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Mean Flow Regulation of a High Frequency Combustion Control Valve Based on Pulse Width Modulation and System Identification

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Abstract— **Strong combustion instabilities within gas turbine engines adverse combustion efficiency, shorten engine life cycles and may even cause hardware or structure failures. Fuel modulations, which introduce externally unsteady heat release rate perturbations out of phase with pressure oscillations, is an effective and practical approach for combustion instability control. A high frequency fuel valve capable of large fuel modulations (above 30% of mean flow) up to 750 Hz is presented in this paper. Fuel modulations are achieved by pushing fuel out of the valve cavity using a Terfenol-D rod that extends or contracts with external magnetic fields, and mean flow is controlled by a step motor using pulse width modulation. However, this valve suffers from significant variations of mean flow when starting fuel modulation. To follow the flow command and effectively reject strong interferences of fuel modulations on mean flow, a LQG pulse width modulation controller based on closed-loop system identification is developed, which achieves faster response than a traditional proportional derivative controller. With effective mean flow regulation, this fuel valve damps out strong pressure pulsations in an unstable swirling atmospheric combustor up to 23 dB.**

I. Introduction

Combustion instability refers to the self-sustained strong combustion oscillations in a combustion process due to positive coupling between pressure and heat release rate[1]. Since 1940s', extensive research has been done on dump combustors or bluff-body stabilized combustors which are typical of rockets, afterburners and ramjet engines [2][3]. In recent years, considerable attention has been focused on swirl-stabilized lean premixed gas turbine combustors [4][5]. Compared with the traditional diffusion combustors, lean premixed combustors generate very low emissions because of the relatively low flame temperature and short residence time. However, they are more susceptible to combustion instabilities and lean blowout. Fuel modulation has been widely used for combustion instability control in laboratories and several large scale industrial rigs. Sensors, actuators and control algorithms are among the main components of a combustion control system [6]. Pressure sensors are reliable, mature and easy to implement. Different from optical sensors, they are not contaminated by carbon deposits. Pressure signals also have higher signal noise ratios than optical fibers. Combustion process is highly nonlinear, multiple dimensional and time-varying. However combustion instability in essence is a nonlinear limit cycle behavior. In many cases, approximating the limit cycle behavior as a linear oscillator allows effective control of combustion instabilities, such as phase shift [7], LQG/LTR [8] and adaptive Smith control [9].

A fuel valve capable of large amplitude fuel modulations up to 1000 Hz is necessary for combustion instability control. An ideal high frequency fuel valve should possess high reliability, wide bandwidth up to 1000 Hz, large fuel modulations (usually 30% of mean flow suffices), proportionality, small size and weight. A gaseous fuel injector implemented in the pilot fuel passage capable of fuel modulations 0.2 g/s below 1000 Hz has been developed at Georgia Tech. This valve achieves fuel modulation by modulating fuel flow area using a magnetostrictive rod [10]. A spinning valve actuator capable of fuel modulations 30% of mean flow up to 800 Hz is developed at United Technology Research Center [11]. A flapper valve manufactured by Jansen Aircraft has been applied for combustion instability control [12].

The high frequency fuel valve presented in this paper achieves large fuel modulations up to 750 Hz. Experiments and theoretical analyses have been performed to optimize valve performance. However significant variations of mean flow occur when this valve starts fuel modulation, which is undesirable for combustion instability control. For fast regulation of mean flow, a pulse width modulation LQG (Linear-Quadratic-Gaussian) controller based on closed-loop system identification is developed. Successful attenuation of combustion instabilities in an unstable atmospheric swirling combustor is achieved using this fuel valve.

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II. Necessity of Fast Mean Flow Regulation

A. Experiment Setup

Fig. 1 shows the Goodrich high frequency combustion control valve, and fig. 2 is a sketch of its internal structures. The fuel injector is located 1 m downstream of the valve and fuel modulation is measured 0.4 m downstream of the valve. Instantaneous fuel flow rate is obtained from pressure drop across a throttle orifice measured by a Sentec differential pressure transducer with bandwidth 2 kHz.

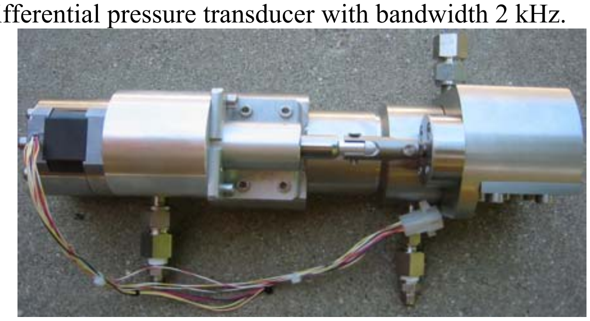

Fig. 1 Goodrich High Frequency Fuel Valve

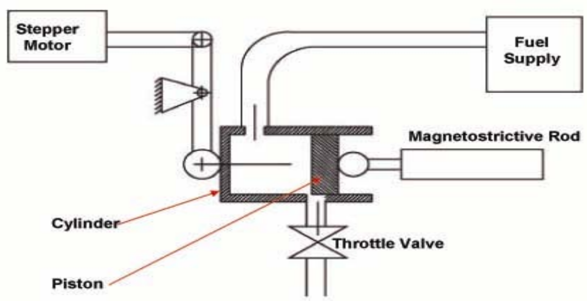

Fig. 2 Schematic of Valve System

Fuel modulation is achieved by pushing fuel out of the valve cavity using a Terfenol-D rod which extends or contacts with external magnetic field generated by an unsteady current going through a surrounding coil [13]. For simplicity, the response of rod displacement to input voltage is modeled as a second order low-pass filter $\frac{Y(s)}{u(s)}=\frac{\beta_1}{s+\beta_2}\cdot\frac{\beta_3}{s+\beta_4}(\beta_1,\beta_2,\beta_3,\beta_4>0)$. The first term on the right side of the equation accounts for the decreasing current with frequency, and the second term accounts for the decreasing displacement of Terfenold-D rod with frequency in the same external magnetic field [13].

Mean flow rate of this valve is controlled by an embedded stepper motor using pulse width modulation (PWM). Jogging direction is controlled by switching between two terminals, fuel lean and fuel rich. The mean flow rate will increase if "fuel rich" terminal is connected, and will decrease if "fuel lean" terminal is connected. The stepper motor has a much faster dynamics than the fluid circuit, so the response of displacement to PWM control input can be simply modeled as a proportional unit $\frac{x_m(s)}{u(s)}=-\alpha_m$ (α_m>0).

B. Fuel Modulations of the High Frequency Valve

Low order models and experiments have been performed to optimize the valve performance [13]. The fluid circuit and piston-cylinder structure are sketched in fig. 3 and fig. 4 respectively. TARS refers to the Goodrich triple annular research swirler that severs as a fuel injector and an air swirler [14].

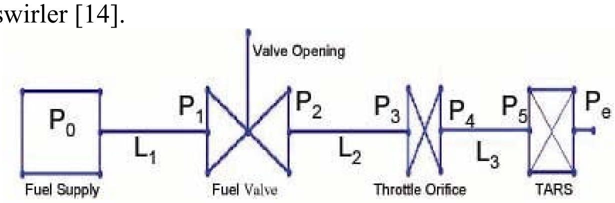

Fig. 3 Schematic of Fluidic Circuit

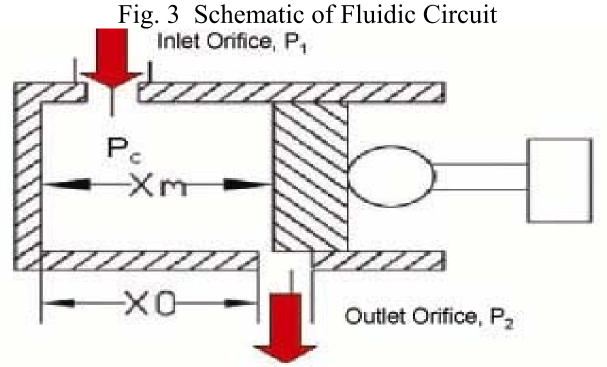

Fig. 4 Sketch of Piston-Cylinder Structure

By applying unsteady Bernoulli's equations to fuel tubes 1, 2 and 3, and considering finite compressibility of fuel within the piston-cylinder cavity based on the definition of fuel modulus and mass conservation, the response of fuel modulations (volume flow rate) to piston movement can be derived as [13]:

$$\frac{\dot{Q}_2(s)}{x_m(s)}=\frac{A_tK}{\rho x_{m0}}\frac{L_1s^2+\zeta s}{[(L_2+L_3)s+\beta](L_1s^2+\zeta s+\psi)+\psi}$$

$$\zeta=\dot{Q}_0A_t\left(\frac{1}{A_t^2}+\frac{1}{A_{e1}^2}\right);$$

$$\beta=\dot{Q}_0A_t\left[\frac{1}{A_{e2}^2(x_{m0}-x_0)^2}+\frac{1}{A_{e3}^2}+\frac{1}{A_{e4}^2}\right];$$

$$\psi=\frac{KA_t}{A_cx_{m0}\rho}$$

For a small mean flow rate, a small valve cavity, a large valve inlet and outlet, a large orifice and large fuel injection ports of TARS, the above equation can be approximated as $\frac{\dot{Q}_2(s)}{x_m(s)}\approx\frac{A_tK}{\rho x_{m0}}\frac{L_1s}{(L_2+L_3)(L_1s^2+\psi)}$. Denotations of the variables are shown in table 1.

Detailed analyses and experiments show that fuel modulations can be improved by implementing the valve as close as possible to the combustion rig. Shorter fuel tubes with larger diameters, a larger valve cavity section area, fuels with smaller modulus, a smaller valve inlet orifice, a higher fuel supply pressure and a larger mean flow also favor large fuel modulations.

Fig. 5 shows the fuel modulation percentage that is defined as the fuel flow rate RMS to the mean fuel flow rate. The mean fuel flow rate is 2.5 g/s. The upstream tube is 0.45 m with ID 3 mm, and the downstream tube is 0.33 m with ID 3 mm.

TABLE I NOMENCLATURE

Symbol	Quantity	Unit
γ	Fuel kinematic viscosity	m^2/s
$A_s + A_t$	Cross-section area of valve piston and fuel tube	m^2
K	Fuel modulus	Psi
$L_{1,2,3}$	Fuel tube length	m
$x_w - x_0$	Valve outlet openning	m
x_m	Length of valve cavity	m
P_s	Dynamic pressure downstream of fuel injector,	Pa
Subscript 0	Mean quantities	

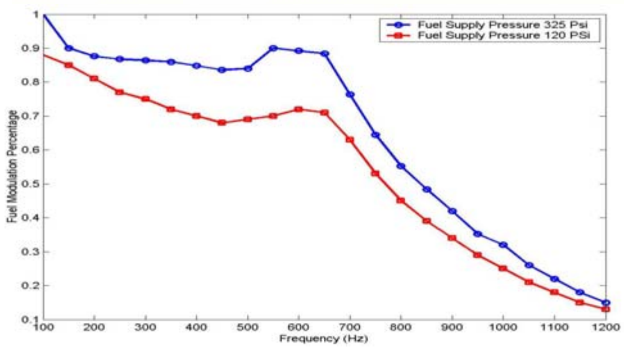

Fig. 5 Fuel Modulation Percentage with Sinusoidal Forcing

C. Mean Flow Variations with Fuel Modulation

Fig. 6 shows large variations of mean flow when the valve starts fuel modulation. Fig. 7 shows that combustion instability would not be well controlled if the mean flow rate changes or mean flow regulation is too slow. To reject the interferences of fuel modulations on mean flow and follow the flow command, it is necessary to develop a fast mean flow controller.

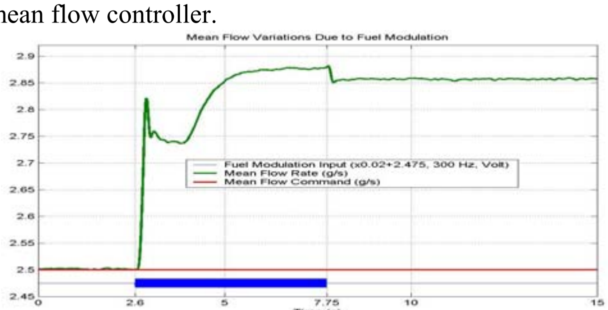

Fig. 6 Mean Flow variations due to Fuel Modulation

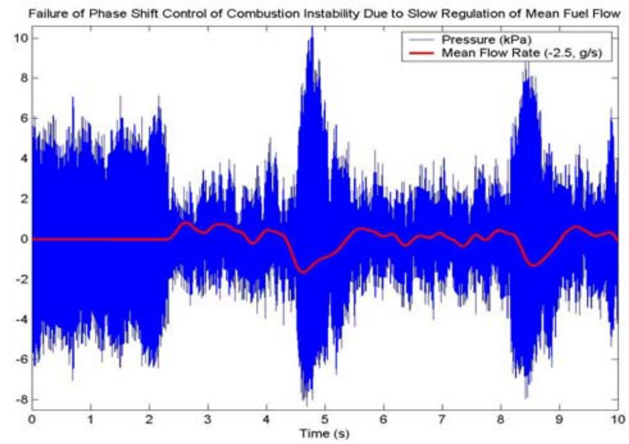

Fig. 7 Failure of Combustion Control Due to Slow Regulation of Mean Fuel Flow

III. Closed Loop System Identification and LQG Control Design for Mean Flow Regulation

A. System Identification

A simple proportional-derivative (PD) pulse width modulation controller is first developed for closed-loop system identification (fig. 8). A Dspace control desk is used to sample the fuel flow rate and output TTL signals to the stepper motor. By trial and error, the best proportional gain is found to be 0.8 and derivative gain to be 0.1. A pulse width modulation scheme converts the continuous control input into duty cycles (0~90%) with frequency 20 Hz. Fig. 9 shows the PWM diagram. The PWM saturation limit is ± 2.5, and saturation gain is 0.36. Step response of the PD controller is shown in fig. 10. The transient period of the PD controller is rather long, more than one second.

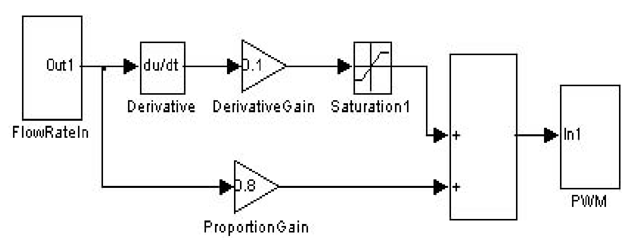

Fig. 8 Matlab Simulink Model of PD Controller

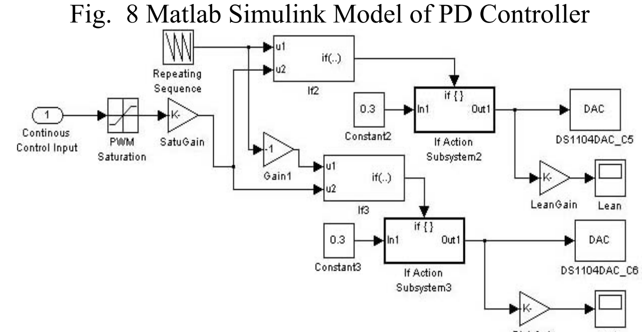

Fig. 9 Matlab Simulink Model of Pulse Width Modulation

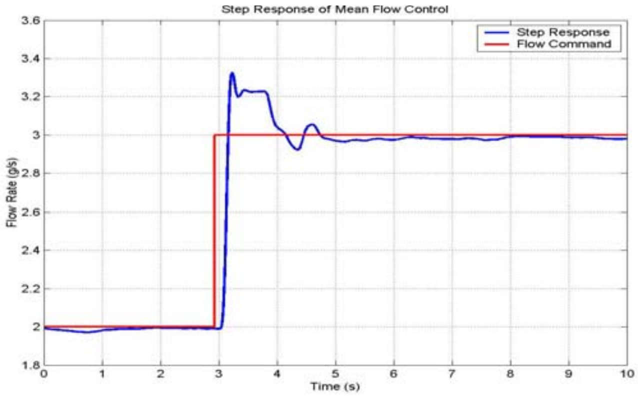

Fig. 10 Step Response of the PD Controller

Fig. 11 shows the Matlab Simulink model for system identification. The standard deviation of white noise is 1.5, and the period of repeating sequence is 10 ms. 100,000 data points are sampled for system identification, and the first half of them are used to obtain the model and the second half are used for model validation. Fig. 12 show the system identification input signal and output for the first 50,000 data points. Transfer function is derived as $\frac{\dot{m}(s)}{u(s)} = -\frac{0.004198s^3 - 1389s^2 - 6.284E6s - 6.431E10}{s^4 + 7287s^3 + 4.155E7s^2 + 2.744E11s + 3.033E9}$, with loss function 0.0014954 and final prediction error 0.00149612. This model shows that the system is of non-minimum phase, i.e. there are zeros lying on the right half phase plane. The eigenvalues of the system are $[-1.1; -157.1 + 6271i; -157.1 - 6271i; -6971.3]$. The following LQG control design is based on this fourth order model.

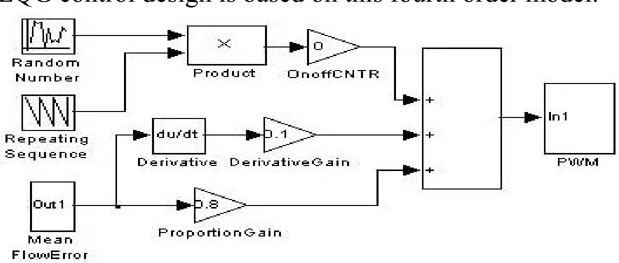

Fig. 11 Simulink Model for System Identification

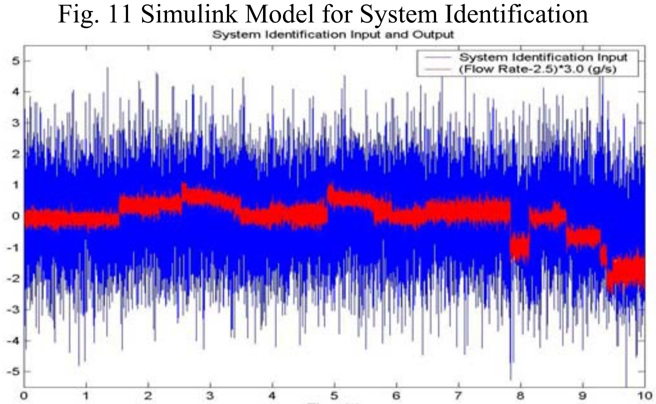

Fig. 12 System Identification Input and Output

B. Model Validation

Autocorrelation function of model error and cross-correlation function between model error and input up to lag 25 for a new dataset are shown in fig. 13. It is clear that majority of the 25 lag points fall within 99% confidence interval.

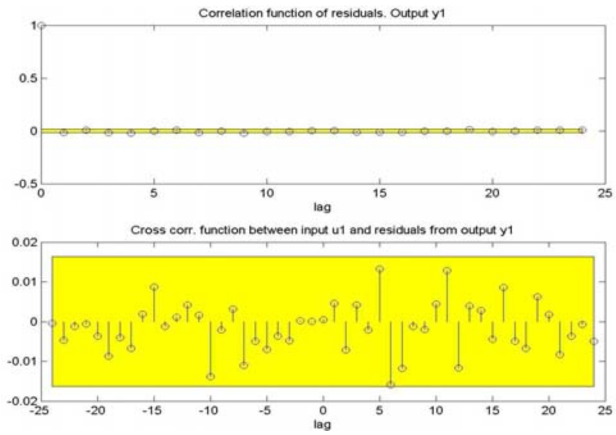

Fig. 13 Residual Analysis

C. LQG Mean Flow Regulation

A standard LQG control design procedure is performed based on the 4[th] order system identification model [15]. After obtaining the state feedback gain K and Kalman filter observer gain L, the LQG controller is expressed as $\dot{\hat{X}} = A_c\hat{X} + Ly$, with $u = -K^T\hat{X}$

$$L = 0.001\begin{pmatrix} 0.3067 \\ 0.005 \\ -0.0027 \\ -0.0013 \end{pmatrix}; K = 1000\begin{pmatrix} -7.0633 \\ -3.3158 \\ 8.3638 \\ 3.4277 \end{pmatrix}$$

$$A_c = 1000\begin{pmatrix} -0.0236 & -0.0171 & 0.0295 & 0.0078 \\ -1.2683 & -0.9643 & 8.1566 & -1.2284 \\ 1.5858 & -3.8604 & -2.0710 & 3.2520 \\ 0.4633 & -1.4697 & -3.0432 & -7.2089 \end{pmatrix}.$$

Fig. 14 shows the step response of the LQG controller, which is much faster than that of the PD controller in fig. 10. Fig. 15 shows that the LQG controller effectively rejects the interference of fuel modulation on mean fuel flow rate. The mean flow oscillation peak has been reduced by eight times.

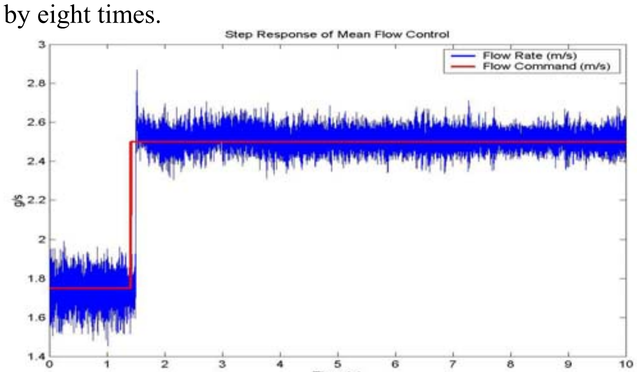

Fig. 14 Step Response of the LQG Controller

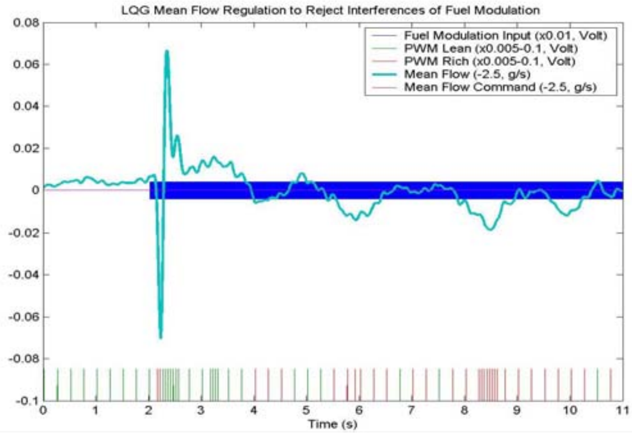

Fig. 15 LQG Control of Mean Flow with Fuel Modulation

IV. Combustion Control with Fuel Modulation

A. *Thermo-Acoustic Response to Fuel Modulation*

Fuel modulation will introduce delayed heat release and pressure oscillations. Time delay due to fuel atomization, air/fuel mixing and evaporation, ignition delay and convective time delay may considerably complicate combustion control design. Due to kinematics of chemical reactions (usually about 10 to 20 ms), high frequency fuel modulations cannot be fully converted to heat release. Fig. 16 shows fuel modulations, heat release and pressure for sinusoidal forcing input at 700 Hz with amplitude 0.45 Volt. The mean equivalence ratio is 0.43 and power is 105 kW.

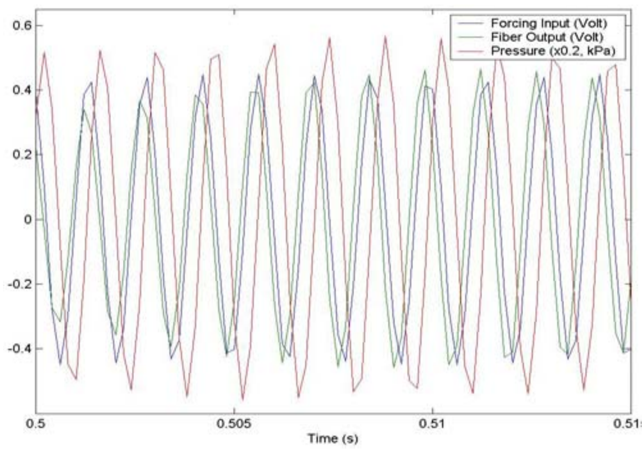

Fig. 16 Response of Pressure and Heat Release to Sinusoidal Fuel Modulation at 700 Hz

B. *Phase Shift Control of Combustion Instability*

Phase-shift fuel modulation is applied to stabilize an unstable atmospheric swirling combustor fueled with turpentine. This combustion rig features a triple annular research swirler (TARS) with distributed fuel injections and a stainless combustion chamber 26'' long with internal diameter 4'' [13][14]. Unstable combustion is observed at equivalence ratio 0.63 and power 120 kW, with the baseline spectrum shown in fig. 17. The unstable mode corresponds to quarter wave mode of the combustion chamber. It is clear that the signal noise ratio of pressure is about 20 dB higher than that of the optical fiber output.

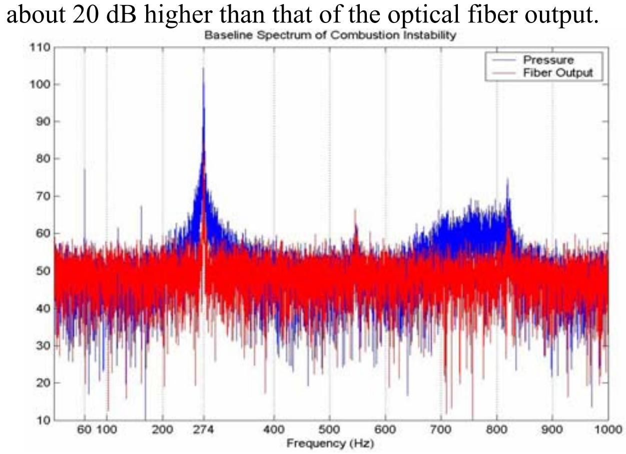

Fig. 17 Baseline Spectrum of Combustion Instability

Equivalence ratio variations may be the dominant mechanism for combustion instability within this rig. With strong pressure pulsations, there may be significant variations of air flow rate into the combustion chamber since the pressure drop across the air swirler is usually 3%~6% of inlet static pressure while the pressure oscillation amplitude can be up to 3 kPa. The fuel flow rate variations induced by pressure pulsation is only 0.25% of the mean fuel flow rate, as shown in fig. 18. Another possible mechanism would be heat release rate oscillations caused by interactions among shear layer flame, precessing vortex core, corner recirculation zone and swirling shear layer helical modes [13][14]. Detailed investigations and modeling will be reported later.

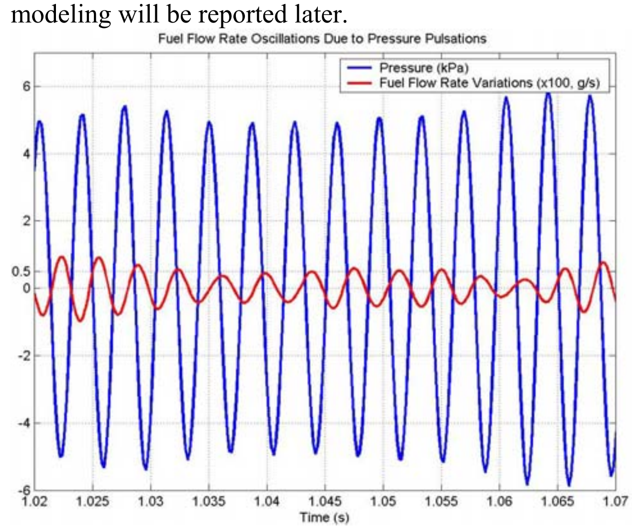

Fig. 18 Pressure Pulsations and Fuel Flow Oscillations

Fig. 19 shows the Matlab Simulink model for phase shift control of combustion instability. Pressure is first sensed by a water-cooled microphone PCB106B, then sampled using Dspace board CP1104, and then band-filtered (225-325Hz), delayed, amplified and magnitude-limited, and finally sent out for fuel modulation. Fig. 20 and fig. 21 show that pressure is reduced by 23 dB within 200 ms with phase shift gain 0.008 and control delay 2.2 ms. Fig. 20 also shows that mean flow variations due to interferences of fuel modulation are effectively suppressed.

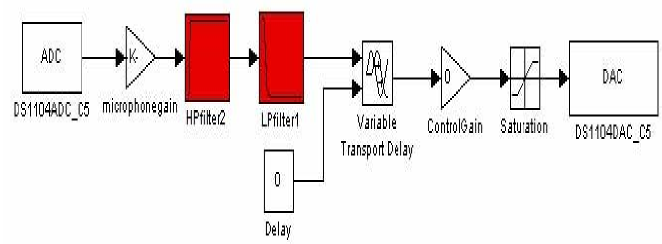

Fig. 19 Phase Shift Control Diagram

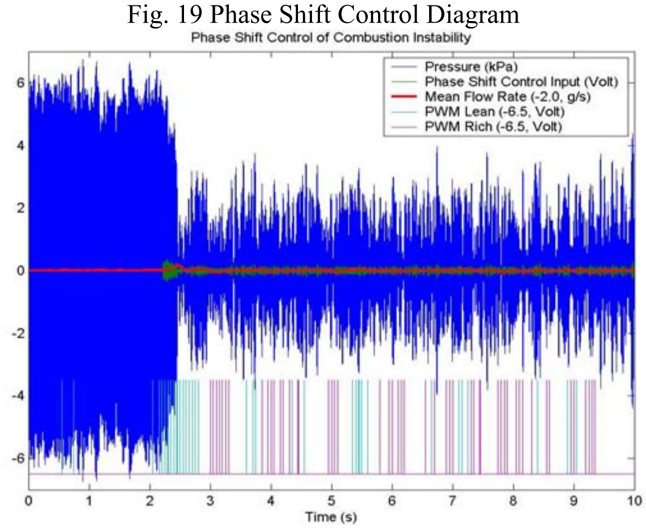

Fig. 20 Phase Shift Control of Combustion Instability

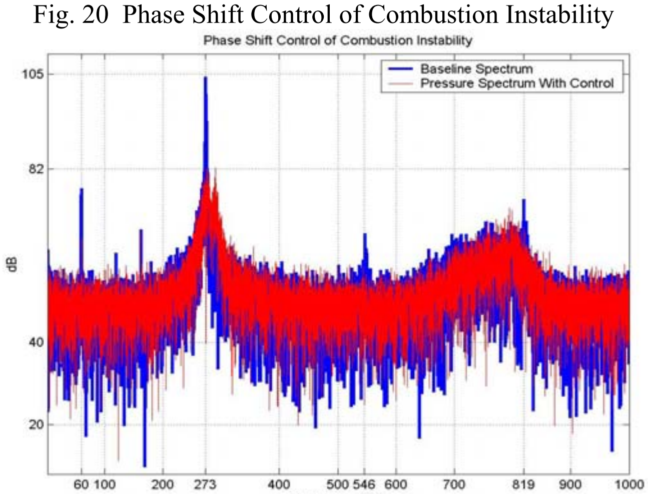

Fig. 21 Phase Shift Control of Combustion Instability

V. CONCLUSION

An active combustion control valve capable of large amplitude high frequency fuel modulations is presented. To reject strong interferences of fuel modulations on mean flow and follow the flow command, a fast LQG pulse width modulation controller based on closed-loop system identification is developed. Successful applications of the fuel valve to combustion instability control are demonstrated in an unstable swirling combustor.

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