input. In the case of a hydraulic system, the operation of a directional control valve may be used to initiate the motion of a linear cylinder. The fluid in the lines between the valve and the cylinder must first compress and then the cylinder motion will commence resisted by gravitational forces until acceleration becomes zero. Such action of elasticity and compressibility along with others create a situation with the system output can be several times that of the input under certain conditions. Therefore, dynamic analysis is essential in the design of efficient and effective engineered systems.

The dynamic analysis of hydraulic systems has always been a great challenge for the design engineer. In order to conduct a worthwhile dynamic analysis, it is necessary to develop a model which adequately describes each and every component and their interaction in the system. These models are nonlinear and require a great amount of time to produce. In addition, the engineer who is charged with the task of performing a dynamic analysis is normally not the engineer who designed any of the components of the system. The system design engineer must rely upon the component manufacturer to provide the necessary data for the modeling process. The last few years at least one computer program (called HyPneu) has been developed which can eliminate much of the effort normally spent in modeling and analyzing hydraulic and pneumatic systems. However, this program requires performance data from the component manufacturers.

One of the most critical parameters of a hydraulic component used for control purposes is the time or dynamic response. In simple terms, the time response of a control component refers to the lag between the input and output when the component is exposed to a dynamic input. For example, consider the pressure compensated pump. When the output pressure is very low the pump will be at full displacement and the output flow will be maximum. If the output flow is suddenly blocked the output pressure will rapidly increase until the pressure setting of the pump is reached. At this point, the pump will begin to destroke. However, due to the time necessary for the stroking mechanism of the pump to change the displacement, the output pressure of the pump can greatly exceed the set point pressure for a very short period of time. In the technology today the time response of a hydraulic component is given in several different ways. Each of these methods of reporting the dynamic or time response of a hydraulic component has some merit but many times the actual definition of the reporting method is not well explained.

This paper will outline several ways in which the dynamic response of a hydraulic component is given in vendors catalogs. The usefulness of each of these ways will be discussed. The main objective of this paper is to demonstrate the benefits which can be gained by properly providing the response characteristics of hydraulic and pneumatic components. Dynamic analysis of engineered systems is a powerful tool but can not be fully exploited without a clear understanding of the time response of each and every control component.

What Time Do You Have?

In a hydraulic and/or pneumatic system, if problems are encountered they will normally appear in the dynamic performance of the system. The steady-state evaluation of these system is fairly straight forward and is usually accomplished before any hardware is assembled or even ordered This avoids any problems in the steady-state performance. However, the dynamic analysis of a hydraulic system has presented great challenges to the engineering community in the past. In recent years new and more useful computer-aided design analysis software has become available for hydraulic and pneumatic system. These programs permit the system designer to use component manufacturer supplied information to define the components of his system and topographical information to define the system configuration. Most of the information supplied currently by the component manufacturer is clearly adequate for use in design analysis software. However, there is one area which needs more explanation and definition. Component suppliers spend a great deal of time and effort to provide performance data on their components. If these data are presented in the wrong or obscure protocol, they may be useless to the system analyst.

When a component is used in a control system, the dynamic performance of that component is extremely important in the analysis of the system. In modern hydraulic systems it is commonplace to use one of the compensated pumps or valves. Today, hydraulic pumps are manufactured with pressure compensation, flow compensation, a combinations of these two. It should be obvious that the dynamic performance of the overall hydraulic system which uses one of these components will depend to a large extent upon their performance characteristic. Most component manufacturers conduct testing to determine the dynamic characteristics of their components and they publish the information in their product literature. However, if it is not given in a defined format or protocol the time and effort may be wasted.

In hydraulic control components such as the servovalve or the compensated pump, one or more of the output parameters are used in a feedback arrangement to control the output according to some algorithm. For example, in a servo valve or a feedback type proportional valve, the position of the valve spool is controlled by a closed loop mechanism. The mechanism used in some servo valves is the flapper/nozzle arrangement. The feedback signal is provided by the mechanical action of the valve spool engaging the spring/flapper assembly. In this case, the input signal comes from the torque motor armature and the feedback signal comes from the main valve spool while the flapper acts as a summing junction. In the case of a pressure compensated pump, the pressure at the output port acts upon internal valving to position the spool. The displacement of the pressure compensated pump is controlled by the position of the swash plate. When the output pressure from the pump exceeds some set point pressure the valving will direct fluid appropriately to decrease the swash plate angle and thus reduce the displacement of the pump. When the pressure is less than the set point pressure the swash plate angle will be increased by the feedback control loop.

Since these control components incorporate dynamic elements, the dynamic characteristics can be described by examining their response to either a step function input or a sinusoidal input[1,2,3,4,5,6]. There are two potential results when employing a step function to determine the dynamic response of a control component under no load conditions. In one result, the component may appear to be a first order system as shown in Fig. 1. As can be seen in this figure, the step function input is suddenly switch on and the output of the component increases much slower but finally approaches the steady-state value. There are two time parameters of interest to control engineers. They are

- *Time Constant*: This is the time to reach about 63% of the demand output. This represents one time constant in control technology terminology.
- *Settling Time*: The time required for the demand output to reach and stay in a defined tolerance band.

The second order type of response is shown graphically in Fig. 2. In the case of an under damped second order system response to a step function input, the output will rapidly increase, over shoot the steady-state condition and eventually settle at the appropriate value. While most control components will exhibit second order behavior, some can be adequately represented by the first order characteristics. The time parameters related to a second order system response to a step input are stated as follows:

- *Delay Time*: The time required to reach 50% of the demand output.
- *Rise Time*: The time required to rise from 10 percent to 90 percent of the demand output.
- *Maximum Overshoot Time*: The time at which the maximum overshoot occurs
- *Settling Time*: The time for the system to reach and stay within a stated plus-and-minus tolerance band around the steady state demand output.

Figure 1
Step Response of a First Order System

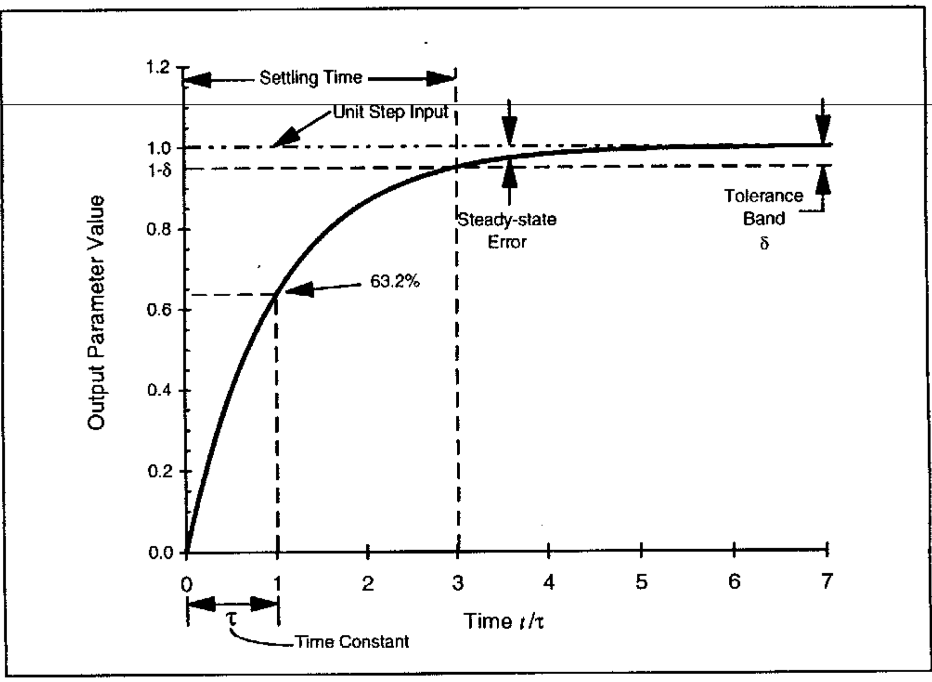

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Figure 2
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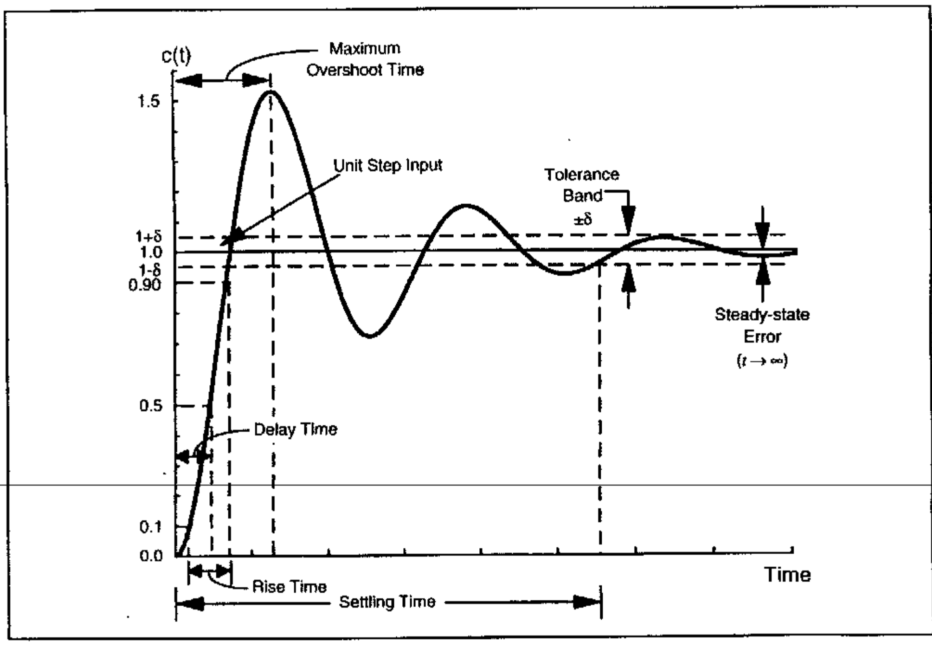
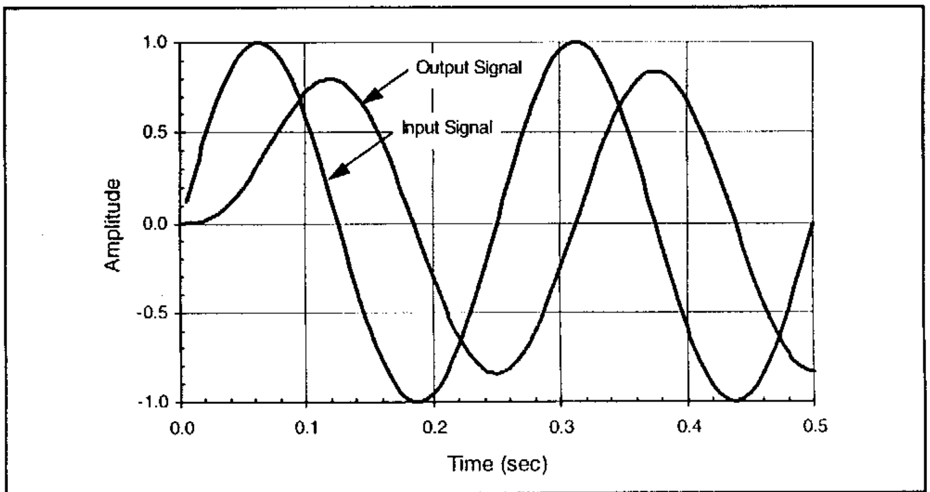


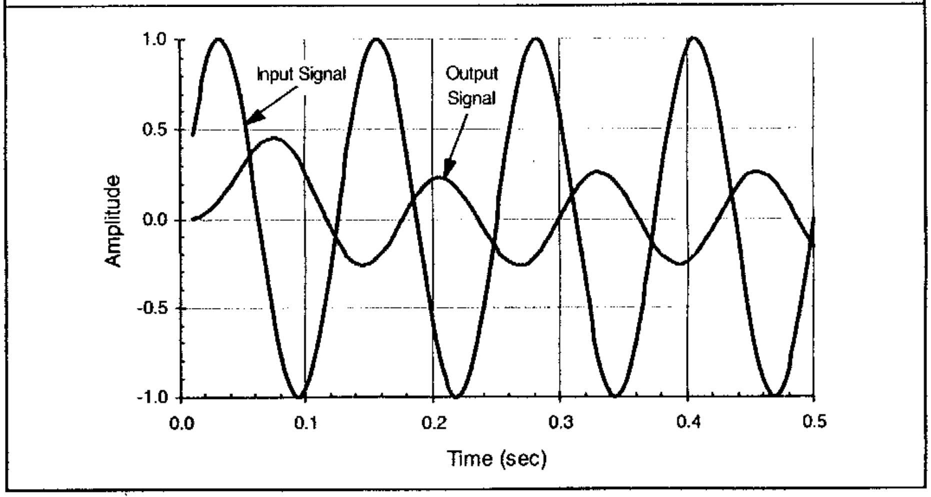
Another and many feel more important measure of dynamic response is its output characteristics when provided a sine wave type input signal. This is normally called a frequency response analysis. As illustrated above, it is very convenient to use time response to evaluate the dynamic performance of a first order and/or second order component. However, it is very difficult to analyze higher order components in the time domain. Nevertheless, frequency analysis provides a very simple and straightforward approach and can be used to evaluate component whose response is higher than second order. In the frequency analysis, the two principle parameters -- overall amplitude ratio and phase angle (shift), can be obtained by summing their individual components graphically.

When a control component is provided a sinusoidal input, the output will follow this input with some error. The amount of error will depend upon the frequency of the input. At low frequencies the output will follow the input very closely. However, as the frequency increases the error will also increase. An example of the input and output of a typical control component when subjected to a sinusoidal input is shown in Fig. 3 a and b. The difference between the peak values of the input and that of the output is normally called attenuation while the time difference is referred to as the phase lag or phase shift. The frequency response information is normally given as a graph of the attenuation (amplitude ratio) and the phase lag versus frequency.

Figure 3
Second Order System
Response to a Sinusoidal Input



(a) Low Frequency



(b) High Frequency

Any of the preceding response information can be used to describe the dynamic response of the control component in a system model. However, the exact definition of the time given by the component manufacturer must be accurately and completely defined. For example, a statement that the response of a control component is 50 milliseconds is useless without knowing the definition of this data. This paper will address various time response information which is many times reported and discuss the various ways that this information can be used in a dynamic modeling and simulation project. In addition, the paper will suggest a protocol which could be utilized to reduce the confusion which pervades the industry today.

Response Time Protocol[7]

Since the actual time that it takes a control component to response is normally a matter of milliseconds, it may not be apparent what the appropriate definition of "step response" entails. From the above discussion, some of the definitions used to define step response can be summarized as follows:

- The time to reach about 63% of the demand output
- The time to go from 10 % to 90% of the demand output, ignoring the initial and final variations
- The time required to reach 100 % of the demand output, ignoring the fact that the output will overshoot in the case of a second order component.

Obviously, the parameter of the utmost interest is the time required to reach a demand final state. From linear control theory, a first order system response to a step input is given as follows:

$$c(t) = 1 - e^{-\frac{t}{\tau}} \tag{1}$$

where $c(t)$ system time response
t time
τ time constant

Eq.(1) indicates that a first order system will take an infinitive time to reach 100% of the demand output. Hence, in practice, the *settling time* concept used to characterize such a system needs to define a tolerance zone. Table 1 shows the response, c(t), as well as the corresponding percent of steady state error, Er(t), in term of the time has elapsed after the step input is imposed onto the system. As can be seen from this table that c(t) will exceed 99% of the steady state value when the elapsed time has reached about more than five times of the time contant. Therefore, if the settling tolerance is defined as less than 1% error of the steady state value, it may be said that the response is virtually complete after a period of approximate five times of the time constant. Thus, the response time (T_r) can be defined as five times of the time constant, namely, $T_r=5\tau$.

Table 1. Step Response of a First Order System.

t/τ	c(t)	Er(t), %
0	0.0000	100.0000
1	0.6321	36.7879
2	0.8647	15.5335
3	0.9502	4.9787
4	0.9817	1.8316
5	0.9933	0.6738
6	0.9975	0.2479
∞	1.0000	0.0000

The dynamic response of an under damped second order system with zero initial states can be mathematically described as follows:

$$c(t) = 1 - \frac{e^{-\xi\omega_n t}}{\sqrt{1-\xi^2}} \sin[\omega_n \sqrt{1-\xi^2}\, t + \tan^{-1}\frac{\sqrt{1-\xi^2}}{\xi}] \qquad (2)$$

where ξ damping ratio
ω_n natural undamped frequency

It is interesting to note that the response speed is governed by an exponential function with an exponent of $-\xi\omega_n t$. This exponential function controls the envelope of the damped sinusoid. Comparing Eq.(2) to Eq.(1) which is for a first order system, the equivelent time constant for a second order system to a step input is

$$\tau = \frac{1}{\xi\omega_n} \qquad (3)$$

Therefore, if we assume the response time is five times of the time constant, the settling time required for a second order system to reach and stay within 1% of the demand steady state value will be 5τ, or

$$T_r = \frac{5}{\xi\omega_n} \qquad (4)$$

In physics, the time constant is an inherent property of a component or system. Conseqently, the use of the time constant concept provides a unique figure of merit to assess the response speed of both the first order and second order system. However, as stated previously, the defination of the response time is very subjective. Someone may use a settling tolerance of 5%, then the response time will be 3τ; while some may take 2% tolerance, then the response time will be 4τ, and so forth. The following equation provides a protocol to communicate between the various definitions of the response time.

$$T_{r0} = T_{rx} \frac{\ln[1-c_0]}{\ln[1-c_x]} \qquad (5)$$

where T_{r0} time required to reach and stay within $[1-c_0]$ of steady state value
T_{rx} time required to reach and stay within $[1-c_x]$ of steady state value
c_0, c_x defined fraction of steady state response

For example, if one defines the response time of his system is 10 ms based on a 95% criterion, then the equivelent response time based on a 98% criterion will be 13.06 ms as calculated below by Eq.(5):

$$T_{r98} = (10\text{ms})\frac{\ln(1-0.98)}{\ln(1-0.95)} = 13.06\text{ms} \qquad (6)$$

From the linear control theory, parameters τ, ξ, and $\omega_n$used in the time domain analysis can also be used to define their counterpart in frequency analysis, or vise versus.

Example of Second Order Component

To illustrate the concepts and thoughts which have been expressed in this paper, consider the generalized servo valve shown in Fig. 4. This is a flapper/nozzle type of servo valve and will be found in many of the hydraulic systems being used currently. In order to model and simulate this component a model must first be developed for the valve and then it must be incorporated into a system model. The servo valve model must include at least three aspects in order to closely represent to actual component. One aspect is the mass, spring, and damping involved in the motion of the main spool. The second is the flow metering produced by the main spool. The third aspect is the summing and feedback situation involving the flapper/nozzle arrangement. A schematic of such a model development using the HyPneu program[8] is shown in Fig. 5. In this figure the metering characteristics are represented by variable area orifices operated by a signal which equals the position of the main spool. The dynamics of the main spool are included by incorporating a mass, a spring,and a damper attached to a double rod double acting cylinder which represents the actual spool. The flapper is a special icon which receives the error from the summation of the actual spool position and the desired spool position demanded by the torque motor. This error is used to adjust the flow area of the two nozzles which in turn creates the appropriate pressure to move the main spool.

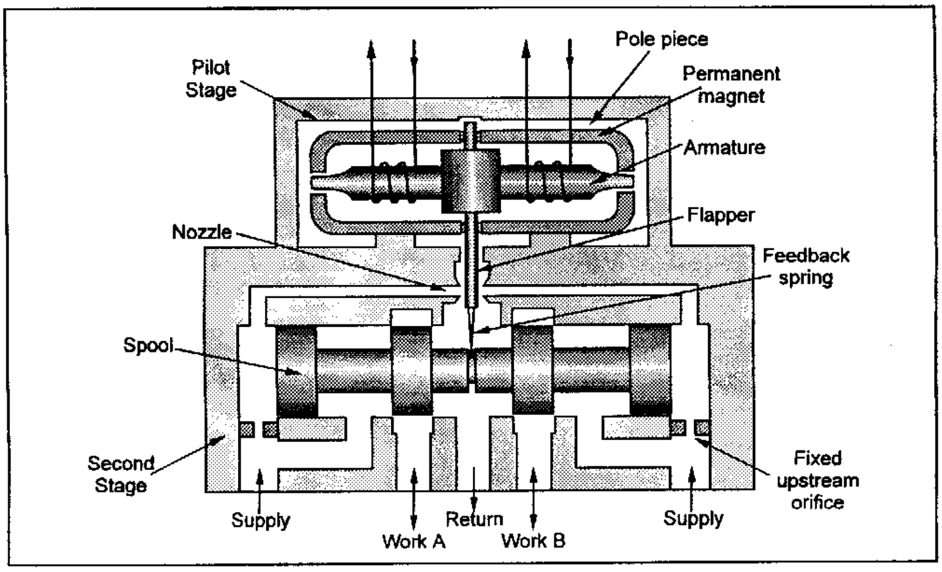


Figure 4
Generalized Illustration of Typical Electromechanical Servo Valve

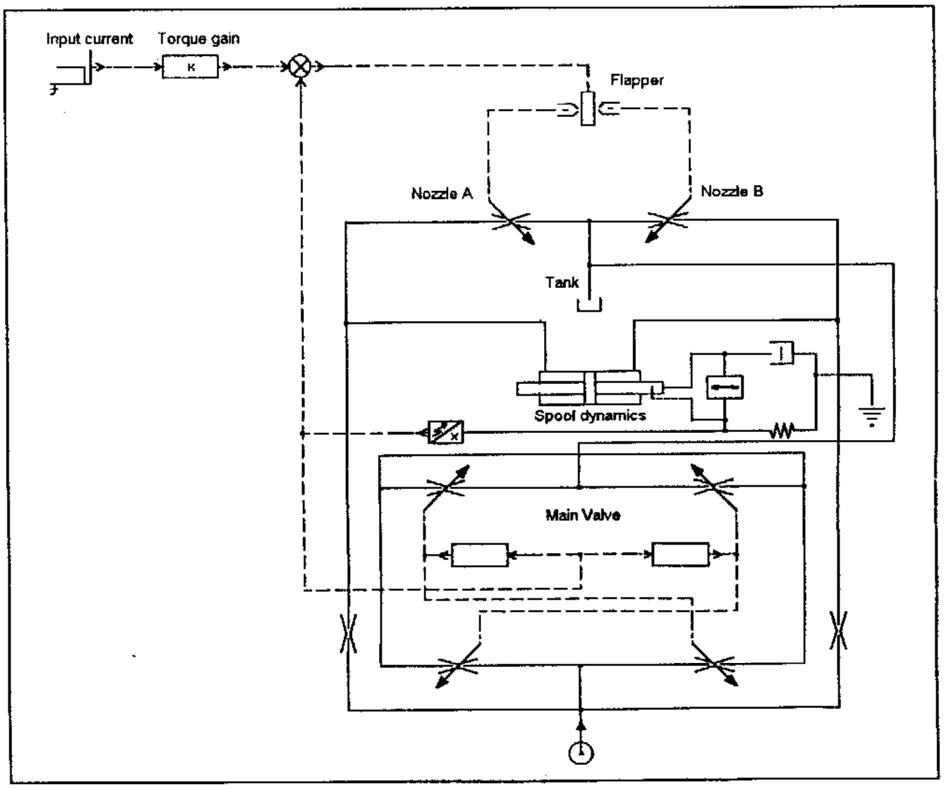


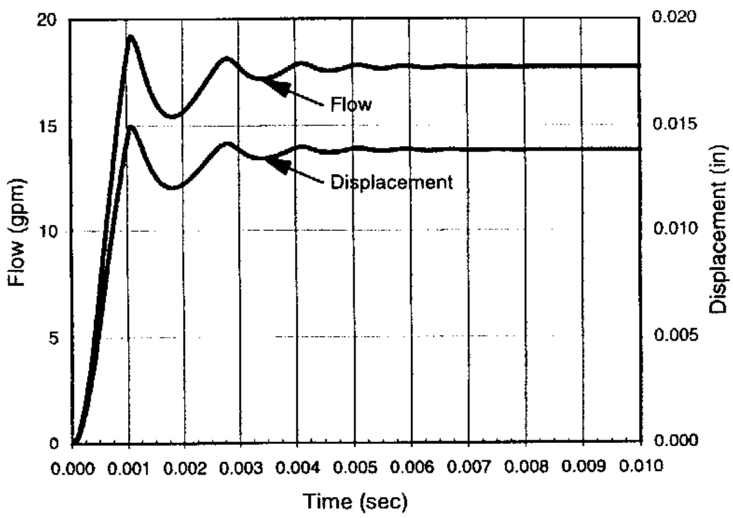
Figure 5
HyPneu Schematic of Servo Valve

This valve model is given a constant pressure source as a pump and a simple tank for the no-load operation of the valve. In other words, the system schematic shown in Fig. 5 incorporated a pressure source as a hydraulic input to the valve. The flow passing through the valve simply goes to the tank. The control input for simulation is a step input to the torque motor. Therefore the system will not flow hydraulic fluid until the torque motor causes the flapper to impide the flow from oneof the nozzles which creates back pressure on the main spool and initiates spool motion. The performance of this valve is shown in Fig. 6. This graph shows the motion of the main spool and the flow rate through the valve due to a step input of current to the torque motor. Notice that the spool movement accelerates to some velocity and overshoots the steady-state position of about 0.014 inches. The spool then decelerates and eventually undershoots the steady-state position. In this example the valve has a settling time of approximately 0.0045 seconds for valve spool position to reach and stay within 1% of 0.014 inches and flow to stay within 1% of 17.7 gpm. The rise time is about .0005 seconds while the overshoot time is approximately .0011 seconds. From this example it should be apparent that the time response is very important but if the values which are reported to describe this performance are not adequately defined or do not follow some standardized protocol they may not be useful for their intended purpose.

Conclusions

The design of hydraulic systems is becoming more sophisticated today because the demand for high performance from these systems has increased. When the systems performance was not very high, design engineers were able to successfully rely upon steady-state evaluation to insure that the system they created would accomplish the intended mission. However, the successful design of a high performance system today requires much more concern with the dynamics of the system. In fact, most of the problems encountered in the performance of engineered system today is involved with the dynamics of the systems. In a great many of these systems one or more components are incorporated which have a dynamic performance themselves. In addition, computer programs have been developed which can be used to perform dynamic system design analysis. These programs usually require component information to complete the system description. One of the most critical parameters needed by the system analyst is the dynamic response time of the various components.

Figure 6
Servo Valve Performance Graph



Manufacturers of components such as compensated pumps and motor as well as servo valves conduct testing to define the dynamic response characteristics of their components. However, the protocol in reporting this information has not been consistent in the past. This paper has addressed the problem of dynamic response time and has outlined various approaches normally encountered in the reporting of response time, time constants, rise time, settling time, etc. In addition, the paper presents a protocol which should be adopted by component manufacturers in reporting the dynamic response of their components. This protocol does not request different data be acquired during the dynamic testing but requests a more complete description of the information normally reported by the manufacturer. For example, when the settling time is reported the tolerance band used in determining this time must be given. In order to determine the value for any of the various response times reported, the manufacturer has to rely on some tolerance band or some limiting definition. This paper simply requests that this information be included when the response data is given. Eq (5) presented in this paper can then be applied to find the response time defined using different tolerance bands.

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