MODELING AND CONTROL OF A HIGH PRESSURE COMBINED AIR/FUEL INJECTION SYSTEM

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ABSTRACT
A high pressure combined air-fuel injection system is designed and tested for an experimental free liquid-piston engine compressor. The application discussed utilizes available high pressure air from the compressor’s reservoir, and high pressure fuel to mix and then inject into a combustion chamber. This paper addresses the modeling, design and control for this particular high-pressure air-fuel injection system, which features an electronically controlled air/fuel ratio control scheme. This system consists of a fuel line and an air line, whose mass flow rates are restricted by metering valves. These two lines are connected to a common downstream tube where air and fuel are mixed. By controlling the upstream pressures and the orifice areas of the metering valves, desired A/F ratios can be achieved. The effectiveness of the proposed system is demonstrated by a lumped-parameter model in simulation and validated by experiments.

INTRODUCTION
In the conventional spark-ignition (SI) engine research community, high-pressure direct injection has been increasingly attracting researchers’ attention for its more complete combustion, high power output and low emissions of substances that affect the environment [3]. Under a high pressure, direct injection systems are able to supply finely atomized fuel to the combustion chamber. Conventional direct injection concerns only the fuel. In the application studied in this paper, a free-piston engine compressor, high pressure air is also always on hand. Such a system presents the opportunity of combining the high pressure injection of fuel with high pressure air. Such a scheme allows an engine to bypass the conventional intake and compression strokes; the simultaneous introduction of high pressure air and fuel presents a high pressure mixture directly before combustion that is equivalent to the end of a conventional compression stroke. Utilizing high pressure for both air and fuel “combined injection” is proposed in this paper for small scale free-piston engines [1, 2] such as that shown in Fig. 1. Although the proposed approach is developed for this particular configuration, it can be applied to any engine application where both high pressure air and high pressure fuel are metered into the combustion chamber.

FIGURE 1. FREE-PISTON ENGINE COMPRESSOR

The proposed injection system has two separate air and fuel lines that can be metered and then combined to supply a controlled air/fuel mixture. If the driving pressure of the air and fuel is considered as arbitrary, and the downstream pressure of the mixture is also considered arbitrary, the proposed combined injection system shares many similarities to the air/fuel injection scheme used in conventional SI engines. However, the proposed combined air/fuel injection ratio control strategy is functional different.

Generally speaking, conventional air/fuel ratio control regulates the fuel flow as a certain ratio of the air flow where the air flow is typically directly commanded by the driver in automobile engines. An illustration of the typical air/fuel
injection system is shown in Fig. 2. The air flow is regulated by throttling and fuel flow is then appropriately controlled. For the specialized application considered here, the system is still applicable but upstream and downstream pressures are somewhat different. A conventional IC engine with fuel injection inducts atmospheric pressure air $P_a$ and injects high pressure fuel $P_f$ into a downstream pressure $P_d$. For the proposed application here, the air and fuel pressures will always be significantly higher than the downstream pressure. Hence, this particular configuration requires a different control strategy.

Conventional air/fuel ratio control for internal combustion engines commonly includes an inner-loop controller based on the deviation of the estimated three-way catalyst stored oxygen state [4][5]. This scheme must account for the delay between the controlled input and the sensor reading. The proposed system directly injects a high pressure air and fuel mixture to the combustion chamber through separately metered air and fuel lines, assisted by high upstream driving pressures. The control strategy will be to regulate the fuel pressure to match that of the air pressure. A local feedback control scheme based on pressure is used before the mixture enters the intake manifold. This pressure based control approach eliminates the delay problem introduced in conventional IC engines that measure exhaust products.

FIGURE 2. GENERALIZED AIR/FUEL RATIO CONTROL

The proposed combined injection system is designed for an experimental free liquid-piston engine air compressor, as shown in Fig. 1, which, by virtue of being an air compressor, has a high pressure air reservoir available for high pressure air injection. The “high pressure” mentioned here is between 300kPa and 700kPa. The pressure based control scheme will control the fuel pressure to match that of the air pressure. Once the two driving pressures are equal, air and fuel will flow and mix simultaneously into a common downstream pressure. The ratio of the two mass flows will be determined not by each flow’s duration, as is the case in conventional engines, but by the ratio of the flow restrictions of the two lines subject to the same pressure drop. The specifics of this strategy will be discussed in detail in the remainder of the paper. Therefore, pressure is used for feedback control in this approach. As a matter of fact, pressure has been used by engineers to control various applications in internal combustion engines [6][7], made possible by pressure sensors that can provide fast, accurate, and convenient real-time pressure for examining the state of the system.

This paper presents modeling, design and control of the combined air/fuel injection system for our free liquid-piston engine compressor. Since this injection system is designed to be simple and compact, the control law is accordingly designed to be simple so that it can be easily transferred to other small scale SI engine applications. The design of this system will be first introduced, followed by a lumped-parameter dynamic model. Simulation results will be shown and compared with experimental data.

PRESSURE-BASED MASS FLOW CONTROL

High-Pressure Combined Air/Fuel Injection System

The conventional approach to the control of fuel flow uses a high speed on/off injection valve and PWM control. The proposed approach here will control the fuel flow relative to air flow by controlling the upstream pressure to a fixed but known flow restriction. Therefore, the configuration in Fig. 2 can be extended to a more specific configuration shown in Fig. 3. The proposed system primarily consists of two supply lines: the air line and the fuel line, respectively. The air used for mixture injection comes from an external air reservoir. On the other line, the fuel source is a 0.5kg bottle of propane, which at room temperature has a vapor pressure of about 1 MPa (154 psia). The mass flow rates of air and fuel, through the air line and the fuel line respectively, are controlled by the metering valves to achieve the stoichiometric ratio, based on the proposed pressure-based mass flow control method. Air and fuel are mixed in the downstream tube before entering the combustion chamber. Note that there is a fuel buffer cylinder between the fuel valve and the metering valve in Fig. 3. Its function will be discussed in the control section.

FIGURE 3. SCHEMATIC OF THE HIGH PRESSURE COMBINED AIR/FUEL INJECTION SYSTEM

The mass flow rates through all valves or any flow restricted areas depend on the upstream and the downstream
pressures. The following equations give the mass rate $\dot{m}$ under subsonic and sonic conditions [8]:

$$\dot{m} = a \psi(P_a, P_f)$$

$$= \begin{cases} 
  aC_1 \frac{P_a}{T_u} & \text{if } \frac{P_a}{P_f} \leq P_{cr} \\
  aC_2 \frac{P_f}{P_u} \left( \frac{P_d}{P_u} \right)^{\gamma_u/\gamma_u-1} & \text{if } \frac{P_a}{P_f} > P_{cr}
\end{cases}$$

(1)

where $C_a$ is a nondimensional discharge coefficient of the valve, $a$ is the area of the valve orifice, $P_a$ and $P_d$ are the upstream and downstream pressures, $T_u$ is the upstream temperature, $\gamma_u$ is the ratio of specific heats of the upstream substance, and $C_1$, $C_2$ and $P_{cr}$ are substance-specific constants given by

$$C_1 = \sqrt{\frac{\gamma_u}{R_u}} \left( \frac{2}{\gamma_u + 1} \right)^{\gamma_u/\gamma_u-1}$$

(2)

$$C_2 = \sqrt{\frac{2\gamma_u}{R_u}} \left( \frac{2}{\gamma_u + 1} \right)^{\gamma_u/\gamma_u-1}$$

(3)

$$P_{cr} = \left( \frac{2}{\gamma_u + 1} \right)^{\gamma_u/\gamma_u-1}$$

(4)

where $R_u$ is the gas constant of the upstream substance. The specific heat ratio of the air is $\gamma_a = 1.4$. Given the orifice areas of the air and fuel metering valves ($a_a$ and $a_f$), the standard model for the isentropic restriction air flow $\dot{m}_a$ and fuel flow $\dot{m}_f$ are given as,

$$\dot{m}_a = a_a \psi_a(P_a, P_d)$$

(5)

$$\dot{m}_f = a_f \psi_f(P_f, P_d)$$

(6)

where $P_a$ is the air upstream pressure, $P_f$ is the fuel upstream pressure and $P_d$ is the common downstream pressure, as shown in Fig. 3. Air and fuel are mixed in the intake manifold, a common downstream volume, with the A/F ratio given by,

$$\lambda_{af} = \frac{\dot{m}_a}{\dot{m}_f} = \frac{a_a \psi_a(P_a, P_d)}{a_f \psi_f(P_f, P_d)}$$

(7)

In this way, the A/F ratio can be controlled by upstream pressures and the orifice areas of air and fuel metering valves.

**Air/Fuel Ratio Control**

The key idea behind the control scheme is to set a proper ratio of the orifice areas of the air and fuel metering valves so that when $P_a = P_f$, Equation (7) results in the desired air/fuel ratio. As seen in Eq.(7), there are four variables that together determine the air/fuel ratio: $a_a$, $a_f$, $\psi_a$ and $\psi_f$. Areas $a_a$ and $a_f$ are the orifice areas of the air and fuel metering valves, which can be manually set to a certain ratio. The area normalized mass flow rate functions $\psi_a$ and $\psi_f$ are determined by the upstream pressures, the downstream pressure and $\gamma_a$, the ratio of specific heats of the upstream substances, as shown in Eq. (1). In the configuration described in the last section, if the upstream pressures for air and fuel lines are the same with $P_a = P_f = P_{cr}$, then the ratio of the two nonlinear functions $\psi_a$ and $\psi_f$ only depends on $\gamma_a$. The fuel used in our application is propane with $\gamma_{prop} = 1.136$, and $\gamma_{air} = 1.4$.

As an example, if we assume the upstream pressures for air and fuel both are $P_u = 372.3$ kPa (54 psia), the ratio of $\psi_a/\psi_f$ for a range of intake manifold pressures between 240 kPa and 372 kPa (where $P_f/P_u > P_{cr}$) is shown in Fig. 4.

Although the $\psi_a/\psi_f$ ratio is not constant while the downstream pressure is changing, the variation is relatively small if the metering valves are large enough to prevent a large pressure drop in the downstream pressure relative to the upstream pressure during injection. As a result, $\lambda_{af}$ is mainly determined by the ratio of the orifice areas of air and fuel metering valves when the upstream pressures for both lines are same. A constant mean value (1.64 in this case) for $C_{af} = \psi_a/\psi_f$, called gamma A/F ratio in Fig. 3, can be chosen for a particular upstream pressure so that the ratio of $a_a$ and $a_f$ can be correspondingly selected to give a desired A/F ratio $\lambda_{af}$ as,

$$\lambda_{af} = \frac{\dot{m}_a}{\dot{m}_f} = \frac{a_a \psi_a(P_a, P_d)}{a_f \psi_f(P_f, P_d)} \approx 1.64 \frac{a_a}{a_f}$$

(8)

**FIGURE 4. THE RATIO OF $\psi_a/\psi_f$ WITH EQUAL UPSTREAM AND DOWNSTREAM Pressures.**

The pressure variation in the downstream CV is determined by the mass flow rates in and out of the CV. In order to keep $C_{af}$ as close to a constant as possible in Eq. (8),
it is desired to keep the downstream pressure from varying too much. In other words, the mixture mass \( m_d \) in the downstream CV should be kept close to a constant, which is calculated as,

\[
m_d(t) = m_d(0) + \int_0^t (\dot{m}_a + \dot{m}_f - \dot{m}_{mix})dt \tag{9}
\]

From Eq. (9), a relatively steady pressure in the downstream CV can be achieved if \( \dot{m}_a + \dot{m}_f = \dot{m}_{mix} \) by maintaining a much larger orifice area into the CV than that out of the CV. In this way, the flow of air and fuel into the intake manifold will replenish the missing mass quickly when it flows out during injection. Accordingly, the orifice areas of the air and fuel metering valves are selected to be bigger than that of the air/fuel mixture injection valve.

Hence, the key point behind this approach is to maintain the upstream pressures of air and fuel to be the same so that the desired air/fuel ratio can be achieved by simply controlling the orifice areas of the metering valves. Above discussions have shown that the idea of controlling \( a_u \) and \( a_f \) to be a desired ratio with a constant \( C_p \) must depend on the same air and fuel upstream pressures. Thus, controlling the air and fuel upstream pressures to be the same is the key for successful air/fuel ratio control. However, the fuel and air supply pressures are usually different and air supply pressure is variable, resulting from the free-piston engine compressor charging the air reservoir and the use of pneumatic actuators depleting it. The vapor pressure of propane (154 psia) is higher than the target injection pressure (50-100 psia). Since it is more convenient to decrease higher pressure to a lower pressure (to be as the same as the air supply pressure), an electronic fuel control valve is used to regulate the fuel mass flow from the fuel source, as shown in Fig. 3. The fuel mass flow into the upstream manifold through this valve is \( \dot{m}_p \). As a result, the upstream pressure of the metering valve on the fuel line is \( P_{buff} \) instead of the original pressure of the fuel source. This pressure needs to be controlled to be the same as the air upstream pressure so that \( P_{air} = P_{buff} \).

To measure the difference between \( P_{buff} \) and \( P_{air} \), a differential pressure sensor is used between the upstream sides of the metering valves on the air and propane lines. The differential pressure is denoted by \( P_{diff} = P_{air} - P_{buff} \). By reading the differential pressure, the electronic on/off valve only opens when \( P_{diff} > 0 \), i.e. the propane pressure \( P_{buff} \) is lower than the air pressure. The control law for the electronic fuel valve is described as,

\[
\text{Fuel} = \begin{cases} 
\text{On} & P_{diff} > 0 \\
\text{Off} & \text{otherwise}
\end{cases} \tag{10}
\]

In this way, if the air upstream pressure drops, the electronic fuel control valve will be closed so that the fuel upstream pressure will drop as well because the upstream fuel flows into the downstream manifold without incoming flow from the fuel source. If the upstream air pressure increases, the electronic fuel valve will be opened to allow for fuel entering into the fuel buffer from the fuel source. In both cases, this control law can effectively regulate \( P_{diff} \) to zero when air supply pressure is changing.

During the injection process, the upstream pressures across the metering valves will drop quickly and will consequentially require rapid on/off action of the electronic fuel valve in order to keep the upstream fuel pressure the same as the upstream air pressure on the air line. Typically, electronic on/off valves offer a tradeoff between response-time and orifice area. If the electronic fuel valve is large enough to support the mass flow of fuel through the metering valve, the response time may be too slow to offer adequate control of the upstream fuel pressure. To alleviate this problem, a buffer cylinder is added after the electronic fuel valve and before the metering valve on the fuel line. Since this buffer creates a large volume at the upstream end of the fuel metering valve, it will provide a capacitance that will decrease the pressure fluctuations of the limited response time electronic fuel valve. This will consequently result in less activity of the electronic fuel valve. Therefore, the buffer helps to keep the upstream fuel pressure at a steady level, equal to the air upstream pressure.

This section has introduced the high-pressure combined air/fuel injection system and how the air/fuel ratio is controlled by 1) orifice areas of the air and fuel metering valves and 2) the air and fuel upstream pressures. In order to verify the validity of the proposed control method, the system is modeled and the controller is then tested in simulation.

**MODELING AND SIMULATION**

Many dynamic engine models have been developed for conventional air/fuel injection system, such as well known Mean Value Engine Model (MVEM) [9]. Similar to this model, the air/fuel injection system was modeled as a lumped-parameter model with a level of fidelity appropriate for only those states of interest, and with accuracy adequate for control purposes. A control volume (CV) approach was taken to model the pressure and temperature dynamics in the combustion constant-volume chamber (subscript “c”), the fuel buffer chamber (subscript “b”), and the downstream tube (subscript “d”), as shown in Fig. 5.

A power balance equates the energy storage rate to the energy flux rate crossing the CV boundaries. The rate form of the first law of thermodynamics is given as follows:

\[
\dot{U} = \dot{H} + \dot{Q} - \dot{W} \tag{11}
\]

By substituting \( \dot{H} \), \( \dot{Q} \) and \( \dot{W} \) and rearranging Eq. (11), the following differential equation can be obtained for the pressure dynamics in a CV (for a more detailed derivation procedure, please refer to [1]):
\[ \dot{P} = (y - 1) \sum m c_{p_{in}} T_{m_{in}} + \frac{(y - 1)Q}{V} - \gamma P V \]  

(12)

where \( P \), \( V \) and \( T \) are the pressure, volume and temperature in the CV, respectively, \( c_p \) is the constant-pressure specific heat of the substance in the CV, and \( \gamma \) is the ratio of specific heats of the substance in the CV. Therefore, pressure dynamics in each CV can be obtained by Eq. (12). Note that, the upstream pressure of the fuel metering valve is \( P_f \); the upstream pressure of the air metering valve is \( P_a \) and the common downstream pressure is denoted as \( P_d \).

The mass flow rates, which connect these CVs, are modeled through four channels: 1) fuel flow rate through a controlled electronic on/off valve (\( \dot{m}_p \)); 2) fuel flow rate through the fuel metering valve (\( \dot{m}_f \)); 3) air flow rate through the air metering valve (\( \dot{m}_a \)); 4) air/fuel mixture flow rate through the electronic on/off injection valve (\( \dot{m}_{mix} \)). These mass flow rates are calculated by Eq. (1-4).

In this paper, two electronically controlled on/off valves (fuel and injection) are modeled as first order dynamic system:

\[ \dot{a} = \frac{1}{\tau_a} (u - a) \]  

(13)

where \( a \) is the orifice area of the valve, \( \tau_a \) is the response time of the valve and \( u \) is the control input of the valve.

Since the check valve has dynamic characteristics that influence its flow area, it has to be properly modeled so that \( \dot{m}_{mix} \) can be computed. Applying Newton's second law to the diagram in Fig. 6, the valve dynamics is thus given,

\[ \dot{m}_{cv} x_{cv} = (P_u - P_a) A_{cv} - b_{cv} x_{cv} - k_{cv} x_{cv} \]  

(14)

where \( m_{cv} \) is the mass of the movable part of the check valve (plunger), \( x_{cv} \) is the position of the plunger (\( x_{cv} = 0 \) when it is sealed), and \( A_{cv} \) is the cross-sectional area of the plunger head. Furthermore, the valve flow area \( a_{cv}(x_{cv}) \) can be described by the following,

\[ a_{cv}(x_{cv}) = \min \left\{ \frac{2\pi r_{cv} x_{cv}}{\pi r_{cv}^2} \right\} \]  

(15)

where \( r_{cv} \) is the radius of the valve head.

The lumped-parameter model described above was built in Matlab Simulink. In this section, the following simulation results during the injection process will be demonstrated: 1) the difference of upstream pressures of the air and fuel supply; 2) the downstream pressure of the air/fuel injection line; 3) the air/fuel mass flow ratios entering the intake manifold and the combustion chamber. In the simulation, the air supply pressure \( P_u \) is considered to be constant at 372.4 kPa (54psia), and propane supply pressure is also constant at 1Mpa (154psia). The initial buffer pressure and downstream pressure (intake manifold) are equal to air supply pressure \( P_u = P_d \). The initial combustion chamber pressure is set to be atmospheric. Initial temperatures in all CVs are ambient. The injection duration for the combustion chamber is 30ms.

Substituting an upstream pressure of 372.4 kPa and a downstream pressure range of (329.5, 372.4) kPa to Eq. (1), the ratio \( C_p = \psi_p / \psi_f \) can be calculated to be within a range of
which varies so little that it can be considered as a constant 1.55. According to Eq. (8), \( \alpha_f/\alpha_a \) is then set to 10.08 for the metering valves for achieving the stoichiometric air/fuel ratio (15.63 for propane). Therefore, a stable air/fuel ratio can be achieved with this \( C_w \) as long as the upstream pressures of air and fuel supply are controlled to be the same. The simulation measures the upstream pressure difference between air supply and fuel buffer, denoted by \( P_{\text{diff}} = P_a - P_b \), as indicated in Fig. 5. According to the control law described by Eq. (10), the on/off fuel control valve is opened when \( P_a > P_b \), otherwise it is closed. Fig. 7 shows the buffer and the downstream pressure dynamics during one injection cycle. Upon injection to the combustion chamber, the downstream pressure quickly drops but it will be compensated by air/fuel refilling from both air and fuel supply lines. This control method results in very small variations of the buffer pressure as desired, and the buffer pressure closely tracks the upstream air pressure. The downstream pressure variation is very small as well, which can be seen from the bottom figure in Fig. 7.

\[
m_{d, \text{air}}(t) = m_{d, \text{air}}(0) + \int_0^t (\dot{m}_a - \dot{m}_{\text{mix, air}}) \, dt \tag{16}
\]

\[
m_{d, \text{fuel}}(t) = m_{d, \text{fuel}}(0) + \int_0^t (\dot{m}_f - \dot{m}_{\text{mix, fuel}}) \, dt \tag{17}
\]

Assuming the air/fuel ratio of the mass leaving the CV is proportional to that of the total mass ratio inside the CV, we then have,

\[
\dot{m}_{\text{mix, air}} = \frac{m_{d, \text{air}}}{m_f} \dot{m}_{\text{mix}} \tag{19}
\]

\[
\dot{m}_{\text{mix, fuel}} = \frac{m_{d, \text{fuel}}}{m_f} \dot{m}_{\text{mix}} \tag{20}
\]

Hence, the air/fuel ratio entering the combustion chamber is given as,

\[
\lambda_c = \frac{\dot{m}_{\text{mix, air}}}{\dot{m}_{\text{mix, fuel}}} \tag{21}
\]

Based on above assumption, \( \lambda_c \) is also the air/fuel ratio of the mixture in the downstream CV. Fig. 9 shows the air/fuel ratios entering and inside (also leaving) the downstream CV in the first one second injection operation. In the top figure, \( \lambda_c \) is close to propane’s stoichiometric ratio during the injection process. Note that \( \lambda_c \) is undefined when there is no injection into the combustion chamber because \( \dot{m}_a \) and \( \dot{m}_f \) both are zero.

\[
m_f(t) = m_{d, \text{air}}(t) + m_{d, \text{fuel}}(t) \tag{18}
\]

Although both entering and leaving air/fuel ratios are not perfectly constant, they are controlled to be close to stoichiometric air/fuel ratio, 15.63 for propane. Initially the air/fuel ratio entering the combustion chamber (bottom figure) is far below the ideal ratio of 15.63 because the combustion chamber is full of pure air in the very first injection. Therefore, misfiring is expected in the practical case due to the fact that
the air/fuel ratio of the mixture injected into the combustion chamber is not stoichiometric in the first injection. However, the stoichiometric ratio can be quickly achieved by ‘flushing’ the combustion chamber for a just few cycles.

**EXPERIMENTAL RESULTS**

Based on the above discussions, an experimental setup was built to validate the model and verify the control approach. The system is shown in Fig. 10. The electronic valve used here is a Parker series 9 on/off valve with an 8 ms response time. The differential pressure sensor is Honeywell 26PC series. From Eq. (8), one can see that the mass flow ratio between air and propane are dependent on the ratio of their effective flow areas. In order to further simplify the air/fuel injection circuitry, the metering valve on the air line can be removed in the experimental setup. The smallest flow area through the air supply line now determines the effective flow area, which is the check valve in our experimental setup. This simplification maximizes the effective flow area on the air supply line, to comply with the desired operational frequency of the device.

Having set a desired $a_d/a_f$, the key control objective is to control the buffer pressure (fuel upstream pressure) to be the same as the air upstream pressure. A differential pressure sensor was used to read this pressure difference, as shown in Fig. 11. The upstream fuel pressure was controlled to track the air pressure, indicated by the small variation of pressure difference (-30 to 10 kPa).

The validity of Eq. (8) is based on the assumption that $C_p$ is a constant. In experiments, the downstream pressure variation is within a very small range, as predicted by the simulations. Fig. 12 shows the comparison of the downstream pressure dynamics between simulation results and the experimental data, both with 372.5 kPa air supply pressure. This pressure dynamics is influenced by the dynamics of two check valves on air and fuel lines and the injection dynamics when air/fuel mixture is injected into the combustion chamber. Hence, this dynamics comparison also validates the simulation models of the system.

Although precise flow meters were not available to experimentally measure the air/fuel ratio, the consistent combustion results verify the validity of the proposed control methodology. With different air supply pressures and engine operation frequencies, continuous and successful combustions are achieved. Fig. 13 shows the experimental results with different injection frequencies, 10Hz on top and 2Hz on
The air-supply pressure was chosen to be 372 kPa for both tests. Initial misfirings can be observed in the 10Hz experimental results. However, once the combustion starts, it is continuous and reliable. Initial misfirings are mainly the result of an incorrect mixture ratio initially present in the feed lines.

CONCLUSIONS

A high pressure combined air/fuel injection system was designed, modeled, controlled and tested. The simulation and experimental data show the validity of the proposed air/fuel ratio control approach. It provides a compact and effective air/fuel injection and control solution for small scale engines, especially free-piston engine applications. For larger scale engines, the air and fuel valves would require much larger orifice areas with fast response times. The challenges of scaling such valves therefore make the proposed concept of high pressure air and fuel injection more appropriate for small-scale engine systems. The primary advantages of this concept for small-scale engines are increased power density, the absence of the need for idling, and trivial start-stop operation. The proposed control approach was shown in simulation and experimentally. Despite unmodeled effects, the modeled response of the controlled system showed good agreement with experimental results.

REFERENCES


