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SIMULATION OF A CAMLESS ENGINE VALVE ACTUATOR WITH MECHANICAL FEEDBACK

Norman K. Bucknor
Powertrain Systems Research Lab
GM Research & Development
30500 Mound Road, MC 480-106-252
Warren, MI 48090
norman.k.bucknor@gm.com

ABSTRACT

Fully flexible engine valve actuation systems are enablers for improvements in engine fuel consumption and power delivery, as well as the implementation of advanced combustion strategies like homogeneous charge compression ignition (HCCI). Hydraulically actuated valve actuation systems provide the greatest operating flexibility but have generally required precision flow control (i.e., servovalves) for viable operation while consuming more power than conventional cam-driven valvetrains. This paper describes an electrohydraulic fully flexible engine valve actuator with a mechanical feedback linkage between the engine valve and the spool in the hydraulic flow control valve. This feedback linkage is intended to simplify the control of the engine valve motion and eliminate the need for servovalve-class performance in the hydraulic control valve. The feedback mechanism reduces the control effort needed to operate the flow control valve since the spool position is not solely a function of the control input. With the assistance of mechanical feedback, the flow through the control valve is throttled in proportion to the engine valve motion. Thus, while throttling losses are not eliminated, there is no excessive flow throttling. This will have a beneficial impact on the energy consumption of the actuator. For preliminary study and validation of the concept, a model of the actuator was developed using ADAMS mechanical system simulation software and AMESim hydraulic simulation software. Results for the combined mechano-hydraulic model are presented to illustrate potential performance benefits and pitfalls of the concept, including effects of dimensional tolerances in the flow control valve. The simulation data was also used to size an electromechanical actuator that would be used to the flow control valve in conjunction with the feedback mechanism.

INTRODUCTION

Variable valve actuation is an enabling technology for improved engine performance. Optimization of engine valve lift, timing and duration can yield benefits in reduced fuel consumption, lower emissions and increased power output. Further benefits in engine performance can be obtained by altering the combustion process itself through novel engine operating modes like homogeneous charge compression ignition (HCCI). Variable valve actuation is a necessary part of HCCI operation if its full potential is to be developed. Of the various types of variable valve actuation systems, camless systems have the greatest flexibility, providing full control over lift, duration and timing. Camless electrohydraulic valve actuation is well known but as yet unavailable in production vehicles because of the significant technical challenges to implementation. One main drawback of these systems has been the requirement for relatively costly high performance flow control valves [1] to properly control engine valve motion, resulting in prohibitively expensive devices suitable only for experimental work. The flow-control technology has been either an aerospace-grade two-stage servovalve or a simpler one-stage spool valve combined with a powerful electromechanical actuator. One system design [2] attempted to address the cost/performance tradeoff by using a 'hydraulic pendulum' design with on/off flow control valves. The primary appeal of electrohydraulic valve actuation is that it is the only currently practical method for achieving fully flexible valve actuation. Practical electromagnetic camless systems are not capable of continuously variable lift without impractical power consumption penalties.

Built-in Feedback Control Mechanisms for Electrohydraulic Actuators

An alternative to an electrohydraulic system using high-performance flow-control valves is one based on built-in position feedback between the flow control valve and the engine valve. "Built-in" feedback control is defined here as an intrinsic function of the system, rather than an external control mechanism involving an external displacement sensor. Systems based on a hydraulic feedback circuit have been disclosed [3]. An alternative approach, uses a mechanical feedback device to tie the spool and engine-valve motions together. A mechanical feedback mechanism has appeared in some belt-type continuously variable transmission (CVT) control systems as a means of tying the desired speed-ratio command to the sheave position and the flow-control valve that controls the sheave movement [6]. Adapting this idea to engine-valve control, the desired engine-valve lift is tied to the actual engine-valve lift and the position of the flow-control valve. A key difference is that the CVT control requires relatively low actuator bandwidth, while engine valve operation occurs at frequencies up to 50-60Hz.

ELECTROHYDRAULIC VALVE ACTUATION (EHVA) WITH MECHANICAL FEEDBACK: OPERATING PRINCIPLES

A schematic of the electrohydraulic engine valve actuator [7] is shown in Fig. 1. The actuator has the following elements: a three-way, three position spool valve (1); an electromechanical control actuator (2); a hydraulic cylinder (3), containing a drive piston (4) to which the engine valve (5) is attached; an engine-valve return spring (6); and a rigid control lever (7) connected to the engine valve, the spool valve and the electromechanical actuator. Slotted rollers (9) connect the spool and engine valves to the control lever to provide the necessary degrees of relative freedom between the components.

The 'closed' position of the engine valve actuation system is shown in Fig. 2a. The spool valve (1) is in the 'null flow' position such that the spool blocks the high pressure port P from the flow port A, which connects to the hydraulic cylinder control chamber (8). The vent port T may be partially unblocked (underlapped spool) in this position to relieve fluid pressure in the control chamber. There is no flow to the hydraulic cylinder and any fluid trapped within the control chamber (8) is at vent pressure. The force of the return spring (6) keeps the engine valve (5) closed. This configuration permits the feedback arm to rest against a mechanical stop (10) when the engine valve is closed. The stop precludes the need to use a spring or similar means to support the arm when the control actuator is de-energized.

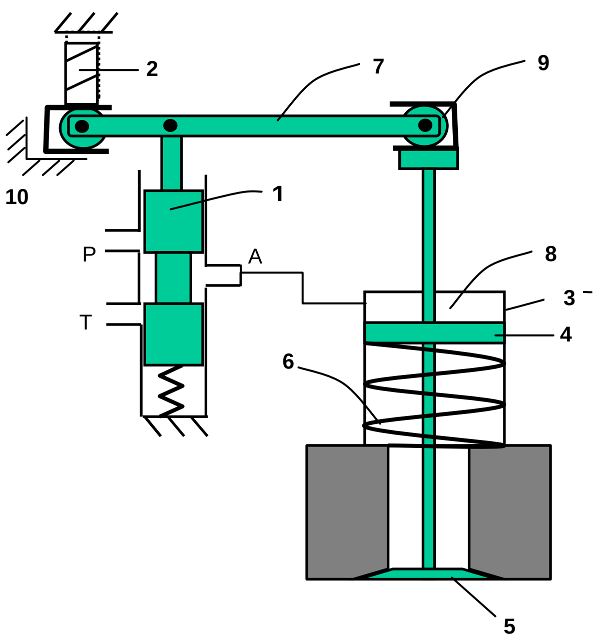

Figure 1 Electrohydraulic Engine Valve

To open the engine valve, the control actuator (2) retracts under electronic control, forcing the control lever upward in Fig. 2b. Because the return-spring holds the engine valve in place, the control lever initially pivots about the attachment point C1, between the control lever and the engine valve. The control lever pulls the spool upward to open port P, causing high pressure fluid to flow into the control chamber (8) while blocking the vent port T. The pressure rise in the control chamber causes the engine valve to open against the return-spring force. As the engine valve moves downward, the control lever now pivots about the control actuator connection point C2, tending to move the spool toward the 'null flow' position (Fig. 2c). Once the control actuator reaches its commanded position, the engine valve motion will stop when the control lever rotates to move the spool to block both high and low pressure ports (Fig. 2d). The control lever motion serves as a mechanical feedback loop between the spool and engine valve motions, regulating the flow into the control chamber such that the engine valve motion is an amplification of the control actuator stroke, as shown in Fig. 2d. Ideally, the control actuator velocity is just sufficient to cause the engine valve motion to track the control actuator motion with a slight lag. This prevents the engine valve from overshooting the desired lift position. If an overshoot occurs, the control lever motion will cause the spool valve to move past the 'null flow' position to the vent position, causing some fluid to exhaust from the control chamber. The engine valve will move upward, moving the spool toward the 'null flow' position. This oscillatory motion is eventually damped out by internal friction and leakage. At the 'null flow' position, the fluid in the control chamber is hydraulically locked and the engine valve stays at the commanded lift. Note that the commanded lift L is approximately a function of the stroke s of the point C2:

$$L = (l_2 / l_1)s \tag{1}$$

The relationship given by Eq. (1) is only approximate because of the roll/slide joints connecting the feedback arm to the spool and engine valve. Consequently, the effective length of the feedback arm changes as the arm pivots from zero lift through full lift. In the case of the prototype design described in the following section, the difference in length is up to 4%.

MODELING THE EHVA CONCEPT

The EHVA concept was modeled to evaluate its suitability for high-speed operation. In particular, it was desirable to determine the control actuator forces required to drive the device, given the inertias of the various mechanical

l_1 = 6 mm, l_2 = 34 mm

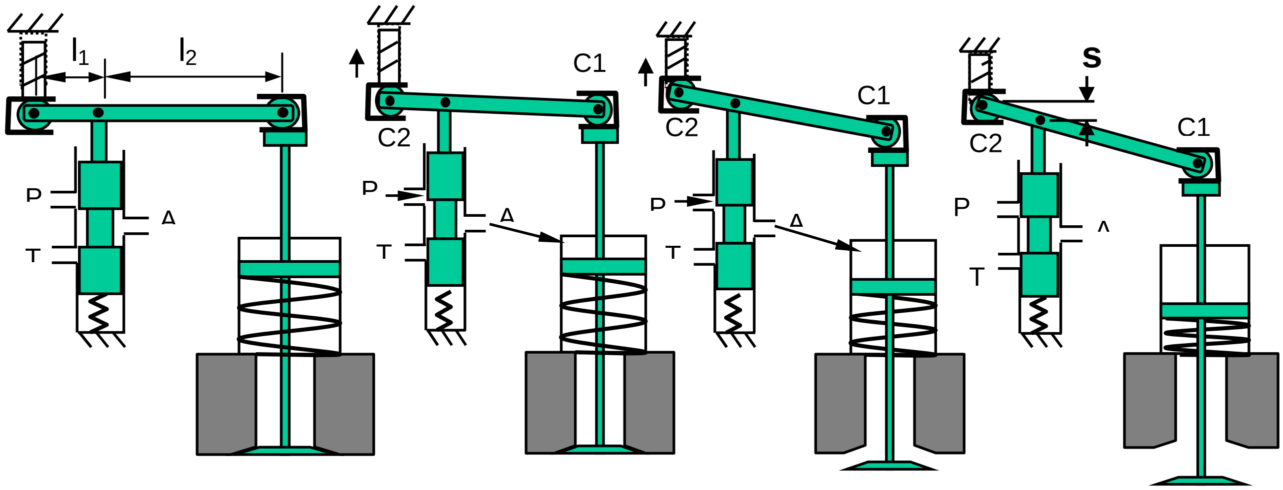

Fig. 2a Engine valve closed

Fig. 2b Control actuator motion (retracting) starts

Fig. 2c Engine valve opening

Fig. 2d Engine Valve at Commanded lift ([l_2/l_1] X **s**)

Figure 2 Operation for Engine Valve Opening (Alternative Construction)

components. Very large control forces would essentially invalidate the concept.

Mechanical System Sub-model

The mechanical portion of the EHVA device was modeled using commercially available multi-body dynamics simulation software. The mechanical model consists of all components except the hydraulic elements. The hydraulic circuit is modeled separately in another commercially available general-purpose simulation code with special hydraulic and mechanical element libraries. The hydraulic circuit model is then integrated with the mechanical model as a submodule. The hydraulic submodel provides a computation of the pressure in the hydraulic cylinder and passes it to the mechanical model. The mechanical model passes the current position of the spool valve to hydraulic model, as well as the current valve seating force if needed. The mechanical model is shown in Fig. 3. It consists of 5 parts with masses, moments of inertia and geometry data approximately matching a prototype hardware design[1]. The EHVA is shown in the deactivated position in Fig. 3, with the engine valve closed. The green link represents the control lever (7) in Fig. 2, the gold-colored link represents the spool (1) and the aqua-colored link represents the engine valve and piston assembly (4,5). The red link attached to the control arm represents a linear actuator. Roll-slide joints connect the engine valve/piston assembly and the linear actuator to the control lever. The control lever in turn is pinned to the spool. The spool, engine valve assembly and linear actuator all are constrained to move vertically in the x-y plane.

The two sets of arrows near the circular discs indicate contact force elements between links. The aqua disk in the lower right of the figure is an extension of the engine-valve, and is used to model contact between the engine valve and the stops at the extremes of its motion range. The stops (the gray disks) represent the engine valve seat and the engine valve guide stop.

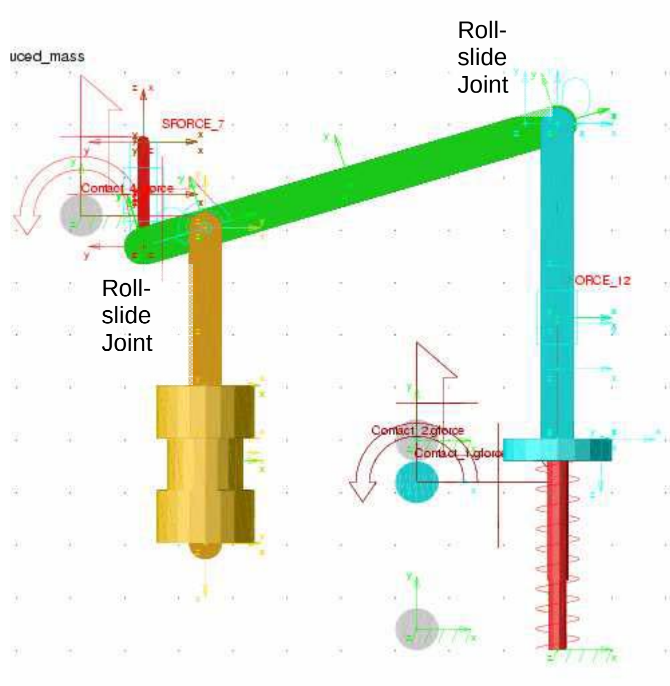

Figure 3 EHVA Multi-body Model

[1] The design differs from Fig. 2 in that the rest position of the control arm is not horizontal. This configuration prevents interference between the control arm and the roll/slide joint structures near the extremes of the control arm motion.

The contact forces are only active if the gray and aqua disks interfere, indicating an interpenetration of bodies. This will generate a contact force based on a nonlinear spring model. The gray disk in the upper left of the figure represents a stop for the linear actuator on which it rests when the actuator is deactivated.

In simulation, the linear actuator is assumed to be a voice-coil actuator with certain force characteristics. The motion of the actuator can be controlled in one of two ways: 1) apply a force or 2) specify the motion directly. The value of the force is generated by a user subroutine written in FORTRAN that implements a PD feedback controller. The desired motion is specified and the subroutine returns force values based on the error in the speed and position of the actuator part. Alternatively, the motion of the linear actuator part can also be specified directly, with the required force to achieve that trajectory computed.

Hydraulic System Sub-model

The hydraulic circuit model is show in Fig. 4. The blue jack connected to the green mass M represents the engine valve/piston assembly. The spool valve is represented by the back-to-back spool components in the left half of the figure. The interface block "Force – Force to ADAMS" passes the current value of the net force (N) exerted by the jack on the mass M to the mechanical model, where it is applied to the engine valve. The net force is the algebraic sum of the hydraulic force acting on the piston, the spring force and any damping forces. As a consequence, the spring shown in the mechanical model in Fig. 3 is deactivated and does not exert a f orce on the valve. The partly mislabeled interface element "Displacement and Velocity from ADAMS" passes the spool position and the vertical component of the force acting at the roll-slide joint between the engine valve/piston assembly and the control lever

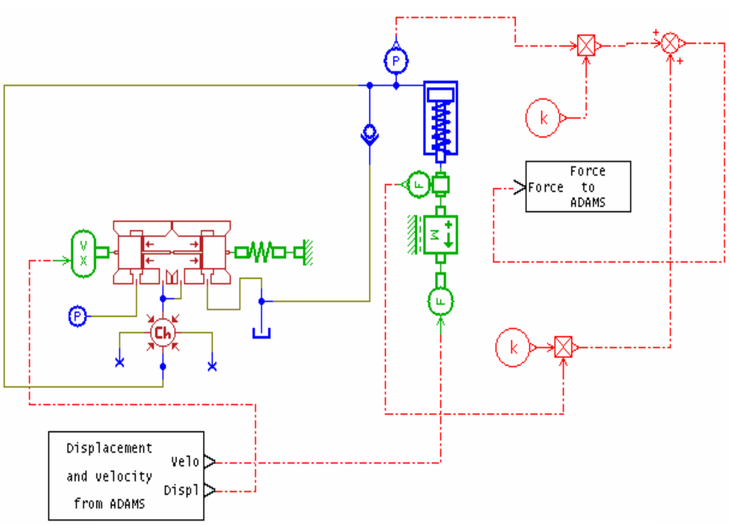

Figure 4 Hydraulic Circuit Model

from the mechanical model to the hydraulic model. This allows the reaction force at the control lever joint to be incorporated into the hydraulic model as an external force acting on the engine valve piston. This is necessary because the external forces will affect the motion of the engine valve/piston assembly and AMESim only models the hydraulic and inertia forces. The displacement of the spool valve directly controls the flow into and out of the jack chamber.

Other elements of the circuit are a high-pressure source 'P,' connected to the high-pressure port of the spool valve and a return line (indicated by the tank symbol). The center port is connected to a hydraulic volume to capture hydraulic pressure dynamics in the jack chamber and the connecting channel to the spool valve. The return line is kept at low pressure, about 3Bar gauge. The hydraulic model parameters allow the modeling of spool valve overlap and underlap (see Fig. 5). The amount of overlap will determine the deadband in the flow control response of the spool valve, while the amount of underlap will affect seating and dwell control of the engine valve.

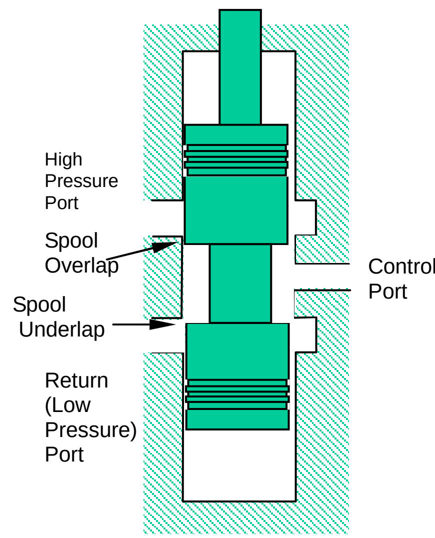

Fig.5 Spool Valve Overlap & Underlap (EHVA Deactivated Position)

Simulation

The simulation is built by creating a custom executable composed of general state equations (GSE) exported from the hydraulic modeling software to the mechanical model. The hydraulic circuit model equations then become a part of the complete executable. The interface variables between the programs are actually variables passed between custom routines within the executable file. To accommodate the interface variables within the mechanical model, state variables must be created corresponding to the interface inputs and outputs in the hydraulic model. The state variables are then associated with the appropriate mechanical model elements, with data being read from or assigned to these variables according to the integration of the GSEs imported from the hydraulic model. One drawback of this setup is that there are two models for the engine valve/piston assembly – one originating from the mechanical model and the other from the hydraulic piston/mass model. Thus there is one more state variable than is absolutely needed.

RESULTS & DISCUSSION

EHVA PERFORMANCE SIMULATIONS

Control Actuator Performance Envelope: Kinematically Prescribed Displacement

The performance envelope for the control actuator can be roughly estimated from the results of a simulation with a prescribed motion for the control actuator (element 2 in Fig. 1). This can be thought of as a worst-case scenario in terms of the actuator control force requirements. In this simulation mode, the desired trajectory for the control actuator is the actual trajectory and the required force to achieve that trajectory is then computed. The rise and fall segments of the trajectory are generated using a Haversine function [4]. Typical results are shown in Fig. 6 with plots of the control actuator displacement ,

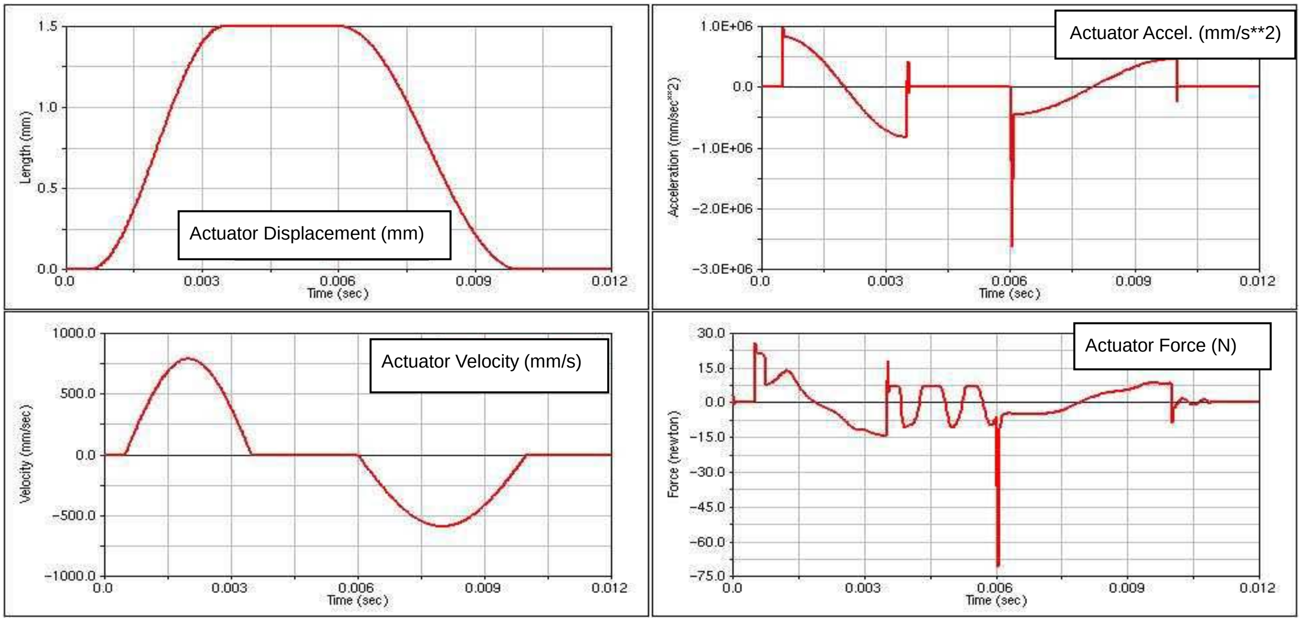

Figure 6 Control Actuator Force with Control Actuator Displacement

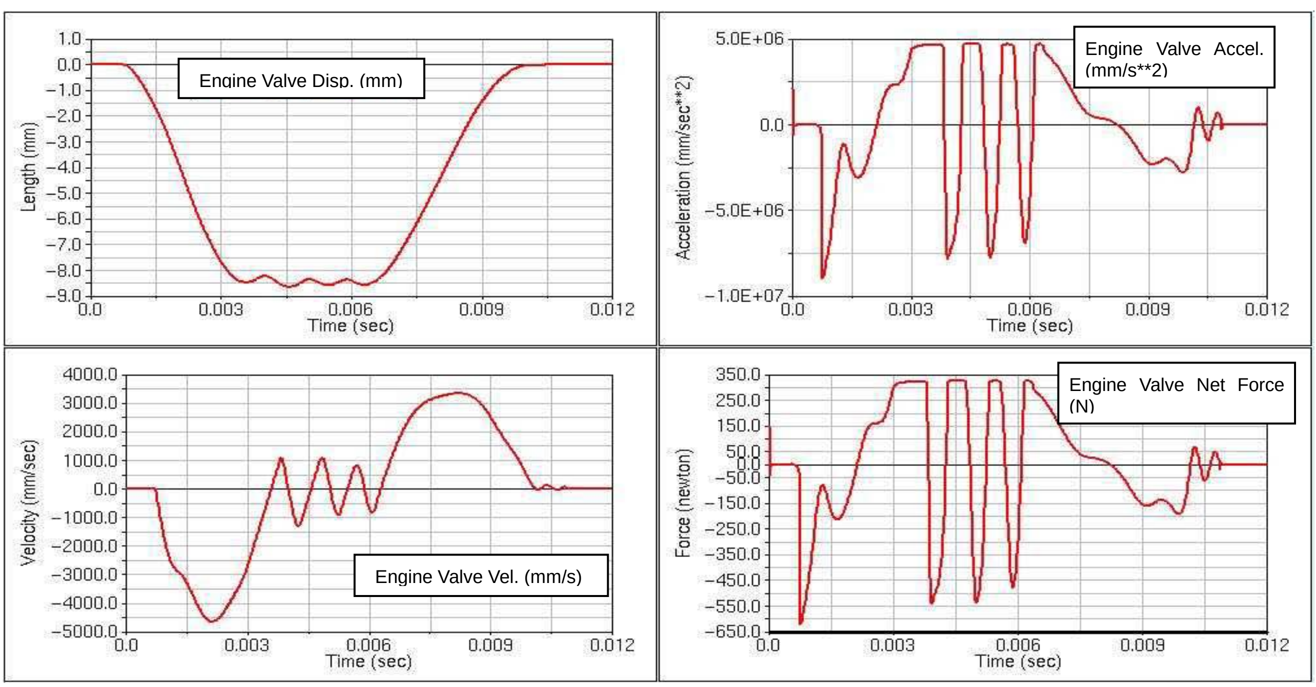

Figure 7 Engine Valve Motion with Control Actuator Displacement Prescribed

velocity, acceleration and force required to achieve the actuator displacement. The hydraulic system pressure is 204Bar (3000 psig). The actuator control force has a peak magnitude of about 71N at one of the acceleration discontinuities. This is clearly an unrealistic result due to the discontinuities in the velocity profile forced by strict tracking of the control trajectory. Excluding that point, the maximum force is approximately 25N. Fig. 7 shows plots of the corresponding engine valve displacement, velocity, acceleration and net force. Note the oscillation of the valve around the commanded lift position during the dwell portion of the motion. Since the hydraulic chamber is sealed from both the high and low pressure ports as the engine valve reaches the desired lift position, the column of fluid trapped in the chamber acts like a spring counterbalancing the mechanical engine valve return spring. Due to inertia, the engine valve overshoots the commanded lift position, causing the fluid pressure in the sealed chamber to fall such that the net force on the engine valve is primarily the opposing spring force. The spring force reverses the valve motion, accelerating it in the opposite direction until the fluid pressure again increases in the chamber, setting up an oscillatory motion around the desired lift position. As the control arm oscillates, the high pressure and

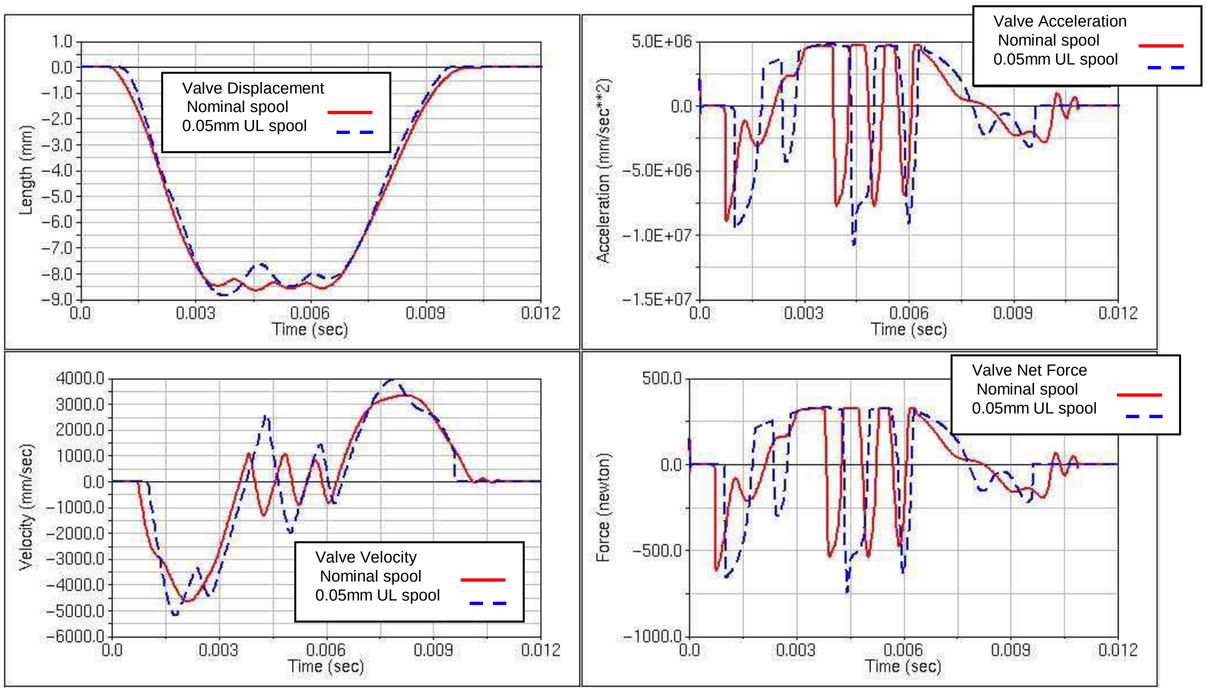

Figure 8 Effect of 0.05mm Spool Overlap and Underlap on High & Low Pressure Ports Respectively

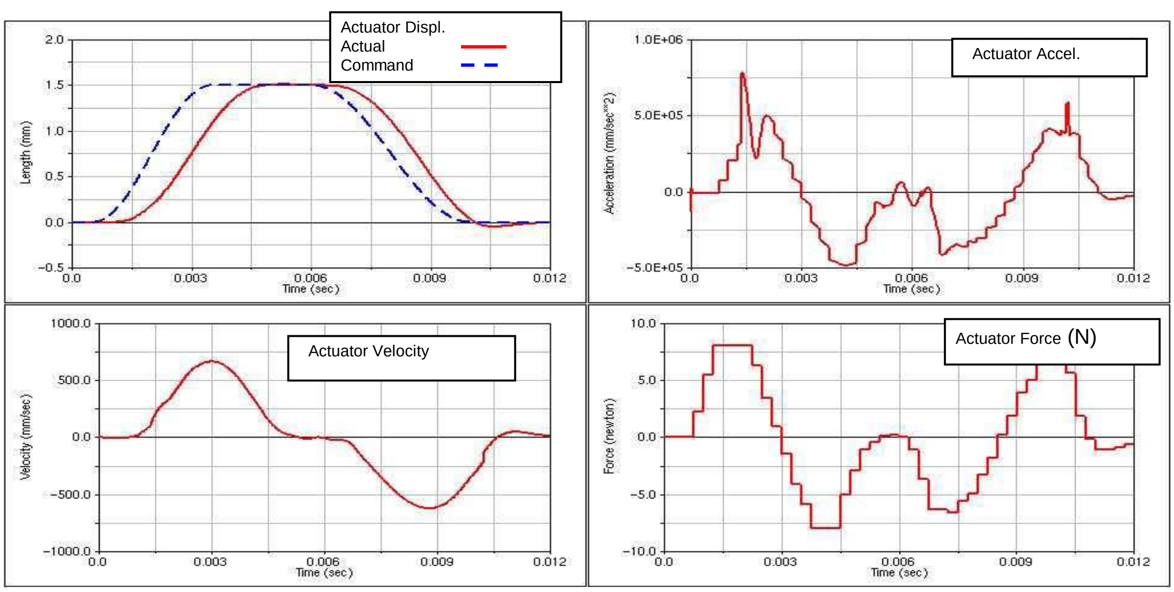

Figure 9 Control Actuator Response with PD Feedback Control of the Actuator Displacement

low-pressure ports are also alternating between the open and closed position with approximately no net gain or loss of fluid in the piston chamber.

The resulting engine valve motion assumes a perfect spool valve with zero overlap or underlap of the flow ports. In reality, it is unlikely that such a situation would occur because of the imprecision inherent in any machining or manufacturing process. The effect of manufacturing variation on the spool valve and flow ports is critical to the performance of the mechanism and must be taken into account during the design phase for both the hardware and the actuator controls. Based on stackup analyses during prototype design, it is estimated that spool overlap at the high-pressure port can be controlled to within 0.05mm to 0.1mm, with a maximum value not exceeding 0.1mm in the worst case. By design, the return port is underlapped by the spool (Fig. 5), with an underlap varying between 0.05mm and 0.1mm. This ensures that the control chamber will always vent with the engine valve in the closed position. For the same control actuator displacement shown in Fig. 6, the simulation was rerun with a spool overlap of 0.05mm at the high-pressure port and 0.05mm underlap at the return port. The results are shown in Fig. 8, with the affected engine valve results in dotted blue lines, overlaying the results of Fig. 7. Two significant effects are the delayed opening of the engine valve due to spool overlap at the high-pressure port (i.e. the control actuator has to move further to open the port) and early closing of the engine valve due to the underlapped condition at the return port. In the latter case, all the fluid pressure in the control chamber is vented to the return line before the engine valve can seat. Consequently, there is no braking force provided by the fluid against the engine valve return spring and the result is a high impact velocity of about 1000 mm/s.

Control Actuator Force Determined by PD Feedback Controller

A more realistic EHVA operational scenario is that the actuator force will be modulated by a control system monitoring the desired control actuator trajectory and supplying corrective input signals to minimize the tracking error. In the simulation, a proportional/derivative controller is used, with a simulated error sampling time to mimic the effect of using a data acquisition system. Thus, the control input to the actuator is a sample-and-hold signal. Compared to a continuous control signal this should produce a more conservative estimate of the potential system performance. Figure 9 shows results for the same actuator trajectory command as in Fig. 6, assuming zero spool overlap and underlap. As in the previous examples, the hydraulic pressure is 204 Bar. The sampling rate is 4kHz. In this example, the output force of the actuator has been clipped at 8N, reflecting about 50% of the peak force available in the voice coil actuator selected for prototype testing. This value, together with the controller gains and sampling time was arrived at via trial and error, yielding the best simulation results for tracking the desired actuator trajectory. A conservative estimate for the maximum force available also allows for some force reserve to overcome stiction and other unpredictable forces acting on the EHVA control mechanism.

The effect of a sample-and-hold control signal on the output actuator force is shown in the lower right-hand plot in Fig. 9. Referring to the upper left-hand plot, note that the actuator displacement (solid line) lags the desired trajectory (dashed line) because of the constraint on the actuator force output. Because of the sample and hold signal, a larger initial force would result in a slightly smaller lag but a larger overshoot near the desired peak displacement. Furthermore, a larger force would necessitate a bigger, more energy-consuming actuator. As can be seen from Fig. 6, a spike of 20N would be required to launch the actuator with sufficient acceleration to match the desired trajectory. While actuators capable of this output exist, they increase the packaging envelope and need much more power than the actuator chosen for prototype testing. Thus, a smaller device is preferred, if not ideal.

As the spool uncovers the high-pressure port, the engine valve begins to move downward as high-pressure fluid flows into the piston chamber. The net effect is to rotate the control lever about the spool attachment point in a clockwise direction, momentarily accelerating the actuator upward. This acceleration spike can be seen in the upper right-hand plot of Fig. 9, at about 1.5ms simulation time. Using feedback control, the desired actuator lift is achieved, but the total duration is slightly shorter than the actual time desired. On the “return” portion of the curve (i.e., the drop back to zero lift), the actuator displacement again lags the desired profile and there is undershoot at the end of the motion. In hardware this would be registered as an impact at the actuator stop. In the simulation a relatively soft contact cushion (10E4N/mm stiffness) is assumed. Control actuator undershoot indicates a high engine-valve seating velocity due to premature fluid-pressure relief at the spool valve return port. The control actuator determines the degree to which the spool land uncovers the relief port, producing a throttling effect on the fluid passing from the piston to the return line when the engine valve nears the seated position. The throttling effect serves as a brake on the engine valve motion by providing a controlled depressurization of the fluid trapped in the piston chamber. If the control actuator advances too rapidly relative to the engine valve return motion, the spool land will uncover more of the return port than is desirable. The braking or cushioning effect will be lost before the engine valve is seated, causing it to slam into the valve seat due to the return spring force. This is reflected in the engine-valve velocity discontinuity at seating shown in the lower left-hand plot of Fig. 10 (red curve), corresponding to the actuator motion shown in Fig. 9. The impact velocity is approximately 1400 mm/s. The curves for the prescribed-motion case are shown in dashed lines. In the feedback-control case, there is comparatively little oscillation during the dwell phase because of the more gradual changes in control actuator deceleration as the engine valve reaches the peak lift point. This limits excitation of the valve and spring mass at the dwell position.

The engine-valve seating velocity can be reduced by modifying the trajectory of the control actuator near the seating position, preventing undershoot of the desired trajectory. To achieve this, a compensation factor is applied to the desired control trajectory, adding a positive offset to the final position at the end of the motion. In trying to track the slightly offset trajectory, the control actuator's true displacement more closely matches the original desired final position. This can also be

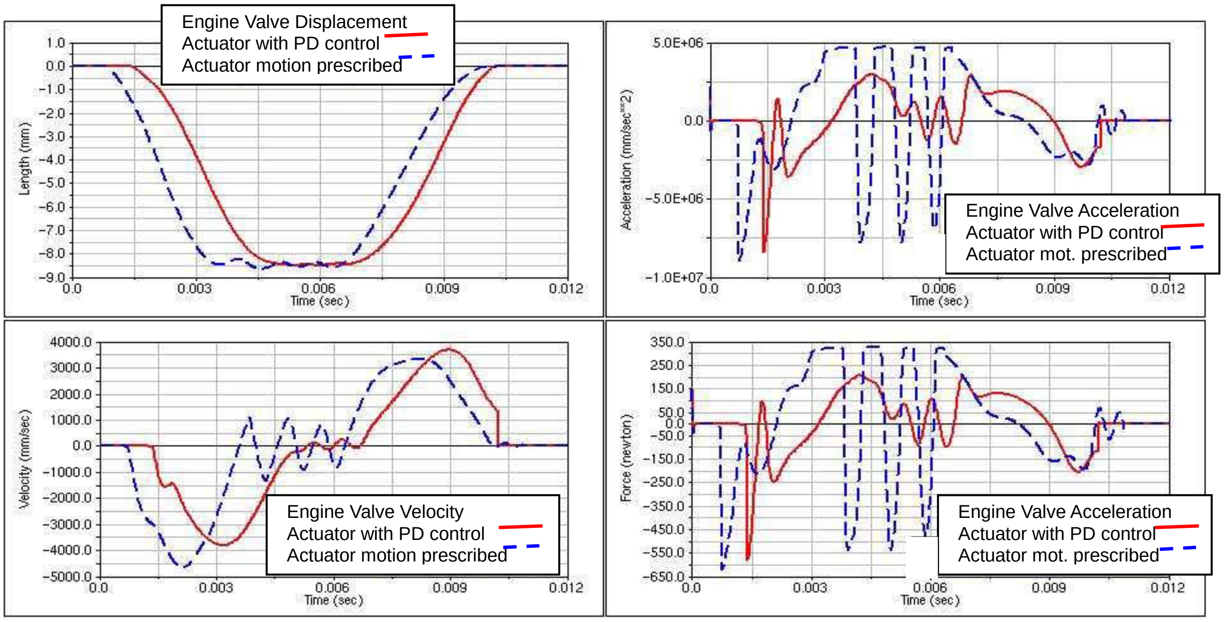

Figure 10 Engine Valve Motion with PD Feedback Control of the Actuator Displacement

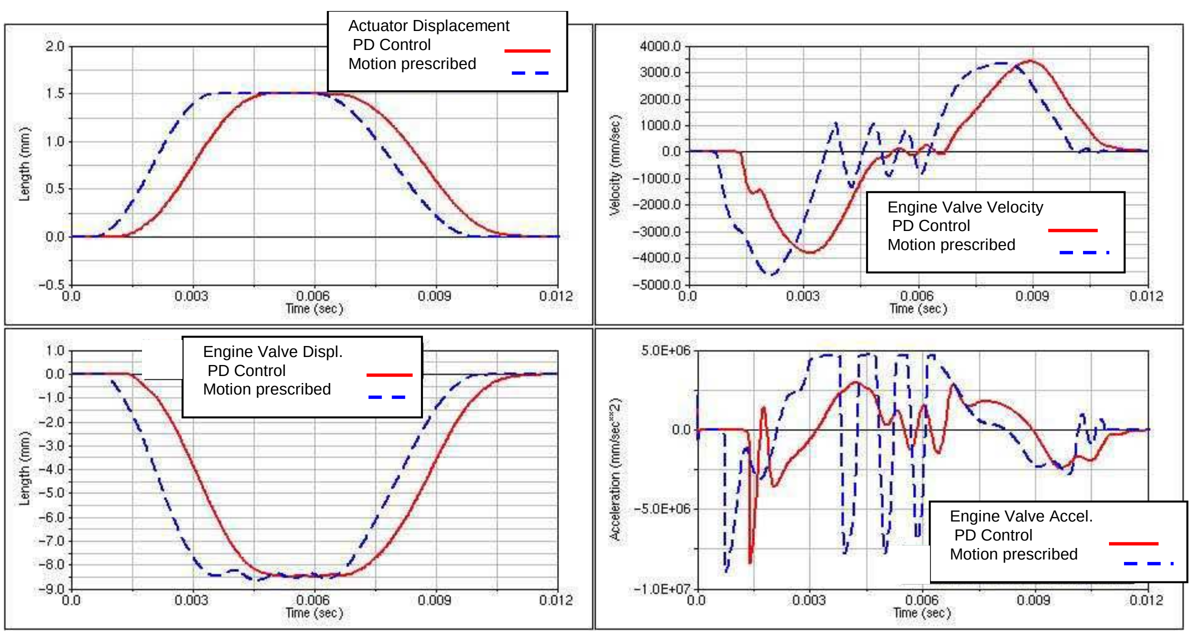

Figure 11 Engine Valve Motion with PD Feedback Control & Offset Final Position of the Actuator Displacement

thought of as trying to halt the motion of the control actuator short of the final desired position. If this can be accomplished, a second command can then be given to pull the actuator against its stop if it does not in fact reach the rest position. The advantage of braking the control actuator before it reaches its stop is that a "soft landing" for the engine valve is more easily attainable since the offset position will cause the spool valve to seal the return port before the engine valve seats (due to the mechanical feedback connection), leaving some fluid in the piston chamber to cushion the engine-valve motion.

Using an offset of 0.075 mm, the engine-valve seating velocity is reduced to less than 30 mm/s, as shown in Fig. 11.

Note that there is negligible difference between the desired and actual final positions of the control actuator. On the other hand, if an offset of 0.05 mm is assumed, a seating velocity of approximately 500 mm/s results. The need to introduce this actuator displacement offset makes the control system less robust since different offsets are required for different engine valve lifts. A more sophisticated control approach may be required to achieve the necessary reliability in soft-seating control.

Effect of Spool Valve Overlap and Underlap on EHVA performance

Generally, it cannot be expected that the flow control spool valve will be perfectly centered with no underlap or overlap. For this analysis it is assumed that there is a minimum overlap on the high-pressure port land and a minimum underlap on the return-port land when the EHVA is deactivated. This will minimize leakage of fluid from the high-pressure port to the piston chamber and return port, while also ensuring that the system is also depressurized. Given manufacturability constraints, it should be easier to produce a spool that has these characteristics since the rest position of the spool can be easily adjusted by shimming the stop for the control actuator. Examining the feedback-control case with a spool overlap of 0.05 mm at the high-pressure port and 0.05mm underlap at the low-pressure port, an overlap at the high-pressure port results in a delay in the opening of that port, so that the net commanded lift is the actuator stroke minus the overlap. Once amplified, this creates a smaller peak lift (8.00 mm vs. 8.50 mm). The underlapped return port results in loss of the fluid cushion at seating, with a resulting impact velocity of 750 mm/s. Inserting an offset in the control actuator trajectory at seating, lowers the valve impact velocity to under 100mm/s.

EHVA FLOW RATES AND ESTIMATED POWER CONSUMPTION

Typical control pressures, flow rates and hydraulic pump power consumption for the actuator are shown in Fig. 12, corresponding to the high-lift valve event shown in Fig. 13. In this simulation, the engine-valve lift profile mimics a conventional mechanical valvetrain profile, using an production valve lift curve as a template. Because of spool overlap and underlap, the actual duration of the resulting valve events is slightly shorter than the commanded duration of 9ms. For the intake valve lift curve used, the commanded event duration is 150 cam angle degrees or approximately 5560 engine RPM for a 9 ms event.

Referring to Fig. 12, the control-pressure droops in the top plot whenever the high-pressure fluid flow rate falls below the corresponding engine-valve velocity. This occurs when the control actuator decelerates, causing the flow-area at the high-pressure port to decrease as the engine valve continues its opening motion under its own momentum, with the return spring acting to decelerate the valve. At several points, the change in control chamber volume is sufficient to drop the control pressure to zero. Due to valve momentum, the high-pressure port flow is cut off before the engine valve reaches its peak lift (at approx. 0.0056s) and the feedback mechanism automatically causes the relief port to open and flow reversal to begin at approx. 0.0053ms. This helps to complete the valve deceleration to zero speed prior to starting the closing motion. The instantaneous hydraulic power expended (in Watts) is shown in the bottom plot of Fig. 12 for a supply pressure of 204Bar (3000psi). The peak value is 3.2kW. The average power required is approximately 1.92kW. A pair of intake valves would consume, if driven by separate actuators, about 3.9kW during operation[2].

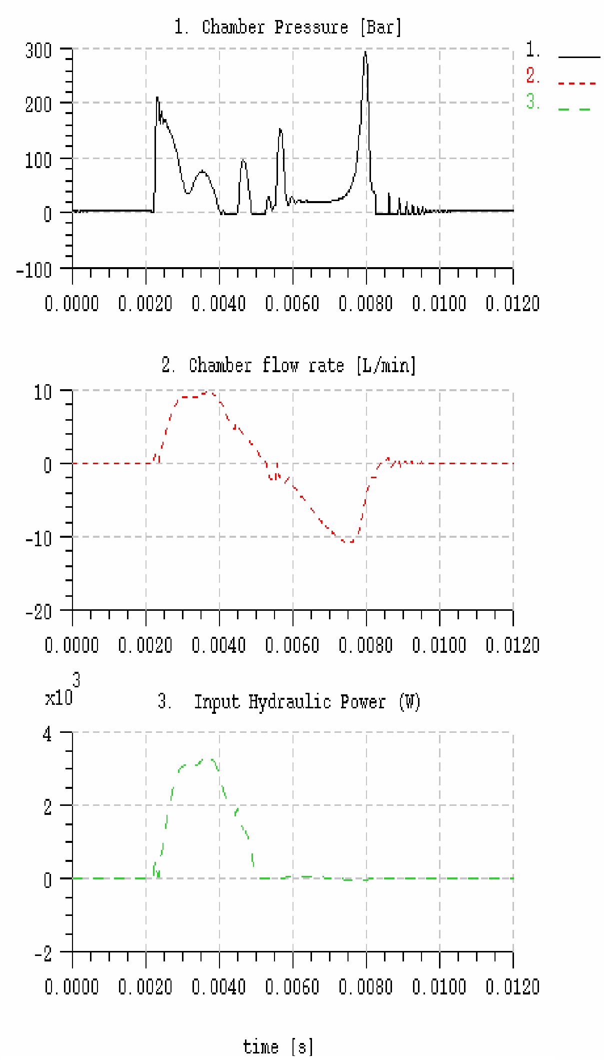

Fig. 12 EHVA Chamber Pressure & Flow Rate, Pump power corresponding to Fig.13

[2] Exhaust valve actuators would likely require more power due to opening against cylinder combustion pressure.

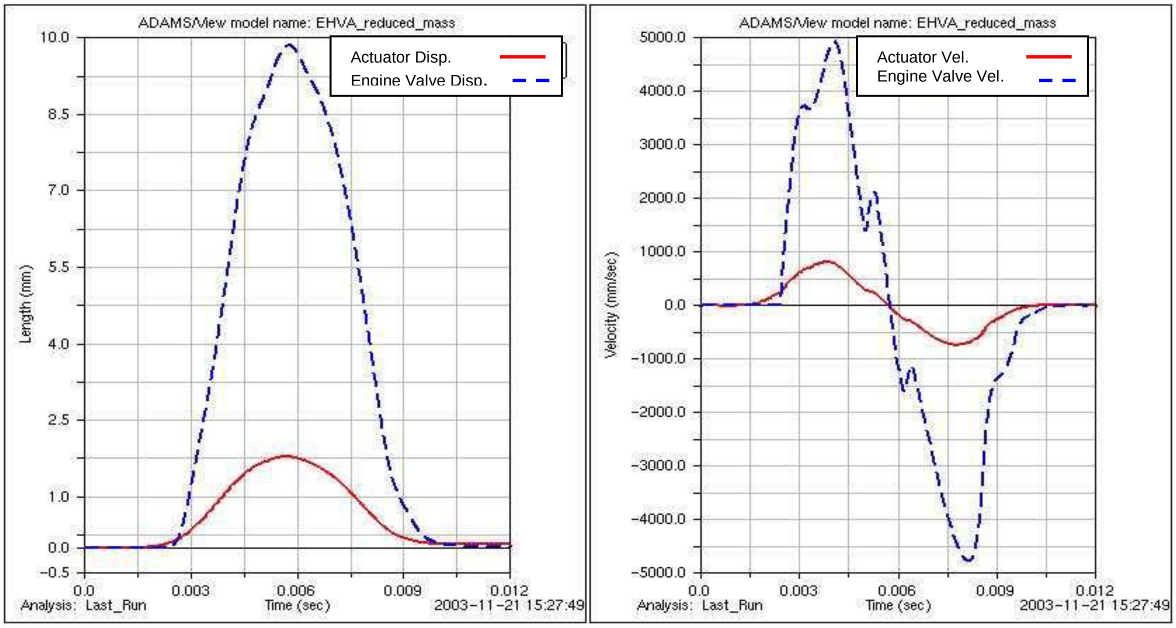

Fig. 13 EHVA Control Actuator & Engine Valve Displacement and Velocity at High Lift, High Engine Speed

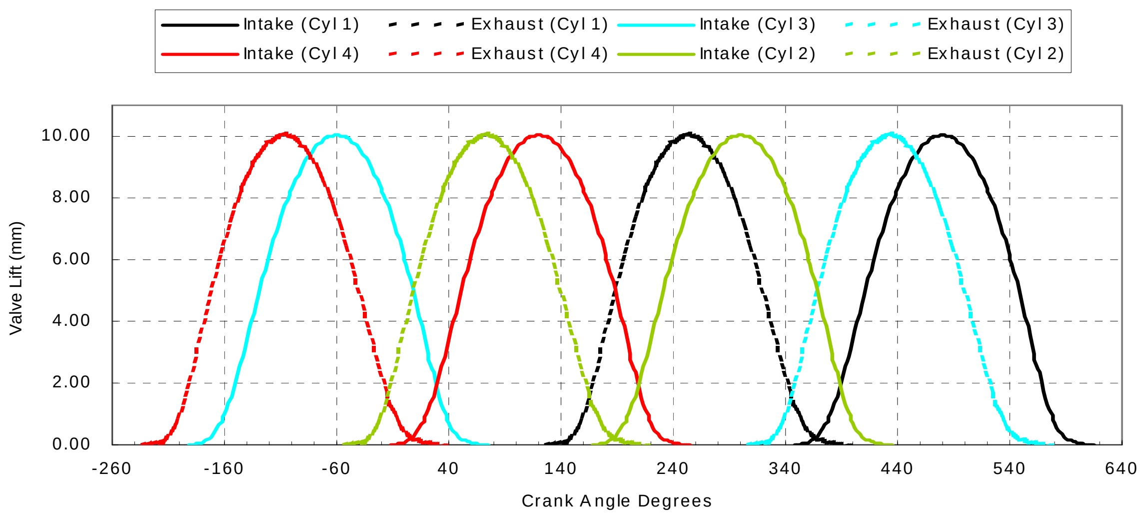

Fig. 14 4 Cylinder Engine Valve Event Timing

For a 4 cylinder, 16-valve camless engine, the total hydraulic power required may be roughly estimated from the power needed per valve pair. In Fig. 14, it can be seen that the exhaust and intake valve openings substantially overlap for pairs of cylinders (i.e., exhaust cyl. 3 & intake cyl.1, 4 & 3, 2 & 4, 1 & 2). In each case, the intake-valve pair lags the exhaust-valve pair by 41 crank angle degrees, or 20.5 cam angle degrees. Total cam rotation from zero to peak lift is 68.5 degrees, so that if we use the same value for average flow rate per valve[3], there will be 70% overlap in flow demand from the exhaust and intake valves. Consequently, the average flow required is 170% of the flow to a pair of valves on each cylinder and the average hydraulic power is 170% of the value for a valve pair. Since no more than 2 pairs of valves are simultaneously opening at any time, the average continuous power consumption for the entire valvetrain will be approximately the same as for any 2 pairs of valves, taking the operating overlap between pairs into account. Note that using a constant value of 170% overestimates the average power demand when there is no operating overlap. Estimated power consumption is then 6.63kW in the high-lift, high-speed case, which would be the worst-case scenario. At moderate to low lift, power consumption is significantly lower. It should be stressed

[3] The exhaust valve flow rate will be higher due to the need for a larger piston area to overcome cylinder pressure.

that these values are approximate, assuming a perfect hydraulic pressure source. The effects of pump efficiency can be estimated by applying the appropriate scaling factors for pump mechanical and volumetric efficiency.

The electrical power consumed by the EHVA is dependent on the control actuator used. As an example a voice-coil linear actuator capable of 15N thrust expends about 43W to generate that thrust. In the high-lift case (Fig. 13), the maximum thrust was less than 10N. Conservatively assuming a constant 10N thrust, the power consumed would be approximately 29W. Thus the hydraulic power consumption in the high lift case far exceeds the electrical power required. At low lift, the maximum actuator thrust is less than 2.5N, with maximum electrical power consumption of approximately 7.17W per actuator (again assuming constant thrust).

The valvetrain power consumption in a 4 cylinder conventional cam-driven engine is estimated from the following sample data: the cam-driven valvetrain FMEP is approximately 16kPa at 556RPM and 22.5kPa at 5560RPM. The FMEP values are converted to equivalent kW using the engine-power formula from [5, p. 418]: 0.16kW at 556 RPM and 2.29kW at 5560RPM. The electrohydraulic valve actuator suffers from much higher energy consumption at high-load conditions relative to the cam-driven system because of the high hydraulic system supply pressure. Consumption at high load may be reduced by lowering the supply pressure and/or the flow rate but this strategy is constrained by the degree to which changes in system parameters such as the drive-piston area can be made while meeting performance requirements.

SUMMARY

A design for an electrohydraulic engine valve actuator with mechanical feedback control was proposed as a means of providing high-performance valve actuation with small electromechanical control actuators. Simulation studies were carried out to determine the efficacy of the design in meeting the required engine valve operating requirements. The simulations show that the use of the mechanical feedback control allows the use of a control actuator with a maximum force output as low as 8N since there is no need to rapidly and directly move the spool back and forth to correct the engine valve position. This also tends to minimize the total flow into and out of the system during a lift event. Using a 4kHz PD position controller for the control actuator, it was also shown that soft-landing control of the engine valve appears feasible. However, the efficacy of the PD control may be questionable for robust soft seating control since there is a need to modify the controller parameters for different lift heights. A more sophisticated control algorithm may be required to achieve required reliability levels. One means of achieving this could be to prematurely close the return port by incorporating a spool overlap at the return port.

The effect of valve manufacturing errors (spool overlap and underlap) was also studied. As expected, spool overlaps introduce additional performance lags (beyond mechanical inertia) in the system since the actual opening and closing of the spool valve flow ports lags the control actuator input. Unless the control actuator profile is adjusted to account for the lag, the effect on the valve event is to shorten its duration, as there is no lag between the return motion of the control actuator and the opening of the return port. Spool underlap leads to difficulty in accurately positioning the engine valve at the appropriate lift height, as well as making soft-landing control of the valve more difficult.

A rough estimate for the total power consumption of a valvetrain using these actuators was obtained by estimating the power consumed by a pair of actuators from the simulation data. This was then extrapolated to take into account the valve timing for a production 4-cylinder 16-valve engine. At an idle speed of 556 RPM, EHVA total system power consumption would be approximately 90W compared to 160W for a cam-driven system. At a high load, high speed condition (5560 RPM), this EHVA power consumption would increase to 6.6kW, compared to 2.2kW in a cam-driven system. These values do not take supply pump efficiency into account. A low efficiency pump (60%, say) would worsen the above consumption figures considerably. This highlights the need to optimize system operating pressures to the greatest extent possible.

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