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Modeling and Control Design of a Camless Valve Actuation System

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Abstract— **This paper presents the modeling and control design of a new fully flexible engine valve actuation system which is an enabler for camless engines. Unlike existing electro-mechanical or servo actuated electro-hydraulic valve actuation systems, precise valve motion control is achieved with a hydro-mechanical internal feedback mechanism. This feedback mechanism can be turned on or off in real-time using simple two state valves which helps reduce the system cost and enables mass production. Since the external control only activates or deactivates the internal feedback mechanism, the trajectory of the entire closed-loop system is purely dependent on the design parameters of the internal feedback system. A mathematical model of the system is developed to evaluate the effect of each of the design parameters. The "Area-schedule" is identified as the key design feature which affects the trajectory of the closed-loop system. It needs to be designed systematically to optimize the performance of the system as well as improve its robustness. By treating this feature as the feedback control variable, the design problem is transformed into a nonlinear optimal control problem which is later solved using the numerical dynamic programming method. The effectiveness of the designed area-schedules is verified with simulations.**

I. INTRODUCTION

The internal combustion engine has been continuously refined during the last century. However, until recently there was very little change in the fundamental subsystems such as those used for fuel delivery and air handling. The use of fuel injectors in place of the carburetor enabled better control of fuel delivery to the engine which helped optimize the engine operation for different load and speed conditions and led to significant improvements in the efficiency and reduction in the emissions of the engine. Similarly, the traditional camshaft based air handling system offers no flexibility. Recent issues like depleting oil reserves, increasing fuel prices, stricter emission standards and the increased use of alternative fuels have all motivated the requirement of a better air management system.

It has been shown that a flexible air handling system with the capability of varying the valve lift, timing, duration or a combination of these parameters can offer significant improvements in the performance and efficiency over a wide range of operating conditions [1]. At present, camshaft based variable valve actuation systems are offered by a number of automobile manufacturers [2]-[4]. Most of these systems have limited flexibility or become mechanically complicated and expensive with the increase in flexibilities.

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One possibility of increasing the flexibility of valve motion control without increasing the associated mechanical complexity is to eliminate the camshaft and use electronically controlled systems for actuating the engine valves. These systems are referred to as "Camless Valve-trains" and have the capability of integrating all the flexibilities into one system. These systems are capable of controlling each valve individually on a cycle to cycle basis which can enable advanced combustion concepts such as Internal exhaust gas recirculation and homogenous charge compression ignition. They can also enable throttle-less load control and hence reduce the pumping losses in the engine.

A significant amount of research has been carried out in this area in the development of both laboratory systems [6]-[9] and production oriented systems [10]-[12]. These systems can be classified into the following broad categories based on the energy source used for actuation.

Electro-mechanical systems use the electromagnetic forces generated between sets of armatures and coils for moving and positioning the engine valve. The nonlinearity of the electromechanical force and the time constant due to the inductance in the coils make it very difficult to control the engine valve at the end of the trajectory. To achieve seating velocity control and lift control, these systems require complicated real-time control strategies [6]-[7] which are difficult to implement in mass produced systems.

Electro-hydraulic and Electro-pneumatic systems use pressurized hydraulic or pneumatic fluid controlled by valves to provide the force required for actuation. These systems depend on precise control of the fluid flow to ensure accurate positioning of the engine valve [8]. Hence they require complicated and expensive proportional valves which increases the cost and hence are not viable for mass production. These systems throttle the hydraulic fluid during a major portion of the operation cycle to control the flow to the engine valve actuator. The large pressure drop across the proportional valves leads to the requirement of a higher supply pressure and thus increases the power consumption. The pneumatic systems [9] are affected by fluid compressibility which leads to difficulties in achieving precise valve motion and seating velocity control.

Hence, for a system to be mass-produced, it is required to have,

- Flexibility in lift, timing and duration
- Low valve seating velocities
- Low power consumption

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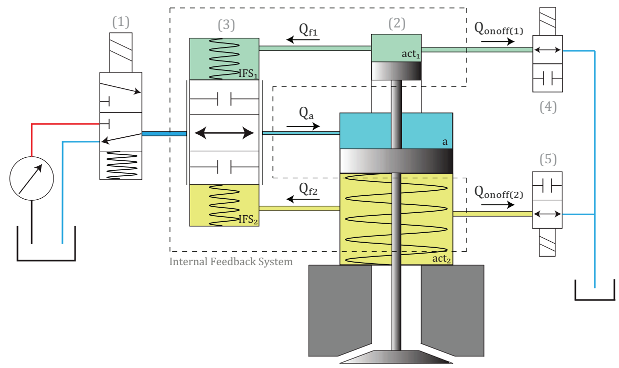

Fig. 1. Schematic of the Variable valve actuation system with internal feedback

- A feedback system with simple and inexpensive components that requires minimum calibration while being capable of precise valve motion control.
- Subsystems which can be packaged compactly and efficiently.

This paper presents a new valve actuation system [13] that can address all the issues mentioned earlier. The first section explains the construction and working of the proposed mechanism. It is followed by the development of a mathematical model which can be used for control design purposes. A systematic method for designing the ‘area-schedule’, which is the key control parameter is presented and later validated using simulations.

II. System Design

The block diagram of the new valve actuation system is shown in Fig. 1.

A. Construction

- Component (1) is a solenoid actuated 2-position valve, which connects the entire system to the high pressure pump or the tank.
- Component (2) is the actuator which is connected to or in contact with the engine valve’s stem. It consists of a piston with a spring on one side and the actuation chamber with the hydraulic fluid on the other side.
- Component (3) is a spool valve which acts as the feedback regulator. The spool’s position depends on the difference in pressure between the top and bottom chambers IFS_1 *and* IFS_2. These pressures are in turn dependent on the pressure of the top and bottom feedback chambers of the actuator act_1 *and* act_2. The spool valve’s orifice area is designed such that it is maximum for the spool’s un-deflected position and decreases in both directions of movement of the spool.
  The chambers act_1, act_2, IFS_1 *and* IFS_2 along with the spool valve form the internal feedback system *(IFS)* as highlighted in the schematic.
- Components (4) & (5) are on-off valves, which connect the two feedback chambers of the actuator to the tank. When both the on-off valves are open, there is a free flow of fluid between the actuator’s feedback chambers and the tank. Closing either of the on-off valves restricts the flow out of the corresponding feedback chamber and hence couples the dynamics of the IFS with the motion of the actuator.

B. Working principle

At the beginning of a typical valve operation cycle, the 2-position solenoid valve is in the de-energized state. The only force acting on the actuator is due to the spring which holds the engine valve in the closed position. Both on-off valves are open and hence all the feedback chambers are at the tank pressure and the spool of the feedback regulator is balanced in the center position due to the force exerted by its springs. To open the engine valve, the system is connected to the high pressure pump by energizing the solenoid. When the pressure in the actuation chamber overcomes the spring force, it accelerates the actuator downwards. The bottom on-off valve is shut off when the engine valve reaches a desired lift. This causes pressure to build up in the actuator’s bottom feedback chamber, which in turn increases the pressure in the feedback regulator’s bottom chamber and accelerates the spool upward. As the spool moves upward, the orifice area of the spool valve decreases and thus restricts the flow through it. This reduces the pressure in the actuation chamber and causes the actuator to decelerate. The synchronized motion between the actuator and the spool valve will continue until flow to the actuator is completely shut off by the spool and the actuator comes to a stop smoothly. Hence by controlling the timing of the on-off valve, we can control the maximum lift of the engine valve.

To close the engine valve, the 2 position solenoid valve is de-energized to connect the entire system to the tank, which reduces the pressure in the actuation chamber and hence accelerates the actuator upwards. The top on-off valve is closed when the engine valve is near the seat. It would deflect the spool in the downward direction and hence gradually decrease the orifice area which restricts the flow out of the actuation chamber. This causes an increase in the actuation chamber pressure which gradually decelerates the engine valve and ensures a desired seating velocity.

In addition to the highly effective lift and seating velocity control, this system has another advantage. The internal feedback system is controlled by hydraulic pressure which makes the entire system inherently stiff. When compared to electro-mechanically actuated proportional valves, this system has a relatively small time constant. This allows the control system to be activated at the last possible moment which decelerates the engine valve very close to the end of the trajectory. The system thus operates without throttling during a large portion of its operating cycle and hence the pressure drop across the feedback regulator is minimal. Hence a relatively lower supply pressure is capable of providing the required actuation effort which leads to a decreased power consumption.

A prototype experimental setup has been developed and has demonstrated the capabilities of the proposed concept [5].

III. Development and Validation of the System Model

A. Mathematical model of the system

The dynamics of the entire feedback system depend on a number of parameters like the physical dimension of each of the components, the stiffness of the various springs and the timing of the on-off valves etc. It is therefore necessary that we have an accurate mathematical model of the system, which will help to evaluate the effect of various parameters and verify any new design ideas that are proposed.

To capture all the dynamics of the high-speed operation of the system, the equations of motion for the actuator and the spool and for the fluctuation of pressures in various chambers need to be determined. The valve operates at 50Hz when the engine is running at 6000 RPM. For modeling the pressure variations, the effect of fluid compressibility needs to be considered since it becomes prominent at such high frequencies. The dynamics of the actuator, spool and pressures in the various chambers are described as follows.

$$\dot{X}_{act} = V_{act} \tag{1}$$

$$\dot{V}_{act} = \frac{1}{M_{act}}[P_a \cdot A_a + P_{act_1} \cdot A_{act_1} - P_{act_2} \cdot A_{act_2} - K_{act} \cdot X_{act} - b_{act} \cdot \dot{X}_{act} - F_{preload}] \tag{2}$$

where, X_{act} and V_{act} are the position and velocity of the actuator, M_{act} is the moving mass of the actuator and the engine valve assembly, P_a is the pressure in the actuation chamber, A_a is the area of the actuator's piston, P_{act_1}, P_{act_2} are the pressures in the actuator's top and bottom feedback chambers, A_{act_1} and A_{act_2} are the areas of the actuator's top and bottom feedback chambers, K_{act} is the stiffness of the actuator spring, $F_{preload}$ is the spring preload and b_{act} is the damping coefficient of the actuator.

$$\dot{X}_{spool} = V_{spool} \tag{3}$$

$$\dot{V}_{spool} = \frac{1}{M_{spool}}[P_{IFS_1} \cdot A_{IFS_1} - P_{IFS_2} \cdot A_{IFS_2} - (K_{IFS_1} + K_{IFS_2}) \cdot X_{spool}] \tag{4}$$

where, X_{spool} and V_{spool} are the position and velocity of the spool, M_{spool} is the mass of the spool, P_{IFS_1} and P_{IFS_2} are the pressures in the spool valve's top and bottom feedback chambers, A_{IFS_1} and A_{IFS_1} are the areas of the feedback regulator's top and bottom feedback chambers, K_{IFS_1} and K_{IFS_2} are the stiffnesses of the IFS springs and b_{IFS} is the damping coefficient of the IFS.

$$\dot{P}_a = \frac{\beta\,(Q_a - V_{act} \cdot A_a)}{(x^*_{act} + X_{act}) \cdot A_a} \tag{5}$$

$$\dot{P}_{act_1} = \frac{\beta\left(-Q_{f1} - Q_{onoff_1} - V_{act} \cdot A_{act_1}\right)}{\left(x^*_{act_1} + X_{act}\right) \cdot A_{act_1}} \tag{6}$$

$$\dot{P}_{act_2} = \frac{\beta\left(-Q_{f2} - Q_{onoff_2} + V_{act} \cdot A_{act_2}\right)}{\left(x^*_{act_2} - X_{act}\right) \cdot A_{act_2}} \tag{7}$$

$$\dot{P}_{IFS_1} = \frac{\beta\left(Q_{f1} - V_{spool} \cdot A_{IFS_1}\right)}{\left(x^*_{spool_1} + X_{spool}\right) \cdot A_{IFS_1}} \tag{8}$$

$$\dot{P}_{IFS_2} = \frac{\beta\left(Q_{f2} + V_{spool} \cdot A_{IFS_2}\right)}{\left(x^*_{spool_2} - X_{spool}\right)} \tag{9}$$

where, β is the bulk modulus of the hydraulic fluid and x^*_{act}, $x^*_{act_1}$, $x^*_{act_2}$, $x^*_{spool_1}$ and $x^*_{spool_2}$ are the clearances in each of the corresponding chambers when the engine valve is in the closed position and the spool in the center position. The flow rates between the chambers and the sign convention adopted is shown in Fig. 1. The flow rates are calculated using the orifice equation.

$$Q = A \cdot C_d \cdot \sqrt{\frac{2 \cdot |P_1 - P_2|}{\rho}} \cdot sign(P_1 - P_2) \tag{10}$$

where, A is the associated orifice area between the chambers, C_d is the discharge coefficient, ρ is the density of the fluid, P_1 and P_2 are the upstream and downstream pressures. In the case of Q_a, the orifice area A_{spool} is a function of the displacement of the spool X_{spool}. This relation $A_{spool} = f(X_{spool})$ is called the area-schedule. In the case of Q_{onoff_1} and Q_{onoff_2}, the corresponding orifice areas A_{onoff_1} and A_{onoff_2} are set to either 0 or maximum depending on the state of the particular on-off valve.

The area used for calculating the flow rates between the feedback chambers (Q_{f1} and Q_{f2}), is the area of the orifice in the channel between the chambers (A_{f1} and A_{f2}).

The dimensions of the feedback chambers are designed such that, after the corresponding on-off valve is closed, the actuator can travel a maximum of $2mm$ while decelerating steadily. Therefore, by the appropriate timing of the on-off valves, the maximum lift and the seating velocity of the engine valve can be controlled precisely.

B. Model simulation and validation

To simulate the system model, it is first discretized as follows,

$$\chi(k+1) = \chi(k) + \dot{\chi}(k)\Delta T \tag{11}$$

The derivatives $\dot{\chi}(k)$ for each of the states are calculated using the equations derived in the previous section. The system design parameters required for the simulation are chosen based on the experimental setup discussed in [5]. The initial values for the position and velocity of the actuator and spool are set to $0mm$ and $0mm/s$ respectively. The initial values for the pressure in all the chambers are set to 1×10^5 Pa. The discrete model is simulated using Matlab.

The lift of the engine valve is controlled by varying the timing of the bottom on-off valve. Simulations are performed by activating the IFS when the engine valve reaches 6 mm, 7 mm and 8 mm which controls the lift of the engine valve to 8 mm, 9 mm and 10 mm accordingly. The results of these simulations are shown in Fig. 2(a). For valve closing, activating the top on-off valve when the engine valve reaches 2 mm, decelerates the engine valve and ensures that the valve

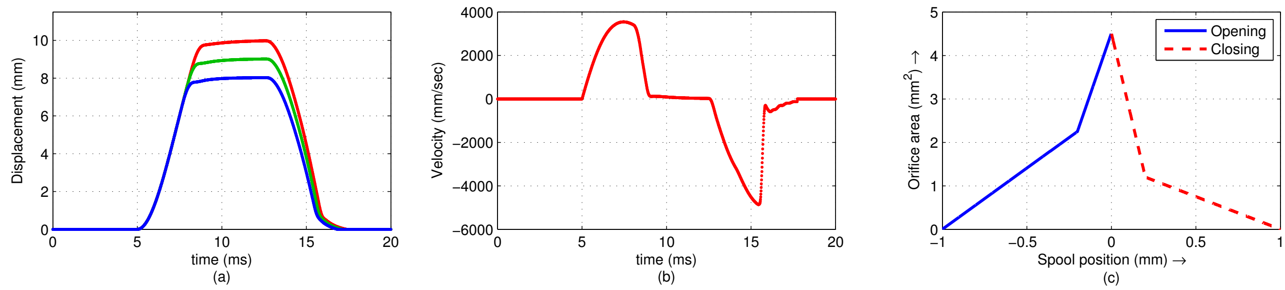

Fig. 2. (a) Lift and seating velocity control using the IFS; (b) Actuator velocity corresponding to 10mm lift; (c) Area-schedule used for simulations

seating velocity is minimized. Fig. 2(b) shows the velocity of the valve corresponding to the 10mm lift case. The position and velocity traces agree closely with the results obtained from AMEsim *(a physics based multi-disciplinary modeling & simulation package)* model simulation in [5].

IV. Control Design

A. Area-schedule as the nonlinear feedback control

The internal feedback mechanism needs to ensure that, during valve opening, the actuation chamber pressure is reduced to decelerate and stop the valve precisely at the required lift. However, it is also required that $P_a > 0$ at all times to prevent cavitation which can damage the system. For valve closing, the actuation chamber pressure should be such that, it can ensure low seating velocities. The actuation chamber pressure thus needs to be precisely controlled during the operation of the IFS. The only way by which the pressure in the actuation chamber can be varied is by controlling the fluid flow to and from the chamber which is in turn dependent on the orifice area of the spool. Since the spool is not directly controlled by external controls, to vary the orifice area as required, the relationship between the spool displacement and the corresponding orifice area i.e., the *Area-schedule* needs to be designed appropriately.

The linear schedule used in proportional valves i.e., ($A_{spool} = k \cdot [1 - |X_{spool}|]$) causes cavitation inside the actuation chamber and fails to slow down the engine valve from speeds greater than $2.5m/sec$. For the simulations shown previously, an area schedule with two slopes as shown in Fig. 2(c) was used. The point where the slope changes is a parameter that is tuned by trial and error to ensure satisfactory performance. It will be referred to as the "Hand-Tuned (HT)" area-schedule.

For valve opening, the best possible hand-tuned area-schedule causes the pressure to drop to $3 \times 10^5 Pa$ for velocities near $3.5m/sec$. Although this is satisfactory when all parameters are exactly as modeled, in practice there will be variations due to dynamic operating conditions, manufacturing defects and wear. Simulations with perturbed parameters using the hand-tuned area-schedule indicated the occurrence of cavitation during valve opening. Hence, if an area-schedule is designed such that the minimum pressure in the actuation chamber is much higher than $0 \times 10^5 Pa$, the system would then become insensitive to parameter perturbations within a certain tolerance limit.

For valve closing, the hand-tuned area schedule is a very conservative design because, it decelerates the engine valve very early and lets it travel with a very low velocity till it lands on the seat. Though this can ensure a low seating velocity, it keeps the valve near the seat for a long time before closing which leads to pumping losses in the engine due to the heavy throttling of the gases to and from the cylinder. An ideal trajectory for closing would be one in which the valve decelerates at the last moment possible as it approaches the valve seat, which would avoid pumping losses in the cylinder and also minimize the closing time.

B. Optimization of the Area-schedule

If the area-schedules are designed considering all the dynamics of the system, the pressure profile can be optimized to satisfy the requirements for both the valve opening and the closing case. The required area-schedules would then be more complicated than the 2-slope design and hence, it can no longer be designed by trial and error due to the increased degrees of freedom. A systematic design procedure which can account for all the dynamics of the system is thus required.

To optimize the area-schedule, assuming that A_{spool} is controllable independently, an $A_{spool}(t)$ that can optimize the pressure profile in the actuation chamber is determined. The corresponding spool displacement $X_{spool}(t)$ is found using simulations and by matching it with the $A_{spool}(t)$, the required $A_{spool} = f(X_{spool})$ can then be determined. However, since the model we have is non-linear, it is difficult to obtain an analytical solution to the optimization problem. The problem is thus solved numerically using dynamic programming [14].

Due to the extremely stiff nature of the feedback chamber pressure dynamics, the system needs to be discretized with a relatively small sampling time which results in a large number of stages. To capture all the dynamics during the operation of the IFS, the approximate number of stages required is 15000 for valve opening and 9000 for valve closing. The accuracy of the results of dynamic programming greatly depends on the number of candidates chosen for each of the 9 states at each stage.

However, even a choice of just 10 candidates for each of the states and 15 candidates for the control input indicates that the number of grid points required at each stage is 10^9 which translates into a very large memory requirement. For each of the grid points at each stage, 15 calculations

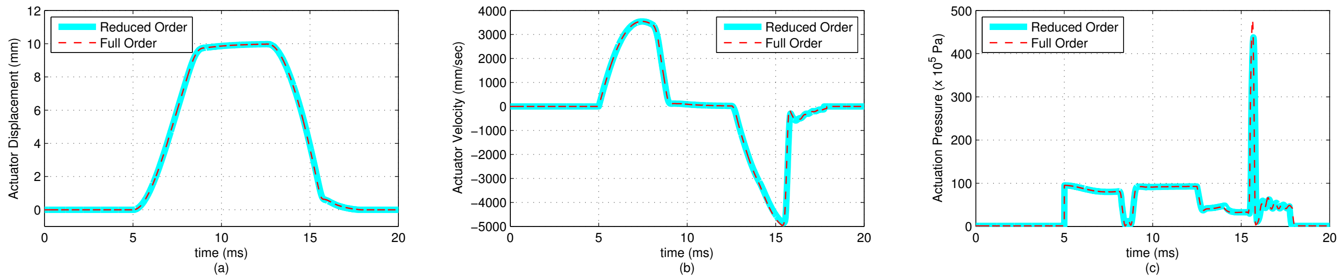

Fig. 3. Comparison between reduced order model and full order model (a) Actuator position; (b) Actuator velocity; (c) Actuation chamber pressure

are required to evaluate the cost function and 15 additional calculations maybe required to perform any interpolations as described in [15], which leads to very large computation times. The high dimensionality of the problem and the stiff nature of the dynamics make the application of the conventional dynamic programming very difficult.

C. Dynamic programming using a reduced order model

A reduced order model for control design purposes is obtained by neglecting the dynamics of the pressure in the feedback chambers. The differential equations describing the pressure in the feedback chambers and the motion of the spool reduce to algebraic equations, which results in a 3^{rd} order model of the system. Simulation results corresponding to the full order model and the reduced order model shown in Fig. 3(a),(b) and (c) indicate a good agreement between both the models. The dynamics of the 3 parameters in the reduced order model are relatively slower and thus lead to a reduction in the number of time stages required.

The use of the conventional DP for the reduced order model also turned out to be challenging due to computational issues. The improved dynamic programming method presented in [15] is thus used for the solution of the reduced order optimization problem. The parameter $A_{spool}(t)$ is the control input which needs to be optimized to control the trajectory of X_{act}, V_{act} and P_a. The feedback regulator design is such that, the area-schedule for the opening and closing of the valve are independent from each other and hence can be designed individually.

For the valve opening case, the range of values for each of the parameters are, X_{act}=$[8,10mm]$, V_{act}=$[0,3500mm/sec]$, P_a=$[0\times10^5, 95\times10^5 Pa]$ and A_{spool}=$[0,4.5mm^2]$.

Let $\chi = [X_{act}, V_{act}, P_a]$. The performance index for the valve opening case is given by,

$$g = \lambda_1\left(\chi(N)-\chi^*_{final}\right)^2 + \lambda_2\left(\chi(0)-\chi^*_{initial}\right)^2 + \sum_{k=1}^{N-1}\Big[\lambda_3^*\left(A_{spool}(k)-A_{spool}(k+1)\right)^2 + \lambda_4^*\left(P_a(k)-10\times10^5\right)^2\Big] \tag{12}$$

$$\lambda_3^* = \begin{cases} 1000\cdot\lambda_3 & \text{if } A_{spool}(k+1) > A_{spool}(k); \\ \lambda_3 & \text{if } A_{spool}(k+1) \le A_{spool}(k). \end{cases}$$

$$\lambda_4^* = \begin{cases} \lambda_4 & \text{if } P_a(k) < 10\times10^5; \\ 0 & otherwise. \end{cases}$$

The first two terms are designed to enforce the required initial and final conditions of the trajectory for all the states. The third term ensures that the $A_{spool}(t)$ decreases smoothly and monotonically. The fourth term is designed to ensure that the minimum P_a value is above a desired value which is chosen to be significantly larger than $0\times10^5 Pa$. Fluid separation can thus be avoided in the presence of parameter variations due to the margin of safety built into the system.

For the valve closing case, the range of values for each of the parameters are, X_{act}=$[0,2mm]$, V_{act}=$[-5500,0mm/sec]$, P_a=$[1\times10^5, 300\times10^5 Pa]$ and A_{spool}=$[0,4.5mm^2]$. The performance index to be minimized is given by,

$$g = \lambda_1\left(\chi(N)-\chi^*_{final}\right)^2 + \lambda_2\left(\chi(0)-\chi^*_{initial}\right)^2 + \sum_{k=1}^{N-1}\Big[\lambda_3^*\left(A_{spool}(k)-A_{spool}(k+1)\right)^2 + \lambda_4^*\left(V_{act}(k)-(-500)\right)^2 + \lambda_5^*\left(V_{act}(k)-V_{act}(k+1)\right)^2\Big] \tag{13}$$

$$\lambda_3^* = \begin{cases} 1000\cdot\lambda_3 & \text{if } A_{spool}(k+1) > A_{spool}(k); \\ \lambda_3 & \text{if } A_{spool}(k+1) \le A_{spool}(k). \end{cases}$$

$$\lambda_4^* = \begin{cases} \lambda_4 & \text{if } V_{act}(k) > -500; \\ 0 & otherwise. \end{cases}$$

$$\lambda_5^* = \begin{cases} \lambda_5 & \text{if } V_{act}(k+1)-V_{act}(k) > V^*_{seating}; \\ 0 & otherwise. \end{cases}$$

The fourth term in the valve closing case is designed to penalize slow actuator velocities. This will ensure that the actuator decelerates at the last possible instant and thus has a minimized closing time. The fifth term is designed to minimize the seating velocity.

The parameters λ_1-λ_5 are the weighting factors which need to be tuned carefully to achieve a balance between all the terms in the cost function.

The dynamic programming algorithm gives an optimized trajectory for all the 3 states and the associated control $A_{spool}(t)$. The corresponding trajectory of the spool $X_{spool}(t)$, is evaluated by simulating the 9^{th} order model with the $A_{spool}(t)$ forced to the values obtained from the dynamic programming. By matching them, the required area-schedule $A_{spool} = f(X_{spool})$ is computed.

D. Validation of the designed Area-schedule

The area-schedules obtained from dynamic programming for the valve opening and closing cases are shown in Figs.

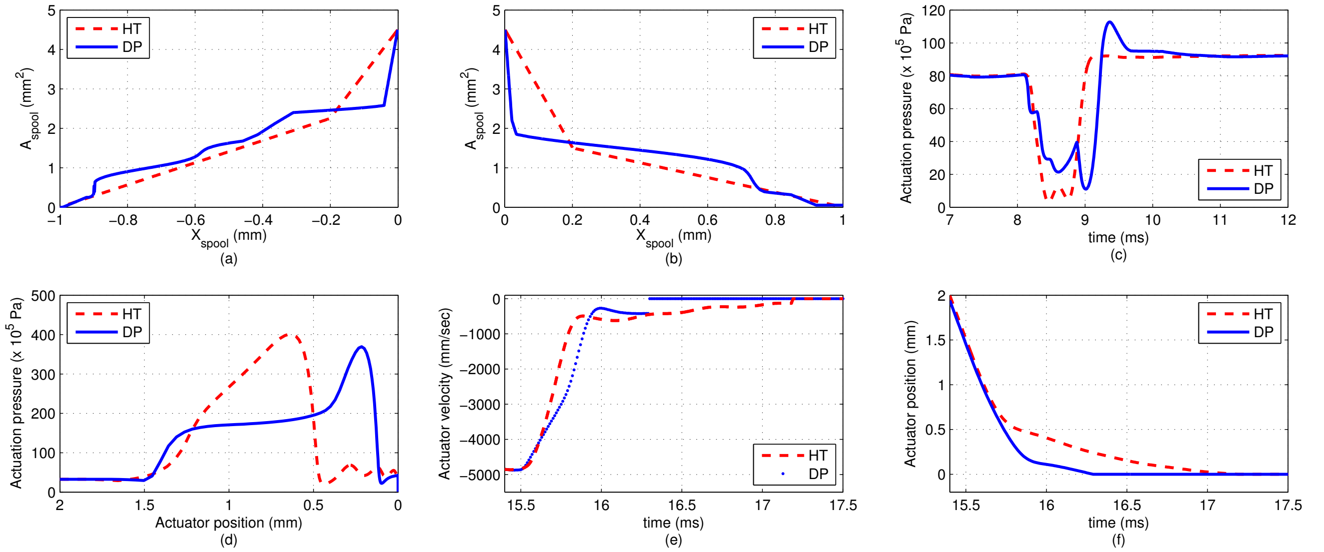

Fig. 4. Comparison of (a) Area-schedule for valve opening; (b) Area-schedule for valve closing; (c) Actuation chamber pressure during closing; (d) Actuation chamber pressure during opening; (e) Actuator velocity during closing; (f) Actuator displacement during closing.

4(a) and (b). The system is simulated using these area-schedules and the results are compared to those obtained using the hand-tuned schedule. The actuation chamber pressure during opening shown in Fig. 4(c) indicates that the minimum value is close to $10 \times 10^5 Pa$. The system should thus be able to operate without cavitation for a reasonable amount of parameter variations. For valve closing, Fig. 4(d) shows that the actuation chamber pressure is raised only very close to the valve seat. This causes the engine valve to decelerate at the last possible instant and thus has a minimized closing time as shown in Fig. 4(e) and (f).

V. Future Work

The main drawback of this procedure is that, it does not account for the dynamics of the IFS. The obtained $A_{spool}(t)$ causes the IFS spool to oscillate which manifests itself as a wave like pattern in the designed Area-schedules. It would be difficult to realize such a design due to the difficulty associated with machining a complex geometry within a small space. If the optimization of $A_{spool}(t)$ is carried out by considering the dynamics of the IFS also, it is possible to add an additional penalty on the spool oscillation. This would result in a solution which minimizes the spool oscillations and thus produces an area-schedule which can be manufactured easily. The entire trajectory of the system can also be optimized to ensure robust performance in the presence of model uncertainty and parameter variations.

VI. Conclusions

This paper presents a model based control design of a new engine valve actuation system. The nonlinear feedback control is built into the design of the system which greatly simplifies the external control and the calibration effort required. To improve the performance and robustness of the system, a critical control parameter is identified and its design is formulated as an optimization problem. Due to the difficulties associated with the high dimensionality of the problem, a numerical solution based on a reduced order model is calculated. The effectiveness of the obtained designs is verified using simulations. The potential issues associated with the design based on the reduced order model motivate the development of a computationally feasible method for solving the higher order optimization problem.

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