

Small Signal Point-to-Point Tracking of a Propellant Mixer *

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Abstract

This paper addresses some theoretical modelling and control issues for a mixing chamber used in rocket engine testing at NASA Stennis Space Center. The mixer is responsible for combining high pressure LH2 and GH2 to produce a hydrogen flow that meets certain thermodynamic properties before it is fed into a test article. The desired properties are maintained by precise control of the LH2 and GH2 flows. The mixer is modelled as a general multi-flow lumped volume for single constituent fluids using density and internal energy as states. The set of nonlinear differential equations is modelled in the SIMULINK environment including a table look-up feature of the fluid thermodynamic properties. A small-signal (linear) model is developed based on the nonlinear model and simulated as well. Pulse disturbances are introduced to the valve positions and the quality of the linear model is ascertained by comparing its behavior against the nonlinear model simulations. Valve control strategies that simulate an operator-in-the-loop scenario are then explored demonstrating the need for automatic feedback control. Finally, classical optimal single-output and multi-output Proportional/Integral controllers are designed based on the linear model and applied to the nonlinear model with excellent results to track simultaneous, constant setpoint changes in desired exit flow, exit temperature, and mixer pressure, as well as to reject unmeasurable but bounded additive step perturbations in the valve positions.

1 Background

The Engineering Division at NASA John C. Stennis Space Center (SSC) continues its efforts to assemble a software simulation package that captures the static and dynamic characteristics of modern

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and future thermodynamic systems. The package is foreseen to fulfill the need for an accurate and verifiable thermodynamic system simulation environment [1, 2, 3, 4].

In this work, we build mainly upon [4] and focus on the point-to-point tracking control problem for an LH_2 and GH_2 mixer subsystem and associated flow controllers. The relevant components of the mixer subsystem are shown in the schematic of Figure 1. The LH_2 valve controls the flow of high

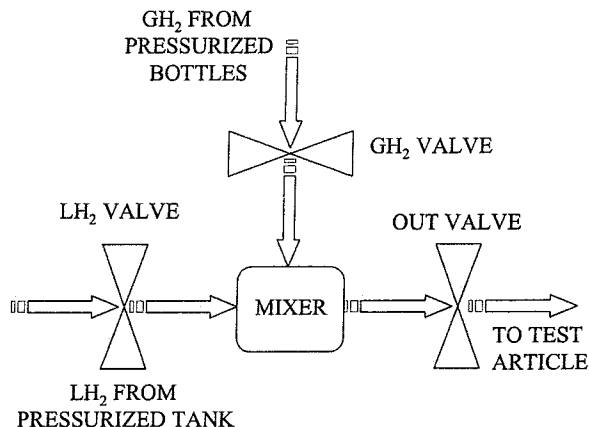


Figure 1: Diagram of HPH2 Mixer and Control Valves.

pressure liquid H_2 from a pressurized tank; the GH_2 valve controls the flow of gaseous H_2 from a set of high pressure bottles; and the output valve controls the flow of H_2 into the test article. The primary objective of the mixer and control valves is to regulate the thermodynamic properties of the out flow of H_2 for optimal performance in spite of the steady depletion of the source bottles, measurement errors, modelling imperfections, and other uncertainties and perturbations.

In [4], we developed an analytical model of the mixer and control valves suitable for dynamic analysis and control design in the vicinity of a thermodynamic equilibrium point. An optimal linear-quadratic regulator was designed and simulated using Matlab to illustrate the effectiveness of such a controller in compensating against perturbations that result in small deviations around the desired equilibrium. The simulations also indicate how one could study which valve or combination of valves is more effective in achieving a desired performance (control authority or relegation).

In this article, the nonlinear mixer equations are simulated in the SIMULINK environment, including a NIST-12 property data look-up table incorporated in the model and continuously accessed via Matlab functions [5]. The small-signal model developed in [4] requires the numerical computation of several partial derivative terms which in turn depend on the equilibrium point. Since there are no analytical expressions that would allow the direct calculation of some of these partial derivatives, a Matlab routine, in conjunction with the property look-up table, computes these terms as needed. The model is then

used to evaluate open-loop control strategies simulating an operator-in-the-loop scenario and testing that reasonable results are obtained based on past operator experience. In the future, we hope to validate the model with real test data as it becomes available. The simulations section describe closed-loop control strategies in more detail based on 1) an optimal multi-input/single-output (MISO) PI controller to track the exit flow only; and 2) an optimal multi-input/multi-output (MIMO) PI controller to track the exit flow, the mixer temperature, and the exit temperature simultaneously and in the presence of step perturbations in the valve positions.

2 Mixer and Valve Models

The mixer has been modelled as a general multi-flow lumped volume for single constituent fluids using density and internal energy as states [1, 4]. The model was independently derived in [5]. In this article, we refer to the mixer’s internal energy and density as the “state”, not to be confused with the definition of state of a system as described by its thermodynamic properties. Letting $z_1(t)$, $z_2(t)$ denote the mixer’s specific internal energy and density, respectively, the two dynamic equations for a constant volume V are given by

$$\dot{z}_1 = \frac{1}{V z_2} [E_{net} - W_{net} z_1 + \dot{Q}] \quad (1)$$

$$\dot{z}_2 = \frac{1}{V} W_{net} \quad (2)$$

where E_{net} is the net (input minus output) energy into the mixer, W_{net} is the net (input minus output) flow, and \dot{Q} is the total heat transfer rate. The net flow is simply

$$W_{net} = w_g + w_l - w_e$$

where the subscripts stand for “gas”, “liquid”, and “exit”, respectively. In this study, only positive flows are considered, that is, the gas and liquid flows are into the mixer and the output flow is out of the mixer. In essence, normal operation requires that each valve always experience a positive pressure difference across it thus disallowing flow reversal. The net input energy term is given by

$$E_{net} = h_g w_g + h_l w_l - h_v w_e$$

where h_g , h_l , h_v denote the enthalpy of the GH_2 stream, LH_2 stream, and volume block (mixer), respectively.

The liquid and exit valves are modelled by an *incompressible* fluid flow expression that compute the flow w_f given the inlet P_i and outlet P_o pressures, the density of the inlet stream ρ_i , and the valve flow coefficient C_{vf} , as follows:

$$w_f = \sqrt{\rho_i \rho_w (P_i - P_o)} C_{vf} = \alpha_f C_{vf} \quad (3)$$

where ρ_w is the density of water (62.4 lbm/ft^3), and we have assumed that $P_i > P_o$. The gas valve on the other hand is more appropriately modelled by the *compressible* fluid flow expression

$$w_f = \begin{cases} 2.857 \times 10^{-2} \frac{\sqrt{T_i \rho_i}}{P_i} \sqrt{P_i^2 - P_o^2} C_{vf}, & P_o < P_i < 2P_o; \\ 2.423 \times 10^{-2} \sqrt{T_i \rho_i} C_{vf}, & P_i \geq 2P_o. \end{cases} \quad (4)$$

where the mass flow is in lbm/s , the pressure in psi , the density in lbm/ft^3 , and the temperature T in $^\circ\text{R}$. By design and under normal operation, the properties of the inlet hydrogen gas and liquid are such that $P_i, P_g > P_o = P_v$ and also $P_v > P_e$; thus, the possibility of reverse flow is not considered in this study but can certainly be added if so required.

2.1 Small-Signal Model

Substituting the expressions for the valve flows and the net internal energy and net flow into equations (1)-(2), and treating the valve coefficients as inputs to be regulated, the dynamic model of the mixer is a 2-state, 3-input system of nonlinear differential equations of the form

$$\dot{z}_1 = F_1(z_1, z_2, C_{vg}, C_{vl}, C_{ve}) \quad (5)$$

$$\dot{z}_2 = F_2(z_1, z_2, C_{vg}, C_{vl}, C_{ve}) \quad (6)$$

where $F_1(\cdot)$ and $F_2(\cdot)$ are nonlinear functions of the state and valve coefficients.

For the remainder of this article, we will denote constant or equilibrium values of any variable by an upper bar ($\bar{\cdot}$). Given constant values of valve flow coefficients $\bar{C}_v = [\bar{C}_{vg} \ \bar{C}_{vl} \ \bar{C}_{ve}]^T$ (superscript T denotes transposition) and constant fluid properties, the state of the model

$$z(t) = \begin{bmatrix} z_1 : & \text{Internal Energy} \\ z_2 : & \text{Density} \end{bmatrix}$$

reaches a constant equilibrium point \bar{z} . Next, consider perturbing such an equilibrium by small signals $x(t)$ and $u(t)$ so that

$$z(t) = \bar{z} + x(t) \quad \text{and} \quad C_v(t) = \bar{C}_v + u(t)$$

where $u(t)$ denotes a small, valve-coefficient correction/regulation signal. Then, a standard linearization of equations (5)-(6) results in the small signal model

$$\dot{x} = Ax + Bu \quad (7)$$

where $x(t)$ is the small perturbation state vector, $u(t)$ is the small perturbation control signal, and the two-by-two matrix A and two-by-three matrix B are given by

$$A = \begin{bmatrix} \frac{\partial F_1}{\partial z_1} & \frac{\partial F_1}{\partial z_2} \\ \frac{\partial F_2}{\partial z_1} & \frac{\partial F_2}{\partial z_2} \end{bmatrix}; \quad B = \begin{bmatrix} \frac{\partial F_1}{\partial C_{vg}} & \frac{\partial F_1}{\partial C_{vl}} & \frac{\partial F_1}{\partial C_{ve}} \\ \frac{\partial F_2}{\partial C_{vg}} & \frac{\partial F_2}{\partial C_{vl}} & \frac{\partial F_2}{\partial C_{ve}} \end{bmatrix}$$

where the partial derivatives are evaluated at the equilibrium state \bar{z} and constant valve flow coefficient vector \bar{C}_v .

Output equations of the form

$$y = Cx + Du$$

where matrices C and D are appropriately dimensioned are easily appended to the model (7) to account for the measurement of certain variables such as temperature, pressure, or flow. An output of interest is the exit flow w_e . Using equation (3) and linearizing around a chosen equilibrium point, we obtain the linear approximation

$$w_{elin} = \begin{bmatrix} \frac{\partial \alpha_e}{\partial z_1} & \frac{\partial \alpha_e}{\partial z_2} \end{bmatrix}_{eq} x + [\alpha_e]_{eq} u_{ve} = C_{vflow}x + D_{flow}u_{ve}$$

A second output of interest is the mixer pressure P_v . Taking this pressure to be a function of the mixer internal states, that is,

$$P_v = f_P(z_1(t), z_2(t))$$

then, the linearization of this function around the chosen equilibrium gives

$$P_{vlin} = \begin{bmatrix} \frac{\partial P_v}{\partial z_1} & \frac{\partial P_v}{\partial z_2} \end{bmatrix}_{eq} x = C_{pv}x$$

A third output of interest is the exit temperature T_e . Taking this temperature to be a function of the mixer internal states, that is,

$$T_e = f_T(z_1(t), z_2(t))$$

then, the linearization of this function around the chosen equilibrium gives

$$T_{elin} = \begin{bmatrix} \frac{\partial T_e}{\partial z_1} & \frac{\partial T_e}{\partial z_2} \end{bmatrix}_{eq} x = C_{te}x$$

The indicated partial derivative terms are calculated numerically using Matlab functions within the SIMULINK model.

It is not reasonable to assume that the mixer states be measurable. However, the *state postulate*¹ implies that the state $z(t) = \bar{z} + x$ is directly available for control design if at least two independent thermodynamic properties were measured. Therefore,

- assume that real-time measurements of temperature T_e and pressure P_e are available at the exit point. Then, using the thermodynamic table look-up, one finds the exit flow enthalpy h_e .

¹For a simple compressible substance, any thermodynamic property is at most a function of two other independent thermodynamic properties.

- Assuming a lossless valve, then enthalpy across the valve is preserved so that $h_v = h_e$, where h_v is the mixer (volume) enthalpy.
- Using h_v and T_v (measured) and the table look-up, one finds the mixer pressure P_v .
- From h_v and P_v and the table look-up, one finds the mixer states (energy, density). Or, using the measured exit flow w_e , the known exit pressure P_e required by the specific test article, the computed P_v , and the exit valve coefficient, one solves Equation (3) for the density ρ_v .

In summary then, it is reasonable to assume that

$$y = x = z - \bar{z} \quad (8)$$

is available or that it can be deduced from other real-time measurements. This may be called a “static state observer”. An alternative is to design a *dynamic observer* to provide real-time, optimal estimates of the states. This is beyond the scope of this article.

The calculation of the equilibrium points and the elements of the matrices A , B , C , and D are not included due to space limitations. The relevant results are compiled in Appendix A.

The evaluation of $\frac{\partial P_v}{\partial z}$ is accomplished numerically via a Matlab routine. The evaluation of $\frac{\partial h_v}{\partial z}$ on the other hand, is done by differentiating the expression

$$h = E + C_{units} \frac{P}{\rho}$$

where $C_{units} = 1$ in the *SI* units or $C_{units} = 0.1852$ when h , E are in *BTU/lbm*, P in *psi*, and ρ in *lbm/ft³*. This results in

$$\begin{aligned} \frac{\partial h_v}{\partial z_1} &= 1 + \frac{C_{units}}{z_2} \frac{\partial P_v}{\partial z_1} \\ \frac{\partial h_v}{\partial z_2} &= C_{units} \left[\frac{1}{z_2} \frac{\partial P_v}{\partial z_2} - \frac{P_v}{z_2^2} \right] \end{aligned}$$

It was verified that the relations between pressure P and internal energy or density are essentially linear in the vicinity of an equilibrium point. The Matlab functions extract the slope of a line constructed in the vicinity of a desired equilibrium or linearization point.

2.2 A Numerical Example

It is to be noted that the thermodynamic tables used by the Matlab routines were generated with the program *MIPROPS* and therefore contain an inherent error because of the resolution used. For example, the step size in the pressure intervals is 100 *psi*. However, linear interpolation is used to calculate values not found in the tables which reduces the error. Table 1 lists the data for a typical equilibrium point EQ_i corresponding to a given set of outflow requirements.

The matrices A and B are evaluated at the equilibrium EQ_i listed in Table 1 to be

$$A = \begin{bmatrix} -17.52 & -3393.08 \\ -0.1597 & -43.45 \end{bmatrix}; \quad B = \begin{bmatrix} 25.11 & 332.74 & -42.01 \\ 0.651 & 0.6612 & -0.4919 \end{bmatrix}$$

For an output $y = w_e$, the linear approximation is

$$w_{elin} = C_{vflow}x + D_{flow}u_{ve} = [0.2386 \quad 66.84]x + [1.23]u_{ve}$$

In the absence of real data, we shall refer to the nonlinear SIMULINK simulation as the *real system*. Several simulations were performed to evaluate the linear model:

- **Disturbance Effect - No Control.** We ran the model under a 20 percent pulse disturbance in the gas and liquid valves that take the state away from the initial equilibrium point. When the disturbance disappears, the mixer returns to the equilibrium. It was observed that the linear model captures the important dynamic characteristics of the system very accurately.
- **Tracking and Disturbance Rejection - Open Loop Control.** The behavior of the mixer model was evaluated as the valves are steered in an *open-loop* fashion from the initial to the final pre-calculated positions. These were generated to simulate the manner in which an operator might open/close the valves in response to a visual monitoring of pressure, temperature, or flow sensors. The simulations included a 20 percent pulse perturbations on the liquid and gas valve positions.

3 Point-to-Point Tracking Control Design

The mixer-valve system is a multi-input, multi-output system with an associated transfer function for each input-output pair. It is in general difficult to design PID-style controllers for each loop because of the interactions between channels. The approach chosen in this work is based on the state-space model (7) rather than on the individual transfer functions.

3.1 Single Output Tracking

Consider the problem of exit flow tracking and define a tracking error

$$e(t) = w_{er} - w_e(t) \tag{9}$$

where the reference or desired *constant* exit flow is w_{er} . We seek to design a controller that steers the error signal to zero, that is, $e(t) \rightarrow 0$ as $t \rightarrow \infty$. To that end, define an augmented state vector

$$X = \begin{bmatrix} \dot{x} \\ e \end{bmatrix} \tag{10}$$

that satisfies the differential equation

$$\dot{X} = \begin{bmatrix} A & 0 \\ -C_{flow} & 0 \end{bmatrix} X + \begin{bmatrix} B \\ 0 \ 0 \ -D_{flow} \end{bmatrix} \dot{u} = \bar{A}X + \bar{B}\dot{u} \quad (11)$$

Given the system (11), a performance criterion of the form

$$J = \int_0^\infty (X^T(t)QX(t) + \dot{u}^T(t)R\dot{u}(t)) dt$$

is minimized where the matrix $Q = Q^T \geq 0$ and the matrix $R = R^T > 0$. The needed controllability conditions for the pair (\bar{A}, \bar{B}) are by now standard and may be found in [6]. The resulting optimal controller is given by the linear full-state feedback law

$$\dot{u}(t) = -KX(t) \quad (12)$$

where the matrix K is *constant* and is found *a-priori* by solving the steady-state Riccati Equation

$$\bar{A}^T P + P\bar{A} - P\bar{B}R^{-1}\bar{B}^T P + Q = 0 \quad \implies K = R^{-1}\bar{B}^T P$$

The closed-loop system

$$\dot{X} = [\bar{A} - \bar{B}K]X$$

is guaranteed to be asymptotically stable, that is, $X(t) \rightarrow 0$, $t \rightarrow \infty$. Note that driving X to zero is equivalent to driving the error e to zero as required, and the mixer state x to a constant. Finally, integrating both sides of (12) leads to the *optimal* multi-input/single-output proportional-integral (MISO-PI) controller expression

$$u(t) = -K_P x(t) - K_I \int_0^t [w_{er} - w_e(\tau)] d\tau \quad (13)$$

For example, select the following weighting matrices:

$$Q = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 20 \end{bmatrix}; \quad R = \begin{bmatrix} 0.1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 0.1 \end{bmatrix}$$

so that it is equally important to steer x_1 and x_2 to zero, but it is 20 times more important to drive the exit flow error to zero. Similarly, the liquid and exit valves have 10 times more authority than the gas valve in achieving the stated goal. The optimal gain matrix is found to be

$$K = \begin{bmatrix} 0.4679 & 22.5556 & -12.1274 \\ 0.8992 & -9.3271 & -0.2184 \\ -0.8653 & 13.2205 & -7.2421 \end{bmatrix}$$

from which

$$K_P = \begin{bmatrix} 0.4679 & 22.5556 \\ 0.8992 & -9.3271 \\ -0.8653 & 13.2205 \end{bmatrix}; \quad K_I = \begin{bmatrix} -12.1274 \\ -0.2184 \\ -7.2421 \end{bmatrix}$$

The final equilibrium point was selected to be that corresponding to a 20 percent increase in exit flow, from $w_e = 37 \text{ lbm/s}$ to $w_e = 44.4 \text{ lbm/s}$. The resulting valve, mixer states, flows, and linear model performance was evaluated by simulations. There are (small) step changes that appear in the temperature and pressure plots caused by the resolution of the thermodynamic table and not by the controller. The PI controller successfully takes the exit flow to its final required value.

Simulation results under the same PI controller and with the added perturbations on the liquid and gas valves show that the controller recovers from these perturbations. This behavior could not be achieved under open-loop schemes. However, the mixer states, and the valves do not go to their unique, pre-calculated final values as would be given by a steady-state equilibrium calculation. This is because the controller is designed to track the exit flow step reference only. This is corrected in the next section.

3.2 Multi Output Tracking

Next, we design an optimal multi-input/multi-output PI (MIMO-PI) controller that simultaneously regulates the exit flow, the mixer pressure, and the exit temperature to arbitrary constant set points w_{er} , P_{vr} , T_{er} , respectively. The new augmented vector is

$$X = \begin{bmatrix} \dot{x} \\ e \end{bmatrix} = \begin{bmatrix} \dot{x} \\ w_{er} - w_e(t) \\ P_{vr} - P_v(t) \\ T_{er} - T_e(t) \end{bmatrix}$$

that satisfies the differential equation

$$\dot{X} = \begin{bmatrix} A & 0 & 0 & 0 \\ -C_{flow} & 0 & 0 & 0 \\ -C_{pv} & 0 & 0 & 0 \\ -C_{te} & 0 & 0 & 0 \end{bmatrix} X + \begin{bmatrix} B \\ 0 & 0 & -D_{flow} \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} \dot{u} = \bar{A}X + \bar{B}\dot{u} \quad (14)$$

Minimizing the quadratic cost introduced in the previous section leads to the linear *MIMO-PI* controller

$$u(t) = -K_P x(t) - K_I \int_0^t \begin{bmatrix} w_{er} - w_e(\tau) \\ P_{vr} - P_v(\tau) \\ T_{er} - T_e(\tau) \end{bmatrix} d\tau \quad (15)$$

As a simulation example, select the following weighting matrices:

$$Q = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 20 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 200 \end{bmatrix}; \quad R = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

so that it is equally important to steer x_1 , x_2 , and the mixer pressure error to zero, it is 20 times more important to drive the exit flow error to zero, and it is 200 times more important to drive the exit temperature error to zero. Several simulations indicated that the temperature tracking is the slowest of the three and hence the more stringent requirement. This time, all valves have equal authority in achieving the stated goals. The resulting optimal gain matrix is found to be

$$K = \begin{bmatrix} -0.0719 & 46.93 & -1.8728 & -0.8897 & 2.5733 \\ 0.9454 & -9.8472 & -0.384 & -0.1607 & -13.91 \\ -0.0439 & -11.455 & -4.0429 & 0.4274 & 0.1296 \end{bmatrix}$$

Figure 2 is the SIMULINK diagram used in the simulations. The details of the mixer are not shown. The set point requirements are a 20 percent increase in exit flow from 37 to 44.4 *lbm/s*; a 100 *psi* increase in mixer pressure from 6000 *psi*; and a 2 °F increase in exit temperature from -240 °F. These are introduced as step commands at time $t = 0.3 \text{ sec}$. The valve disturbances are set on and a rate limiter with a slew rate at 50 is put in series with each valve to include another level of realism in the simulations. The simulation plots are in Figure 3 through 6. The effect of the rate limiter is an increase in overshoots and reduced overall performance quality while precluding unrealistically fast valve commands rises especially at the start of the command. The outputs are controlled as required. It is expected that real valves will not only exhibit rate limiting but other linear and nonlinear characteristics.

4 Conclusions

An analytical small-signal model of the high-pressure H_2 mixer and control valves has been derived. The model is suitable for a variety of analysis and control designs valid in the vicinity of an operating or equilibrium point. A classical approach to control design would involve the consideration of a total of nine single-input, single-output transfer functions. Unless the control loops are weakly coupled, such a design is difficult to complete because of the interaction among the control channels. Instead, a state-variable control design has been pursued that results in an optimal multi-input/multi-output proportional/integral law that can be implemented in real-time. It is straightforward to design the controller and it is possible to assign different authorities to each control valve. Precise and simultaneous tracking of constant setpoint changes in exit temperature, exit flow, and mixer pressure were achieved in the presence of constant and unmeasurable additive valve position perturbations. Current efforts include the validation of the model with test data.

	\bar{P} (<i>psi</i>)	\bar{T} (<i>F</i>)	$\bar{\rho}$ (<i>lbm/ft</i> ³)	\bar{h} (<i>BTU/lbm</i>)	C_{vopen}	% <i>OPEN</i>	\bar{w} <i>lbm/s</i>	\bar{z}_1	\bar{z}_2
<i>GH</i> ₂ IN	13500	90	2.91	2113.6	230	<u>0.96</u>	-	-	-
<i>GH</i> ₂ OUT	6804	130.4	1.702	2113.6	-	-	<u>3.72</u>	-	-
<i>LH</i> ₂ IN	8500	-340	5.042	329	115	<u>17.74</u>	-	-	-
<i>LH</i> ₂ OUT	6804	-326	4.519	329	-	-	<u>33.21</u>	-	-
Mixer	6804	-273.32	3.84	508	-	-	-	181.39	3.84
Outflow	5533	-266.21	3.404	508	270	<u>11.12</u>	36.93	-	-

Table 1: Initial Equilibrium Data. Underlined Values are Computed. (IN=Into Valve; OUT=Out of Valve). z_1 in *BTU/lbm*; z_2 in *lbm/ft*³.

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Appendix A

Given a set of required exit mass flow, pressure, and temperature \bar{w}_e , \bar{P}_e , \bar{T}_e , then in steady-state [4]

$$\bar{C}_{vg} = \frac{\bar{h}_v - \bar{h}_l}{\bar{h}_g - \bar{h}_l} \frac{1}{\bar{\alpha}_g} \bar{w}_e \implies \bar{w}_g = \frac{\bar{h}_v - \bar{h}_l}{\bar{h}_g - \bar{h}_l} \bar{w}_e \quad (16)$$

$$\bar{C}_{vl} = \frac{\bar{h}_g - \bar{h}_v}{\bar{h}_g - \bar{h}_l} \frac{1}{\bar{\alpha}_l} \bar{w}_e \implies \bar{w}_l = \frac{\bar{h}_g - \bar{h}_v}{\bar{h}_g - \bar{h}_l} \bar{w}_e \quad (17)$$

To determine the linear perturbation model (7), it is necessary to find the analytic expressions of the indicated partial derivatives. For simplicity we include only those terms that do not evaluate to zero at the equilibrium point. Omitting the details, these are found to be:

$$\left[\frac{\partial F_1}{\partial z_1} \right]_{eq} = \frac{1}{V \bar{z}_2} \left[(h_g - z_1) C_{vg} \frac{\partial \alpha_g}{\partial z_1} + (h_l - z_1) C_{vl} \frac{\partial \alpha_l}{\partial z_1} - (h_v - z_1) C_{ve} \frac{\partial \alpha_e}{\partial z_1} - w_e \frac{\partial h_v}{\partial z_1} \right]_{eq}$$

$$\left[\frac{\partial F_1}{\partial z_2} \right]_{eq} = \frac{1}{V \bar{z}_2} \left[(h_g - z_1) C_{vg} \frac{\partial \alpha_g}{\partial z_2} + (h_l - z_1) C_{vl} \frac{\partial \alpha_l}{\partial z_2} - (h_v - z_1) C_{ve} \frac{\partial \alpha_e}{\partial z_2} - w_e \frac{\partial h_v}{\partial z_2} \right]_{eq}$$

$$\left[\frac{\partial F_2}{\partial z_1} \right]_{eq} = \frac{1}{V} \left[C_{vg} \frac{\partial \alpha_g}{\partial z_1} + C_{vl} \frac{\partial \alpha_l}{\partial z_1} - C_{ve} \frac{\partial \alpha_e}{\partial z_1} \right]_{eq}$$

$$\left[\frac{\partial F_2}{\partial z_2} \right]_{eq} = \frac{1}{V} \left[C_{vg} \frac{\partial \alpha_g}{\partial z_2} + C_{vl} \frac{\partial \alpha_l}{\partial z_2} - C_{ve} \frac{\partial \alpha_e}{\partial z_2} \right]_{eq}$$

$$\left[\frac{\partial F_1}{\partial C_{vg}} \right]_{eq} = \frac{\bar{\alpha}_g}{V \bar{z}_2} (\bar{h}_g - \bar{z}_1) ; \quad \left[\frac{\partial F_1}{\partial C_{vl}} \right]_{eq} = \frac{\bar{\alpha}_l}{V \bar{z}_2} (\bar{h}_l - \bar{z}_1)$$

$$\left[\frac{\partial F_1}{\partial C_{ve}} \right]_{eq} = -\frac{\bar{\alpha}_e}{V \bar{z}_2} (\bar{h}_v - \bar{z}_1) ; \quad \left[\frac{\partial F_2}{\partial C_{vg}} \right]_{eq} = \frac{1}{V} \bar{\alpha}_g$$

$$\left[\frac{\partial F_2}{\partial C_{vl}} \right]_{eq} = \frac{1}{V} \bar{\alpha}_l ; \quad \left[\frac{\partial F_2}{\partial C_{ve}} \right]_{eq} = -\frac{1}{V} \bar{\alpha}_e$$

where

$$\left[\frac{\partial \alpha_g}{\partial z_1} \right]_{eq} = \begin{cases} -2.857 \times 10^{-2} \frac{\sqrt{T_g \rho_g}}{P_g} \frac{P_v}{\sqrt{P_g^2 - P_v^2}} \frac{\partial P_v}{\partial z_1} & P_v < P_g < 2P_v; \\ 0, & P_g \geq 2P_v. \end{cases}$$

$$\left[\frac{\partial \alpha_l}{\partial z_1} \right]_{eq} = \frac{-1}{2} \left[\frac{\rho_l \rho_w}{\alpha_l} \frac{\partial P_v}{\partial z_1} \right]_{eq}$$

$$\left[\frac{\partial \alpha_e}{\partial z_1} \right]_{eq} = \frac{1}{2} \left[\frac{\rho_v \rho_w}{\alpha_e} \frac{\partial P_v}{\partial z_1} \right]_{eq} ;$$

$$\left[\frac{\partial \alpha_g}{\partial z_2} \right]_{eq} = \begin{cases} -2.857 \times 10^{-2} \frac{\sqrt{T_g} \rho_g}{P_g} \frac{P_v}{\sqrt{P_g^2 - P_v^2}} \frac{\partial P_v}{\partial z_2} & P_v < P_g < 2P_v; \\ 0, & P_g \geq 2P_v. \end{cases}$$

$$\left[\frac{\partial \alpha_l}{\partial z_2} \right]_{eq} = \frac{-1}{2} \left[\frac{\rho_l \rho_w}{\alpha_l} \frac{\partial P_v}{\partial z_2} \right]_{eq} ; \quad \left[\frac{\partial \alpha_e}{\partial z_2} \right]_{eq} = \frac{1}{2} \left[\frac{\rho_w}{V \alpha_e} \left(P_v - P_e + z_2 \frac{\partial P_v}{\partial z_2} \right) \right]_{eq}$$

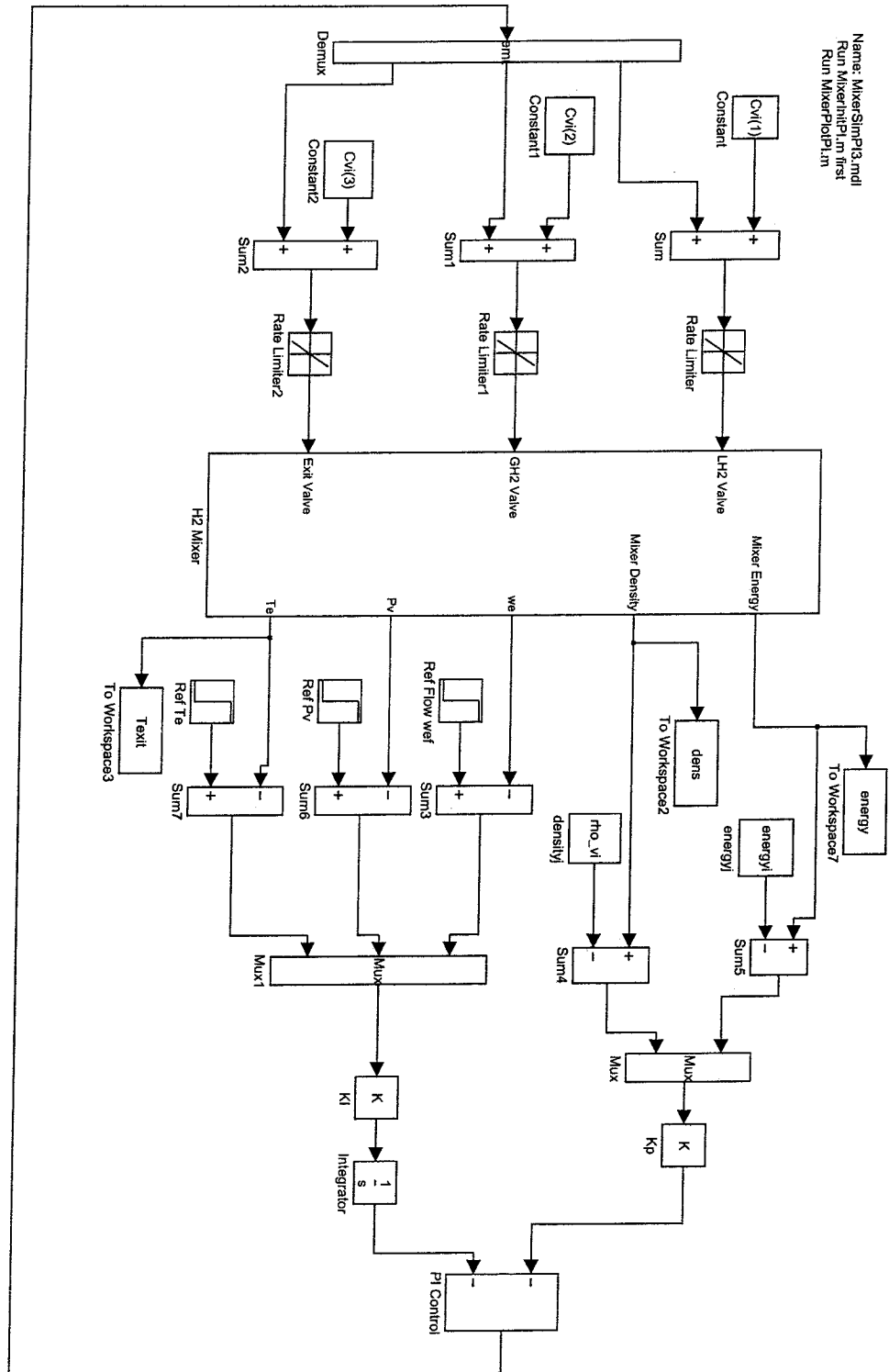


Figure 2: SIMULINK Diagram of the Mixer with MIMO PI Control Loop.

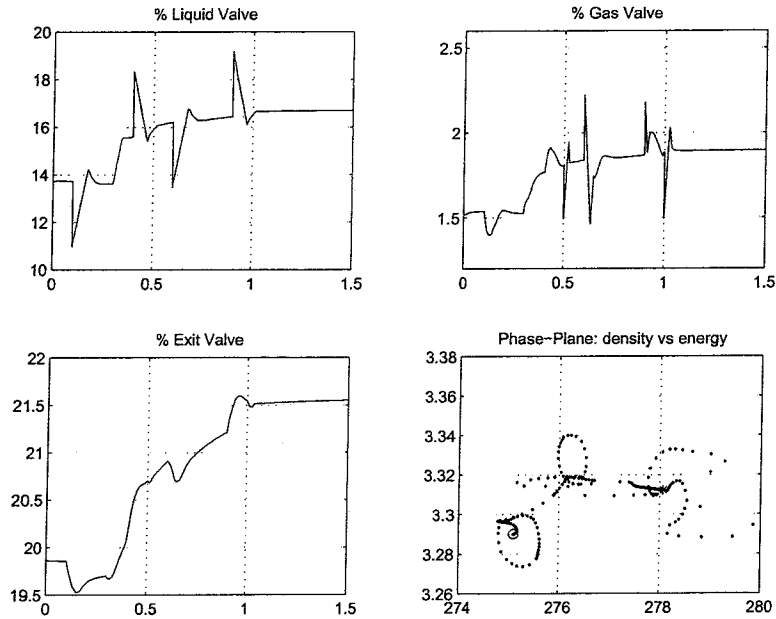


Figure 3: MIMO PI Control vs Time (sec) (with valve perturbations). Bottom right plot is the resulting State Trajectory: Start at \circ ; End at \times .

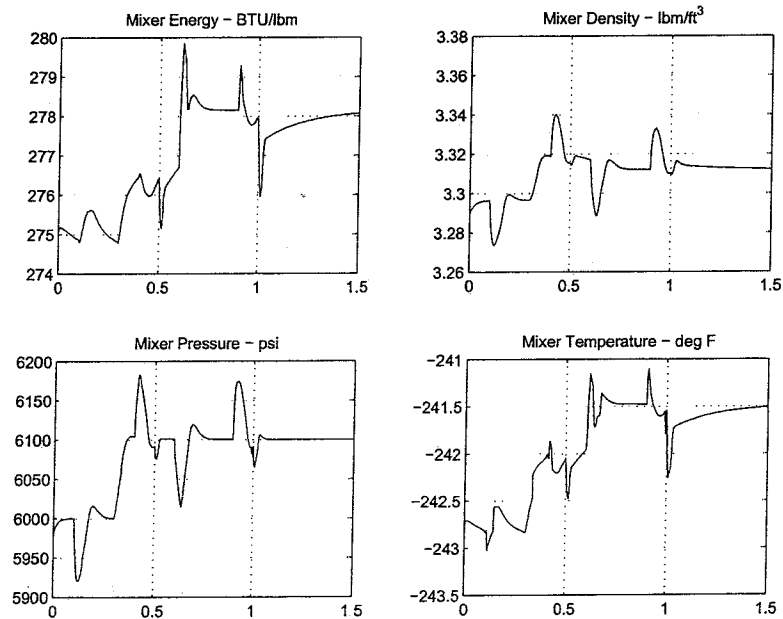


Figure 4: Mixer Behavior vs Time (sec). MIMO PI Control (with valve perturbations)

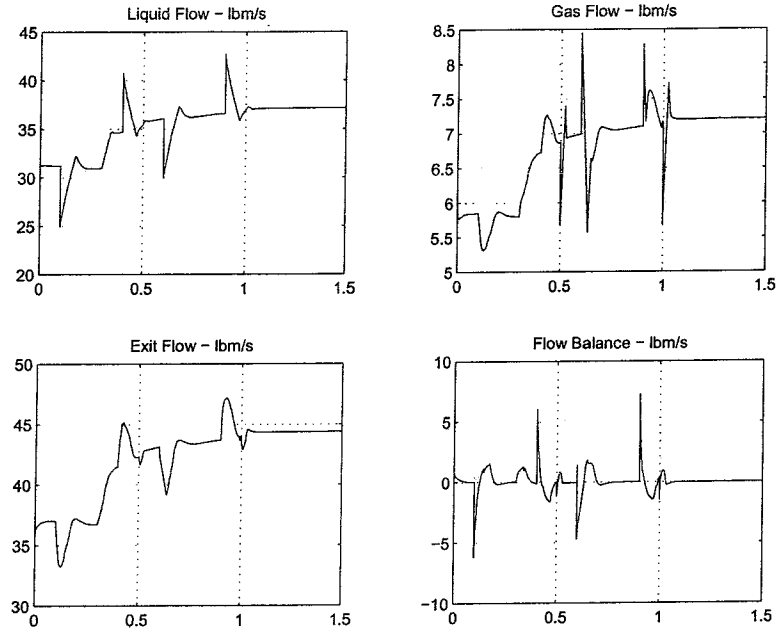


Figure 5: Input and Output Flows with MIMO PI Control vs Time (sec) (with valve perturbations)
Flow Balance is $w_g + w_l - w_e$.

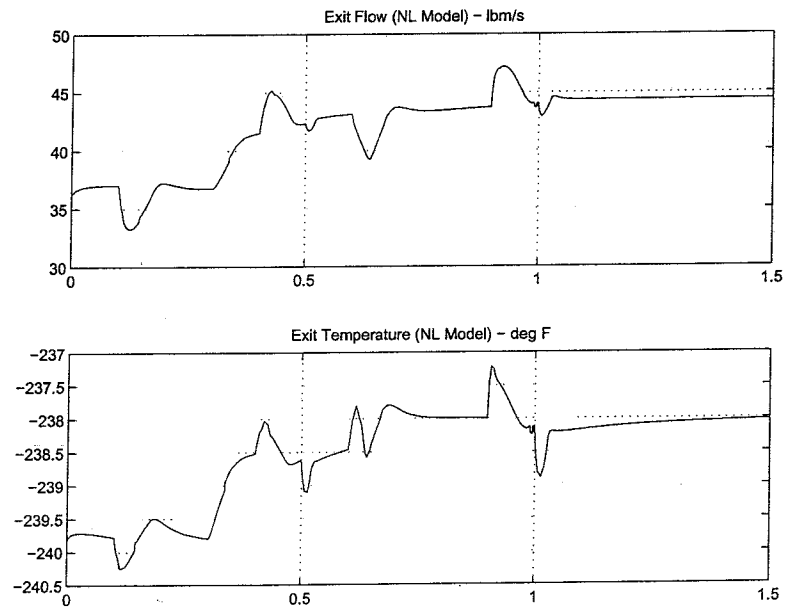


Figure 6: Exit Flow and Temperature Error vs Time (sec). (MIMO PI Control with valve perturbations).

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14. ABSTRACT This paper addresses some theoretical modelling and control issues for a mixing chamber used in rocket engine testing at NASA Stennis Space Center. The mixer is responsible for combining high pressure LH2 and GH2 to produce a hydrogen flow that meets certain thermodynamic properties before it is fed into a test article. The desired properties are maintained by precise control of the LH2 and GH2 flows. The mixer is modelled as a general multi-flow lumped volume for single constituent fluids using density and internal energy as states. The set of nonlinear differential equations is modeled in the SIMULINK environment including a table look-up feature of the fluid thermodynamic properties. A small signal (linear) model is developed based on the nonlinear model and simulated as well. Pulse disturbances are introduced to the valve positions and the quality of the linear model is ascertained by comparing its behavior against the nonlinear model simulations. Valve control strategies that simulate an operator-in-the-loop scenario are then explored demonstrating the need for automatic feedback control. Finally classical optimal single-output and multi-output Proportional/Integral controllers are designed based on the linear model and applied to the nonlinear model with excellent results to						
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