

Modeling and Simulation of Metal Forming Equipment

W.G. Frazier, E.A. Medina, J.C. Malas, and R.D. Irwin

The demand for components made from hard-to-form materials is growing, as is the need to better understand and improve the control of metal forming equipment. Techniques are presented for developing accurate models and computer simulations of metal forming equipment for the purpose of improving metal forming process design. Emphasis is placed on modeling the dynamic behavior of hydraulic vertical forge presses, although similar principles apply to other types of metal forming equipment. These principles are applied to modeling and simulation of a 1000 ton forge press in service at Wright-Patterson Air Force Base, Ohio, along with experimental verification.

Keywords
control systems, forging, hydraulics, metal forming

1. Introduction

THE MOTIVATION for developing accurate models and computer simulations of metal forming equipment is threefold. First, the ability to quickly develop improved press control algorithms is greatly enhanced by the ability to perform repeated computer experiments without the need for costly and time-consuming experimental tests that can interfere with production. Second, as the use of finite-element modeling (FEM) techniques for the analysis and design of metal forming processes continues to increase and become more sophisticated, the need to integrate accurate equipment models into the FEM-based simulations will increase. Third, with the possibility of sensing workpiece conditions directly during forming operations (Ref 1), it is possible that in the future these measurements could be fed back to the control computer of the metal forming equipment in order to achieve final workpiece qualities even in the presence of variations in initial workpiece and equipment conditions. From a process control perspective, this approach should provide for the highest level of robustness and repeatability in production.

Behind each of these motivational components is the need for improved control and predictability of equipment behavior. This need is driven by the desire to achieve near-net shapes, higher quality, higher yields, and better control of microstructure for parts made from hard-to-form materials.

2. Press System Description

A simplified block diagram of a typical hydraulic press system is shown in Fig. 1, where the lines represent possible directions of fluid flow. The system is powered by an electric motor that drives a hydraulic pump. Transient demands for high ram speeds are met by an accumulator system. A counterbalance is used to support and return the main ram to the top of its stroke after a forging operation is completed. The servomanifold controls the flow of fluid to the main ram cylinder and to the tank.

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2.1 *Hydraulics*

The primary considerations involved in the modeling of hydraulics in a press system include the bulk modulus of the fluid, the flow rate of fluid from the pump, the head pressure, the flow rate through the servomanifold, and the main ram pressure. Other items that must be included are the hydraulic pressure on the counterbalance subsystem and pressure losses due to the flow of fluid through circular pipe.

2.1.1 Pumps

Pumps usually are driven at a constant speed by an electric motor, while the amount of fluid being delivered by the pump at any moment usually is governed by the position of an actuating spool of a servovalve (Ref 2). The time response of this servovalve is the dominant factor in the dynamic performance of the pump. Therefore, from a mechanical response perspective, the modeling of pump behavior can be viewed as being similar to the mechanical response of servovalves in general, as will be discussed in section 2.2.

2.1.2 Head Pressure

The head pressure of a pump-only system or a system using an accumulator in which the separator tank is completely full or empty is modeled from first principles of fluid mechanics (Ref 2) by the relationship:

$$\dot{P}_{\mathrm{head}}(t) = \frac{q_{\mathrm{pump}}(t) - q_{\mathrm{sm}}(t)}{C_{\mathrm{head}}(t)} \quad \text{(Eq 1)}$$

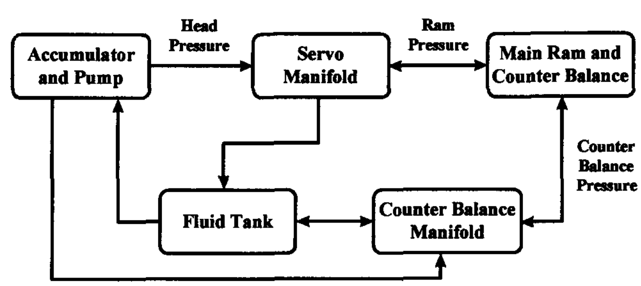

Fig. 1 Simplified block diagram of a typical hydraulic press

where q_{pump}, q_{sm}, and C_{head} are the volumetric flow rate into the volume between the pump and the servomanifold, the flow rate out, and the hydraulic capacitance of that volume, respectively. The hydraulic capacitance is simply:

$$C_{head}(t) = \frac{V_{head}(t)}{\beta} \quad \text{(Eq 2)}$$

where V_{head} and β are the volume between the pump and the servomanifold and the bulk modulus of the fluid, respectively. The volume is shown to possess a time dependence to account for the fact that the volume can change if there is an accumulator in the pump circuit. In systems employing accumulators in which the separator tank is not full (the usual case), the head pressure is given by the differential equation:

$$P_{head}(t) = \frac{P_0 V_0^n}{V_{gas}^n(t)} \quad \text{(Eq 3)}$$

where P_0, V_0, and V_{gas} are initial head pressure, the initial volume of nitrogen, and the instantaneous volume of nitrogen, respectively. The parameter n is the molar specific heat of the gas. Assuming the hydraulic fluid is imcompressible with respect to the nitrogen, the instantaneous volume of nitrogen is determined from the net flow of hydraulic fluid into the head volume:

$$\dot{V}_{nit}(t) = q_{pump}(t) - q_{sm}(t) \quad \text{(Eq 4)}$$

2.1.3 Ram Pressure

The most important hydraulic component affecting ram speed is the pressure on the ram piston. This pressure is modeled by the differential equation:

$$\dot{P}_{ram}(t) = \frac{q_{sm}(t) - A_{ram} v_{ram}(t)}{C_{ram}(t)} \quad \text{(Eq 5)}$$

where A_{ram}, v_{ram}, and C_{ram} are the cross section of the ram piston, the velocity of the ram, and the hydraulic capacitance of the fluid volume between the servomanifold and the ram piston, respectively. This capacitance is given by:

$$C_{ram}(t) = \frac{V_{nom} + A_{ram} \Delta x_{ram}(t)}{\beta} \quad \text{(Eq 6)}$$

V_{nom} and Δx_{ram} are the nominal volume and the displacement of the ram from its nominal position, respectively. The term $A_{ram} v_{ram}(t)$ in the numerator of Eq 5 takes into account the effect of the rate of change of the volume as the ram descends. The force applied to the main ram due to the ram pressure is:

$$F_{sm}(t) = P_{ram}(t) A_{ram}$$

2.1.4 Counterbalance Pressure

The counterbalance pressure in press systems is commonly maintained by a relief valve. The relief pressure of this valve is chosen so that the weight of the ram can be supported entirely by the counterbalance cylinders. The weight that can be supported by such a system is given by:

$$W_{sup} = A_{cb} P_{rv} \quad \text{(Eq 7)}$$

where A_{cb} and P_{rv} are the cross-sectional area of the counterbalance pistons and the relief valve pressure, respectively. When pressure is applied to the ram by fluid from the servomanifold, the load on the counterbalance exceeds W_{sup} and the relief valve opens, allowing fluid to flow and the ram to descend. The force (no load) applied to the ram via the counterbalance is given by:

$$F_{cb}(t) = A_{cb}\left[P_{rv} + \gamma_{cb} A_{cb} \dot{x}_{ram}(t)\right] \quad \text{(Eq 8)}$$

where the term $\gamma_{cb} A_{cb} \dot{x}_{ram}$ represents the additional pressure applied to the ram due to the frictional effect of fluid flowing through pipe. The parameter γ_{cb} is a factor for determining the pressure change per unit of volumetric flow rate. It is determined from the Hagen-Poiseuille law (Ref 3):

$$\gamma_{cb} = \frac{128 \mu L}{\pi D^4} \quad \text{(Eq 9)}$$

where D and L are the diameter and length of the pipe in inches and μ is the viscosity of the fluid. This effect can add a significant amount of viscous damping to the overall system.

2.2 *Servovalves*

The primary factors determining the flow of fluid through the servomanifold are the current state of the servovalves and the differential pressure across the valves. Although higher-order nonlinear analytical models of the mechanical behavior of servovalves can be developed from first principles, second- or third-order linear models identified from experimental data have proved quite satisfactory for producing accurate simulations. A typical model in state-variable form (Ref 4) is given by:

$$\begin{aligned} \dot{x}_1(t) &= x_2(t) \\ \dot{x}_2(t) &= A\omega_n^2 v(t) - 2\zeta\, \omega_n x_2(t) - \omega_n^2 x_1(t) \end{aligned} \quad \text{(Eq 10)}$$

where x_1, x_2, and v are the spool position, spool velocity, and applied voltage, respectively; A, ω_n, and ζ, respectively, are the gain (e.g., mm/V), natural frequency, and damping ratio of the valve. Equation 10 describes only the mechanical response of the power spool. The volumetric flow rate of fluid through a three-way servovalve into the main ram fluid volume can be modeled as a function of the position of the power spool and the differential pressure across the valve by a modified orifice flow equation (Ref 2):

$$q = \mathrm{sgn}[x_1(t)]\mathrm{sgn}[\Delta P(t)]\, f[|x_1(t)|]\, \sqrt{|\Delta P(t)|} \quad \text{(Eq 11)}$$

where

$$\Delta P(t) = P_{\text{head}}(t) - P_{ram}(t) \quad \text{(Eq 12)}$$

and the function f is an experimentally determined relationship between spool position and flow rate. Experience reveals that a third-degree polynomial is usually sufficient for the function f. Typical flow curves for a servovalve are shown in Fig. 2 for several values of differential pressure. The use of the signum function provides a means for modeling the flow-direction switching capabilities of a three-way valve. The four flow possibilities are defined in Table 1.

2.3 *Ram Dynamics*

The ram can be modeled as a rigid body with a mass M possessing a single degree of freedom. More complex bending and inertial effects can be included if increased precision is justified. The equation of motion for the ram is given by:

$$a_{\text{ram}}(t) = \frac{[F_{\text{sm}}(t) - F_{\text{cb}}(t) + W_{\text{ram}} - F_{\text{fric}}(t) - F_{\text{load}}(t)]}{M_{\text{ram}}} \quad \text{(Eq 13)}$$

where a_{ram} is the acceleration of the ram, F_{sm} is the force due to hydraulic pressure from the servomanifold, F_{cb} the force due to hydraulic pressure from the counterbalance, W_{ram} is the weight force of the ram, F_{fric} is frictional force, F_{load} is forces due to workpiece loading, and M_{ram} is the mass of the ram.

The force of friction F_{fric} between the ram piston and the seals can be modeled as coulomb friction (Ref 5). Values for the parameters associated with this difficult-to-measure effect are best determined by adjusting the simulation parameters to best match the experimental data. A model for this effect is given by:

$$F_{\text{fric}}(t) = \mathrm{sgn}[v(t)][c|v(t)| + b]$$

where $v(t)$ is ram velocity, c is similar to the effect of viscous friction, and b is an offset that models the effect of sticking. Note that if b is zero, then this model reduces to the standard model of viscous friction. The accurate model of the workpiece loading, F_{load}, is very difficult for situations involving the forming of complex shapes due to the interaction of the workpiece and dies and the mechanical properties of the workpiece material. In most instances it is necessary to make simplifying assumptions in order to formulate a practical model.

Table 1 Flow possibilities for a three-way servovalve

	$\Delta P < 0$	$\Delta P \geq 0$
$x_1 \leq 0$	Tank to ram	Ram to tank
$x_1 > 0$	Ram to head	Head to ram

2.4 *Sensors*

A modern press system can employ several types of sensors for safety, diagnostics, and feedback control. For the purpose of feedback control, measurement of pressure, linear displacement, and linear velocity are most important. The pressure transducers (Ref 6) are used to measure the head pressure and the pressure on the ram piston. Since the difference in these pressures provides for the differential pressure across the servomanifold, the control computer can use this difference for determining commands to the servovalves. This is the most common method employed for ram speed control. Used alone, it has the disadvantage of requiring very accurate models of the flow curves of the servovalves since the actual ram velocity is not used. Any error in the flow curve models will translate directly into errors in ram speed. The measurement from the head pressure transducer is also useful for actuating the main pump in order to maintain the appropriate head pressure. The response time of a pressure transducer is significantly shorter than that of a press; therefore, these transducers and their associated electronics can safely be modeled as simple gains:

$$\text{Voltage} = \text{Gain} \times \text{Pressure} \quad \text{(Eq 14)}$$

Linear displacement transducers (Ref 7) play a critical role in measurement of ram stroke. High accuracy and precision of these sensors is very important for ensuring repeated part quality. Various types of linear displacement transducers are available that use a variety of technologies. Most have very rapid response times and can, like pressure transducers, be modeled as simple gains. Direct measurement of ram velocity is difficult due to the lack of reliable sensors for measuring translational velocity over a large dynamic range. A common method for ob-

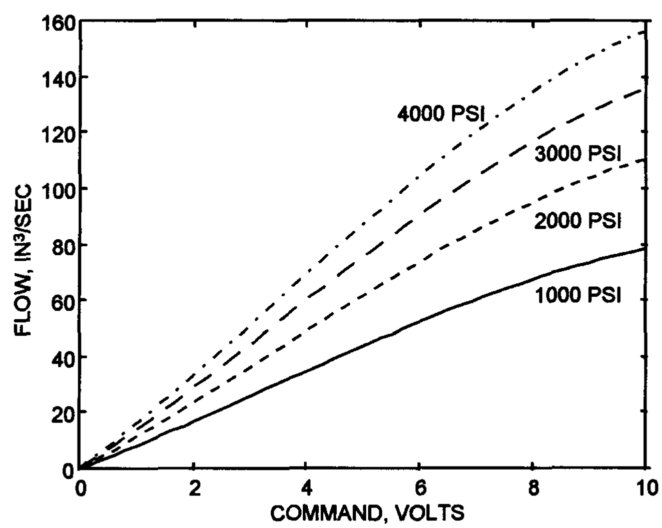

Fig. 2 Typical flow curve for a servovalve as a function of spool position

taining translational velocity estimates is to numerically differentiate successive position measurements. This method is fraught with pitfalls and must be used with extreme caution due to the presence of electronic and analog-to-digital converter quantization noise (Ref 8) on the position measurements. The quantization noise problem is especially acute in situations where low velocities and high computer sampling rates are present. The simplest velocity estimate is calculated by:

$$\hat{v}(t) = \frac{s(t) - s(t - T)}{T} \quad \text{(Eq 15)}$$

where T is the sampling period and s is ram position. More complex schemes can be employed that are effectively digitally filtered estimates of velocity (Ref 8). These techniques reduce the effect of noise problems at the expense of reduced response time. In many applications this trade-off is justified.

Another method is to numerically integrate accelerometer measurements. This technique considerably reduces the noise problem, but can introduce errors due to drift in the accelerometer output. The formula for the trapezoidal integrator is given by:

$$\hat{v}(t) = \hat{v}(t - T) + \frac{T}{2}[a(t) + a(t - T)] \quad \text{(Eq 16)}$$

where a is ram acceleration.

2.5 *Control Processor*

Press systems typically use two levels of processing for control. The higher level of control is usually called a supervisor and provides functions such as engaging safety locks, monitoring pressure switches for excessive pressures, and so on. This level of control is usually provided by an industrial programmable logic controller (PLC). The lower level of control is usually called the servoloop and is responsible for having the main ram track the desired velocity or position profile. This level of control can be provided by a PLC with special servocontrol features or by an industrial PC-based system with custom software. Input to and output from the computer control system is usually provided by 12-bit analog-to-digital and digital-to-analog converters, respectively. Modern electronic computer control systems have the capability of providing very high sampling rates, which in theory provide the capability of rapid response to changes in load conditions. However, the use of excessively high sampling rates can introduce problems, as described previously.

3. Computer Simulation of Dynamic Systems

The use of a system model to predict the behavior of an actual system is desirable in many situations. In the case of design, the actual system may not yet exist and several possible configurations may need evaluation. In the case of analysis, an experiment on the actual system may take too much time or too little time, may be too expensive, or may even destroy the system (very likely an undesirable situation). Many complex, real-world systems cannot be accurately described by mathematical models that can be evaluated analytically to obtain responses to a particular set of inputs or parameters. The simulation alternative consists of evaluating the model numerically with the inputs and parameters in question to determine how the outputs of interest are affected (Ref 9).

Simulation has been a tool for analysis and design of engineering systems for many years. In particular, simulation of continuous-time and discrete-time dynamic systems has been the subject of vast research for several decades. Simulation languages such as CSMP and ACSL have been specifically developed for building dynamic system simulation models.

3.1 *Graphical Simulation Paradigm*

The state of the art in simulation of dynamic systems includes software packages that not only aid the engineer in building simulation models, but also facilitate the creation of models and interpretation of results by means of sophisticated graphical user interfaces. Simulink (The MathWorks, Inc., Natick, MA), VisSim (Visual Solutions, Inc., Westford, MA), and Matrix$_X$ (Integrated Systems Inc., Sunnyvale, CA) are three of the several packages of this nature that are available.

In the sense used here, a simulation model is represented by a block diagram in which all relevant elements and relationships that model the system are included. For the case of the forging system under consideration, which include the forge press, the workpiece loading, and the internal mechanisms by which the microstructure of the workpiece evolves, the top-level block diagram of a VisSim simulation model is shown, for illustration purposes, in Fig. 3. Each block in such a simulation diagram models an element or group of elements in the actual system, a data input port, or an output port. Each line represents a path for the flow of information or energy. Note in Fig. 3 that the fundamental blocks in the forging process simulation are the forge press, the controller, and the tooling package. Each of these blocks comprises several levels of subsystems, which can be observed with convenient editing tools in the graphical simulation environment.

3.2 *Automatic Equipment Simulation Code Generation for the Metal Forming Industry*

One objective underlying this work is to provide the metal forming industry with scientifically based tools that will expedite design by allowing the user to perform "what if" studies that include all aspects of the manufacturing process. For that reason, a goal is to develop a software module that will generate the response of the system to specific inputs without having to use a simulation package—that is, a stand-alone program or a set of routines that will simulate the system without the need for a dedicated simulation software package on the part of the end user. This can be achieved by using automatic code-generation features that can be purchased for any of the state-of-the-art simulation packages mentioned previously.

The idea here is to build the simulation model in one of the available simulation software packages and then generate high-level language codes that can be compiled to generate a program or library that will simulate the system. The original simulation model can be developed by a consulting firm that can deliver executable codes or libraries for simulation of the particular system. Such executable code or library can then be used in conjunction with existing finite-element analysis software for the simulation of the forging process by the engineer in charge of the design.

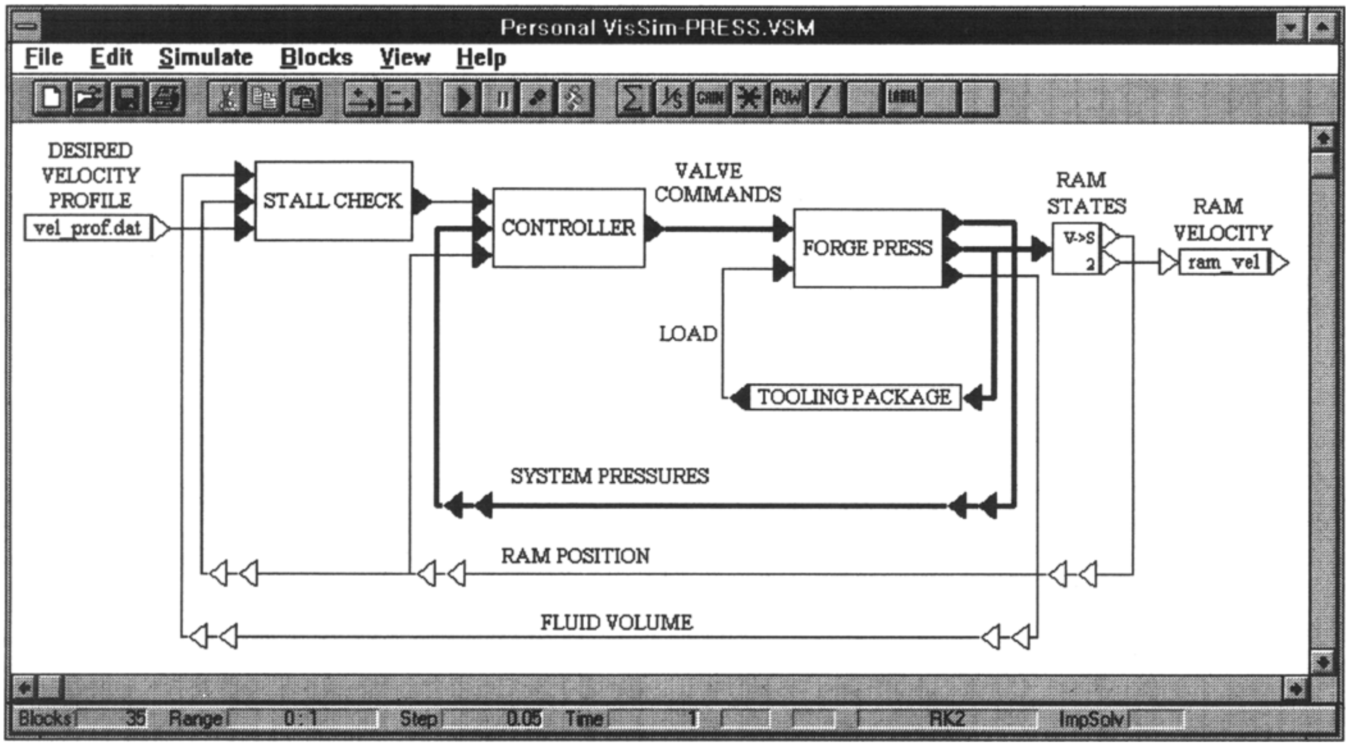

Fig. 3 VisSim window showing top-level diagram of a forging process simulation

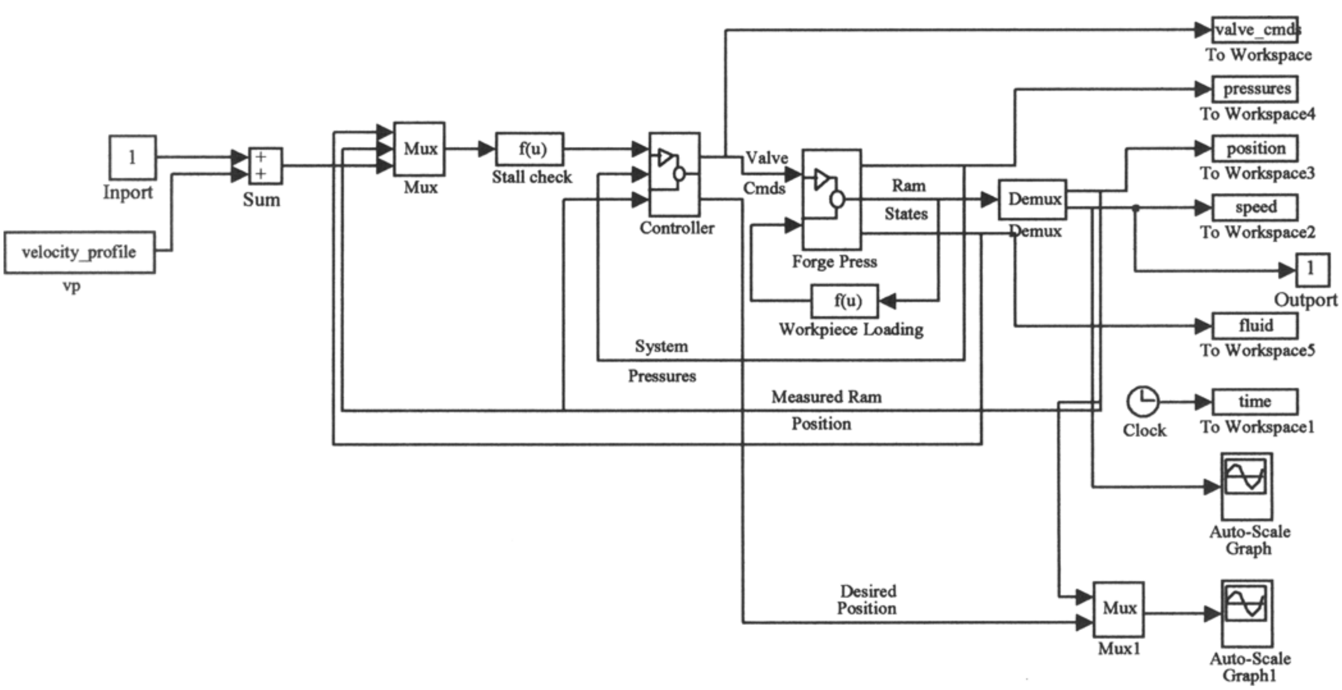

Fig. 4 Block diagram of the overall press system

4. Application to the Erie 1000 Ton Forge Press

The Erie 1000 ton forge press located at Wright-Patterson Air Force Base (WPAFB) is a vertical hydraulic forge press with a programmable, computer-based ram velocity control system that employs hydraulic pressure and ram position feedback. The press was manufactured by Erie Corporation (Erie, PA, USA), and the hydraulic control system was designed and built by Oilgear Inc. (Milwaukee, WI, USA). The press has been in service since 1992 for performing manufacturing and metallurgical research at WPAFB.

The power plant consists of an axial piston pump with an 5.17 L/s capacity driven at 1200 rev/min by a 149 kW electric motor. Transient demands for higher flow rates are provided by a 60.6 L hydraulic separator tank and a 738 L nitrogen bottle. Nominal head pressure is 26.2 MPa. The stroke of the ram is 381 mm, and the maximum speed of the ram is approximately 127 mm/s. The maximum speed cannot be maintained over the entire stroke due to the limited capacity of the pump and the size of the separator tank. The cross-sectional area of the main ram piston is 3813 cm^2. The main ram is supported by two counterbalance pistons with cross-sectional areas of 186 cm^2 each. The relief valve pressure on the counterbalance is 6.89 MPa. This implies that the counterbalance can support 257 kN. The nominal weight of the main ram is 187 kN.

The speed of the press is regulated by the use of two Oilgear three-way servovalves and one Parker proportional throttle valve. The gains of these valves are:

Oilgear 800 three-way servovalve	0.379 (L/s) at 20.68 MPa
Oilgear 1600 three-way servovalve	2.32 (L/s) at 20.68 MPa
Parker TDA 100 proportional throttle valve	20.8 (L/s) at 20.68 MPa

The use of these three valves in parallel provides capability for wide dynamic range and precise control of ram velocity. The valves are controlled by an Intel 80386-based industrial computer that uses head pressure, ram pressure, and ram displacement feedback for velocity control. The details of the Oilgear-developed control law for the valves are proprietary.

Using the principles described in section 2, a computer simulation was developed using the simulation software Simulink. A block diagram for the overall press system is shown in Fig. 4. This diagram shows the interconnections be-

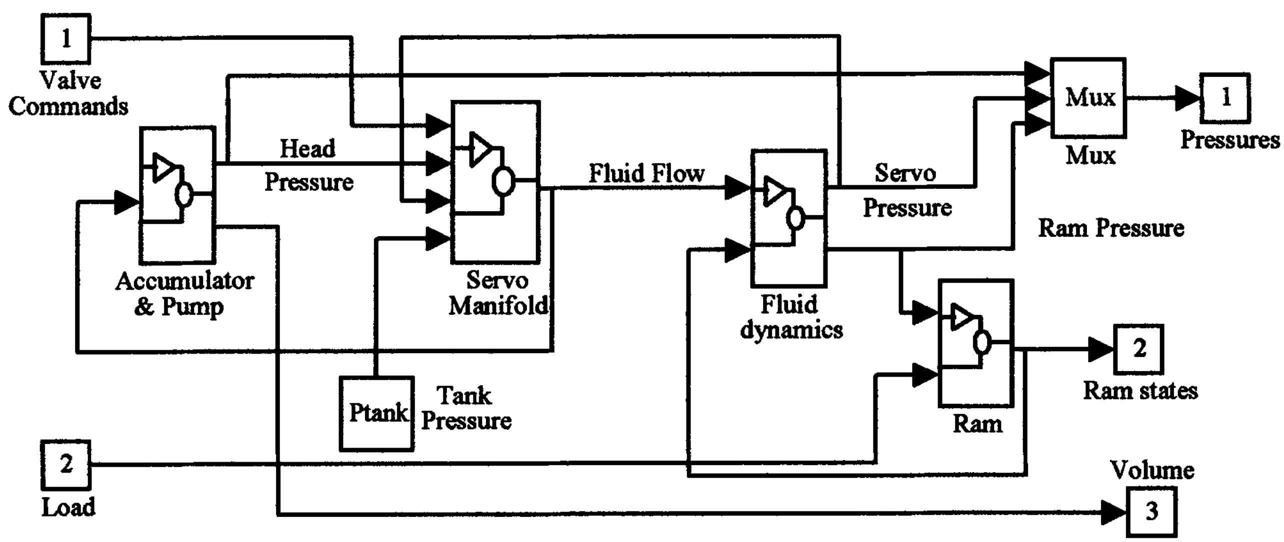

Fig. 5 Model of the forge press

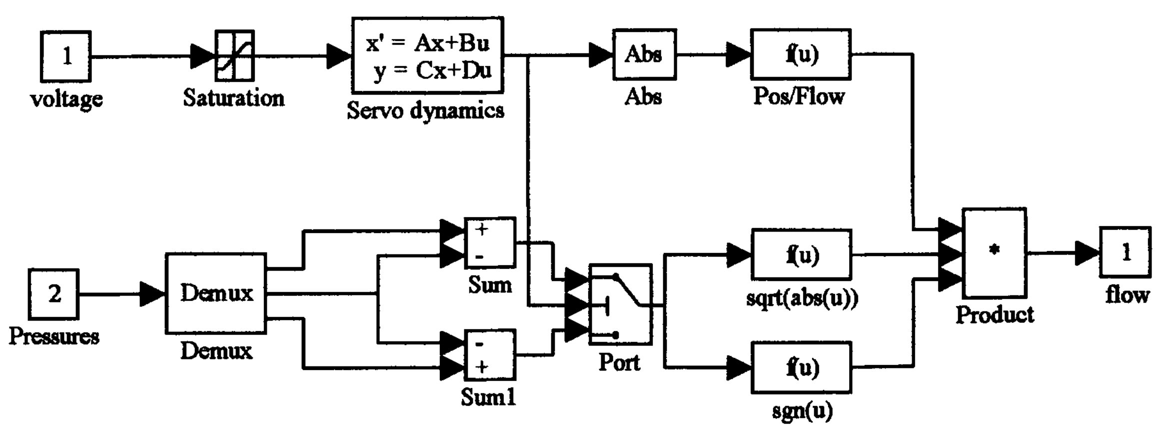

Fig. 6 Block diagram of a three-way servovalve

tween the forge press, the control computer, and the tooling package. Figure 5 shows the model of the forge press, including the accumulator, pump, servomanifold, fluid dynamic effects, and the ram. Figure 6 shows a block diagram for a three-way servovalve. Block diagrams for the rest of the press components and systems are similar.

In order to verify that the computer model is accurate, experimental data from a simple forging were recorded and compared with the corresponding simulation data. The experimental forging was a cylindrical upsetting of steel. The press was programmed to forge at a constant velocity of 12.7 mm/s. Plots of the experimental and simulated ram velocity and position are shown in Fig. 7 and 8, respectively. The irregularity in the experimental data of Fig. 7 is due to the method used by the press computer to estimate velocity. This behavior is not present in the actual press motion. Other than this effect, the simulation data closely resemble the experimental data. The data clearly reveal the ramping up and overshoot of the desired velocity. The brief change in velocity due to impact with the workpiece is clearly observed near 3.5 s.

Figure 9 shows the experimental and simulated results for the ram load as derived from the ram pressure measurement. The plot reveals that approximately 88.96 kN are needed to overcome the counterbalance and frictional forces. The load increases rapidly beginning at approximately 3.5 s. This corresponds to impact with the workpiece. Beginning at approximately 4 s, elastic deformation of the workpiece and tooling ceases and plastic deformation of the workpiece begins. It is clear from the data that the press was able to maintain the desired velocity under load, as long as the load was not increasing too rapidly.

The press computer does not record the commands to the hydraulic valves, but it is interesting to observe the plots of these commands and the corresponding valve flows from the simulation. These plots are shown in Fig. 10 and 11. Notice that a particular valve command does not always result in the same

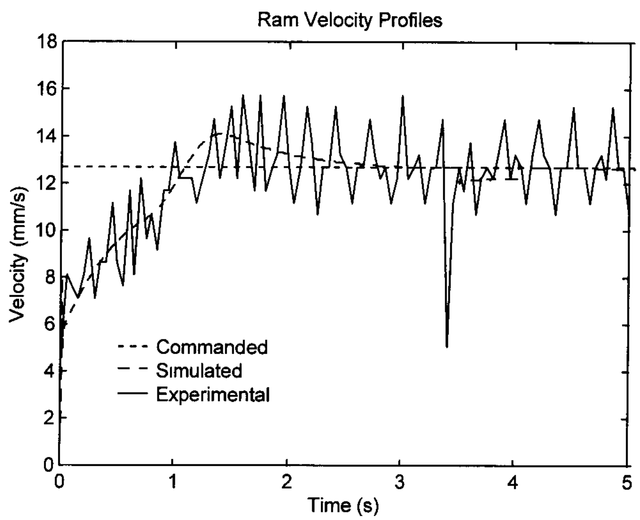

Fig. 7 Plots of experimental and simulated ram velocities

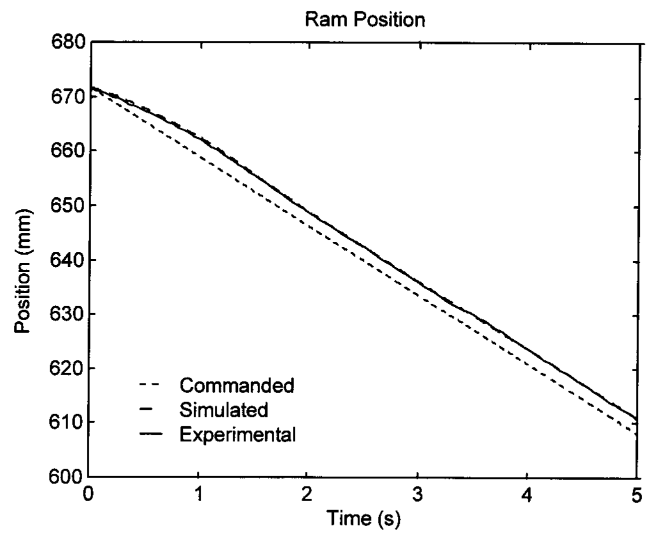

Fig. 8 Plots of experimental and simulated ram position

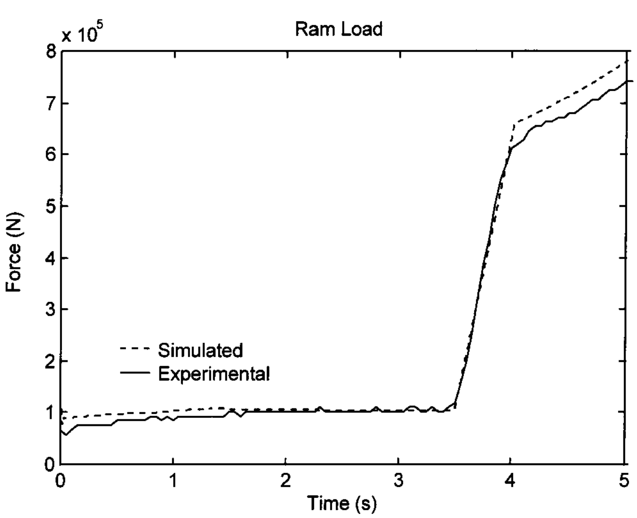

Fig. 9 Plots of experimental and simulated ram load

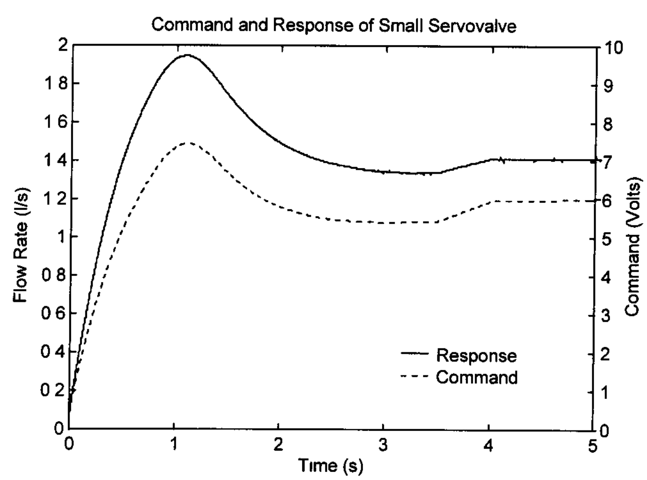

Fig. 10 Simulation results for small servovalve command and response

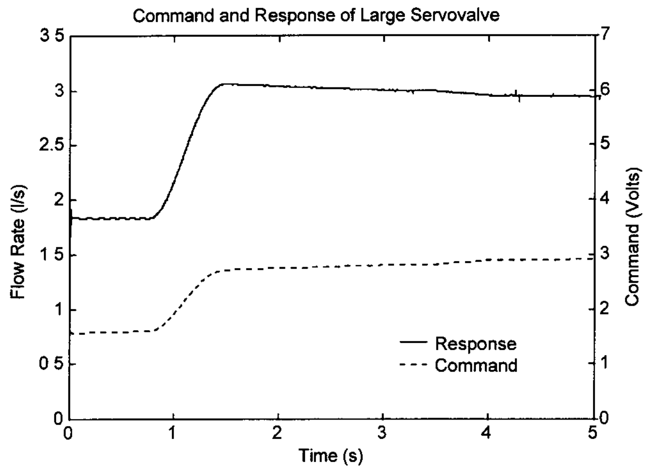

Fig. 11 Simulation results for large servovalve command and response

flow rate. This is due to the changing pressure across the servomanifold. Figure 11 clearly reveals that flow can decrease even as the command increases.

5. Conclusions

The use of graphics-based system modeling and simulation software greatly assists in making the modeling process systematic, self-documenting, accurate, and time efficient. The ability to quickly perform "what if" experiments with regard to different control strategies, sensors, actuators, and tooling aids the engineer in making informed decisions regarding potential changes to existing metal forming equipment and/or processing operations. For example, this approach makes it possible for the engineer to answer questions concerning the wisdom of possibly adding a sensor for direct measurement of ram velocity for feedback control of the ram velocity as opposed to simply using pressure feedback. As the need for precise control of ram velocity, as well as position, increases, the need for direct measurement of velocity as a feedback control signal will become more acute. Current technologies for controlling velocity (inversion of servovalve flow models combined with pressure measurements and numerical differentiation of displacement measurements) are not adequate for high performance.

Experience has shown that press operators must be able to customize the press control law in order to achieve the desired velocity profile for different forming operations. Having a custom control law for equipment that repeatedly makes the same part for several weeks or more is satisfactory, but if the same equipment is to be used for several different parts in a day or for custom small lots, repeatedly tuning the control law can waste a significant amount of time. Control laws need to be designed to be robust so that different loading conditions and velocity profiles can be handled successfully without customization.

It is anticipated that future work will include the use of the automatic code-generation features of modern simulation packages to integrate accurate equipment models into finite-element simulation software. This will further improve the accuracy of computer simulation of metal forming processes.

Acknowledgments

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