AIR-FUEL RATIO CONTROL OF A LEAN BURN SI ENGINE USING FUZZY SELF TUNING METHOD

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Abstract Reducing the exhaust emissions of an spark ignition engine by means of engine modifications requires consideration of the effects of these modifications on the variations of crankshaft torque and the engine roughness respectively. Only if the roughness does not exceed a certain level the vehicle do not begin to surge. This paper presents a method for controlling the air-fuel ratio for a lean burn engine. Fuzzy rules and reasoning are utilized on-line to determine the control parameters. The main advantages of this method are simple structure and robust performance in a wide range of operating conditions. A non-linear model of an SI engine with the engine torque irregularity simulation is used in this study.

Key Words Fuzzy Control, Self Tuning, Engine Roughness, Air Fuel Ratio

INTRODUCTION

The most important objective of a fuel control system is to provide accurate control of air to fuel (A/F) ratio so that desired drivability and emission levels can be achieved. For most automobiles, this translates to very tight control near the stoichiometric ratio to maximize the three-way catalyst efficiency. Furthermore, if either best economy or maximum power is a priority requirement for an engine, then the air-fuel ratio must be respectively quite lean and just rich of stoichiometry. If in the interests of maximum power the air-fuel ratio is quite low and there is an excess of fuel, the quality of combustion will be poor and the emissions of both unburnt hydrocarbones and carbon monoxide will be high. These can be reduced by moving the air-fuel ratio to a region that is just rich of stoichiometry, which in fact better enables maximum power to be attained, but also incurs a penalty of increased emissions of nitrogen oxides. Should the primary aim be best economy and the air-fuel ratio move across the dividing line to a region just lean of stoichiometry, then combustion temperature will be raised to a maximum and will produce the highest emissions of nitrogen oxides. These can be much reduced and the emissions of carbon monoxide kept to a minimum by further weakening the mixture until it comes into the 'lean burn' region, which has been defined as one where the air-fuel ratios lie between 18:1

and 21:1 but the gains mentioned are bought at the expense of increased emissions of hydrocarbons and the possibility of rough running.

In sum, it will be evident that, the exhaust emissions from an engine are mainly influenced by the region of air-fuel ratio in which it is operating.

Precise control of the internal combustion engine air-fuel ratio is important during all phases of engine operation. During normal hot engine operation the exhaust gas oxygen sensor is used within a closed-loop air-fuel ratio feedback system to maintain the air-fuel ratio very close to the stoichiometric value. An average air-fuel ratio accuracy of about ± 0.02 air-fuel ratio units is required[2]. This accuracy is required to ensure that the catalyst of the after treatment system operates at high conversion efficiency for all three pollutants, i.e. HC, CO, and NOx. This occurs only within a very narrow air-fuel ratio window. The schematic representation of a closed-loop fuel-injection system is given in Figure 1[3].

There are several methods for controlling the air-fuel ratio. Asik and Meyer [2] estimate the air-fuel ratio based on inducing and detecting crankshaft speed fluctuations caused by modulating the engine's fuel injection pulse widths in a predetermined manner. For closed loop control, a conventional PID controller was implemented. The error is calculated as the difference between the desired A/F ratio and the estimated A/F ratio.

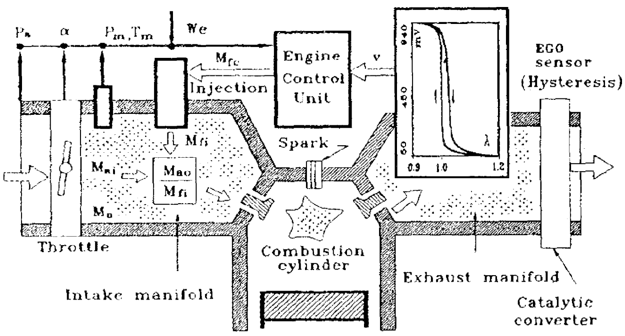

Figure 1. Schematic representation of closed-loop fuel injection system.

Nam, Kim and Yoo [3] have proposed a fuzzy sliding mode control method for designing a fuel injection controller to maintain the stoichiometry air to fuel ratio.This method is used because of it's capability to consider the incomplete information of the current oxygen sensor and its applicability to non-linear engine models. The simulation results of this algorithm show good performance regardless of sensor hysteresis and model errors.

Cho and Keun Oh [4] have designed a fuel-injection controller based on the theory of variable structure systems. This method is formulated to be compatible with production sensors and actuators, including the switching type feedback sensor.

Inagaki, Ohata and Inove [6] developed an adaptive fuel injection control system for accurate A/F ratio control. This system consists of an estimator of fuel behavior model parameters and a modified internal model control system with an inverse system.

Also Cho and Hedrick [7] have designed a nonlinear sliding mode injection controller based on a physically motivated, mathematical engine model. The designed controller can achieve a commanded A/F ratio with excellent transient properties, which offers the potential for improving fuel economy, torque transients. and emission levels.

This paper presents a method for controlling the air-fuel ratio for a lean burn SI engine. The control parameters are obtained on-line by Fuzzy self tuning method. The main advantages of this control strategy are simple structure and robust performance in a wide range of operating conditions. A non-linear model of an

SI engine with the large non-linearities of the intake manifold state equation is used in this simulation.

LEAN BURN ENGINE

Figure 2 Shows in a qualitative way the interconnection between legislatively limited exhaust constituents. specific fuel consumption, torque fluctuation. and excess air-fuel ratio of the air-fuel mixture. At l = 1.4 to l = 1.6 the NOx emission values are as low as those achieved with a three way catalyst. As the diagram shows, the specific fuel consumption at lean mixture is lower than l = 1. This fuel consumption improvement is the most significant advantage of the lean burn concept versus any other concept based on the three-way catalyst.

Other problems connected with the lean burn concept are shown in the graph. One of them is the significant increase in torque unsteadiness which results in rough running. Another is connected with HC emission, which is higher for lean burn engines than tolerated by most legislative limits, even when the mixture is leaned out. In addition, HC emission increases rapidly as the limit for smooth running is approached. As can be seen, a lean burn engine needs, aside from measures for extending the range for smooth running, measures to reduce HC emissions [5].

A lean burn concept does not necessarily mean that the engine must be operated at setting of l>1 across the entire operating range. Mixture metering is achieved as follows:

É Idle: preferably l>1

É Part throttle: lean control to l>>1

É Full throttle: l <1 or l = 1 to achieve optimum engine output.

Tuning of production engines shows that idle and part throttle setting may be adjusted to

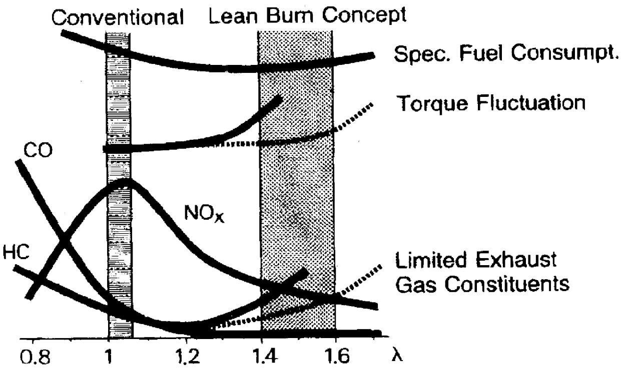

Figure 2. Basic trends of exhaust emissions, specific fuel consumption and torque fluctuation as a function of excess air coefficient(l).

givea lean mixture for optimum fuel economy. During acceleration the engine is supplied with a stoichiometric or rich A/F mixture. When designing a lean burn engine two basic criteria have to be taken into account:

1. Operate engine in a specified l range in the lean burn range to keep NOx emissions low.
2. Avoid exceeding a certain level of rough running or cyclic variation so as to ensure correct drivability and to avoid the increase of HC emissions.

If the above criteria are taken into account, a l range results must be adhered to by implementing suitable controls. Figure 3 shows a schematic diagram of this l range and resulting NOx reduction.

As pointed out above, different strategies must be used for different map ranges to

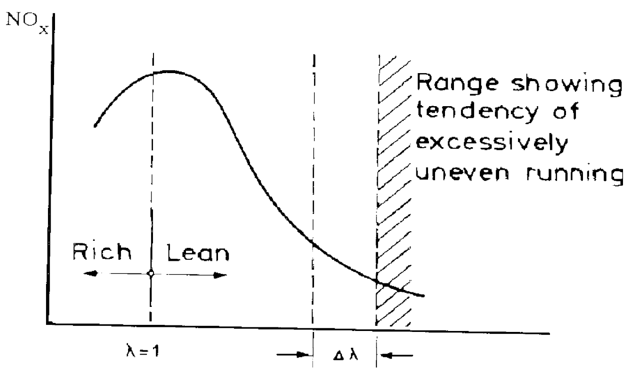

Figure 3. schematic diagram of Dl range and resulting NOx reduction.

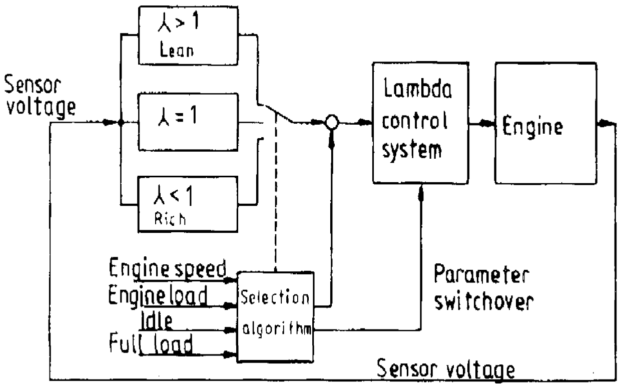

Figure 4. Block diagram of a l range formation.

achieve optimum tuning of an engine. Figure 4 shows one way of generating a suitable l range [4].

ENGINE ROUGHNESS

Engine roughness is a measure of the irregularity of the angular velocity of the crankshaft which is caused by the variation in energy release from cycle to cycle as well as cylinder to cylinder. The variation of mean effective pressure causes torque changes and results in angular speed changes of crankshaft. The engine roughness corresponds to changes of the mean angular acceleration between successive crankshaft rotations. It is approximately proportional to the change the mean torque or mean effective pressure during one rotation of the crankshaft [8]. Figure 5, shows crankshaft speed for increasing the air-fuel ratio, which increases the engine roughness. We can also see the optimal ignition point for smooth running in Figure 6, [8-11].

ENGINE MODELLING

Engine modelling efforts for control have been underway for less than 30 years [12]. There have been many different formulations of engine models. In recent years, dynamic models for automotive engines have been developed that are accurate enough to be used for non-linear controllers, but simple enough to be

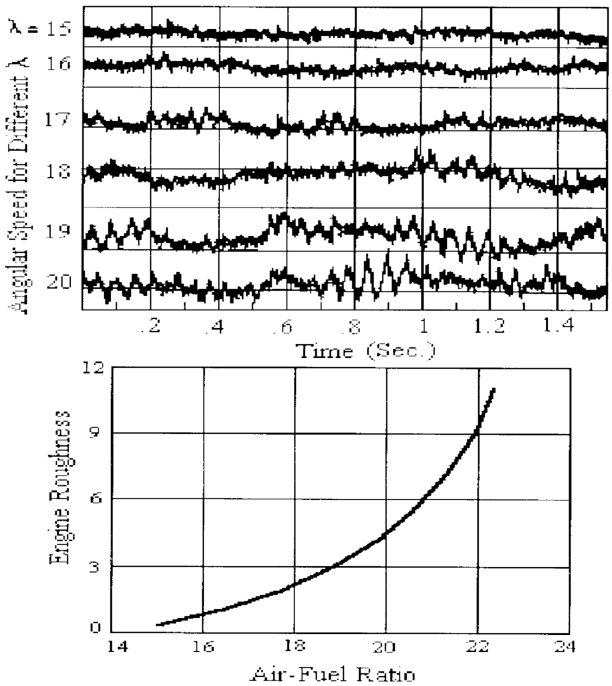

Figure 5. Variation of engine roughness versus air-fuel ratio.

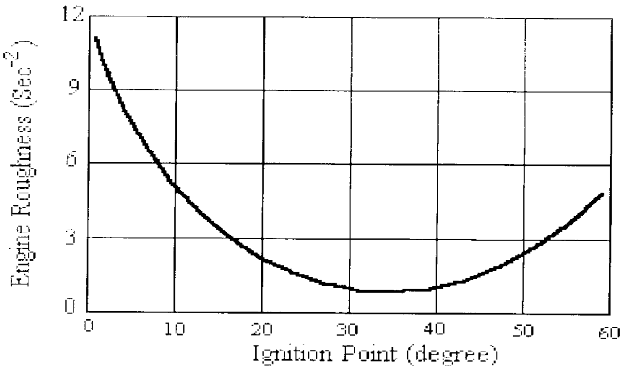

Figure 6. Engine roughness as a function of ignition point.

computed in real time. Some of these models are based entirely on measurements and are often refered to as "input-output" models.

The engine model which is presented here is for various parts of an automobile so any dynamic loads and other transients that affect exhaust emissions and fuel economy can be predicted. The vehicle system is divided into the following dynamic subsystem [13,14,15]:

É Intake manifold

É Engine inertia dynamic

É Vehicle inertia dynamic

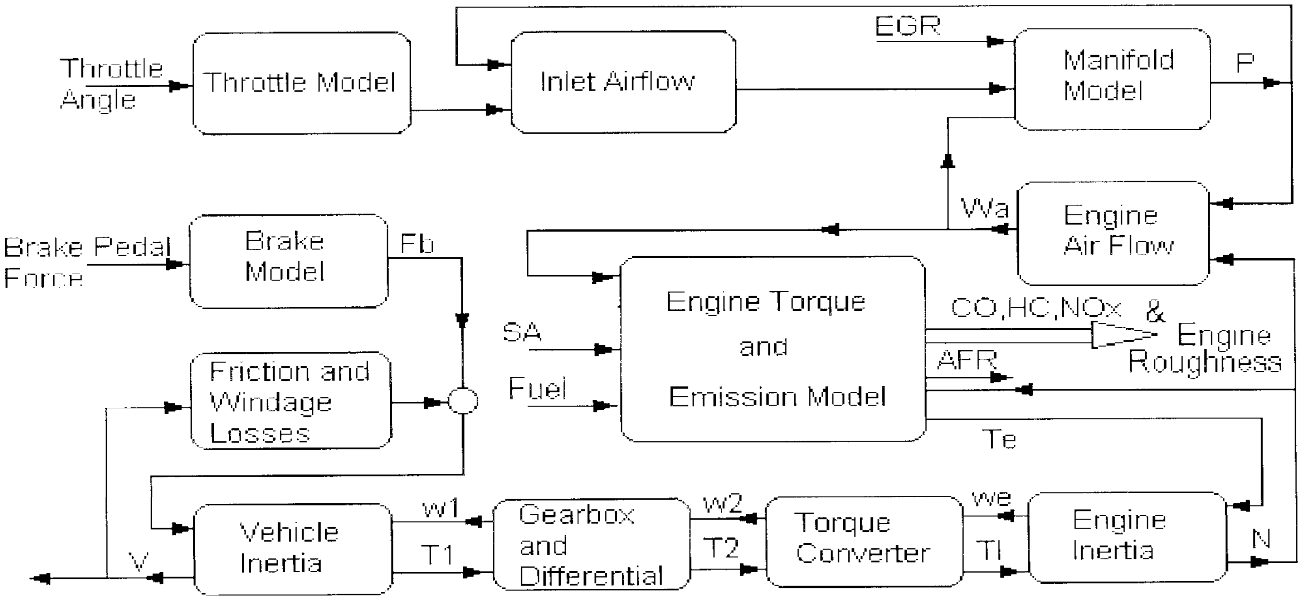

Figure 7. Schematic block diagram for a vehicle system.

Figure 7, represents the entire model in block diagram form. The engine input variables are the throttle angle, the fuel rate, the ignition timing and exhaust gas recirculation (EGR). In this model the state variables are the crank shaft speed and absolute manifold pressure. The engine output variables are the vehicle velocity, exhaust emissions, engine roughness and air-fuel ratio.

Manifold pressure state equation The State equation for manifold pressure is obtained by applying conservation of mass. This can be expressed as [14,15]:

$$\frac{dP_m}{dt} + \frac{(\text{١}-\frac{EGR}{\text{١٠٠}})V\eta_v P_m}{V_m} = \frac{RT_m}{V_m} W_{th} \qquad (1)$$

where,

$$W_{th} = W_{max} \cdot f_1 \cdot f_2 \cdot f_3 \cdot f_4 \qquad (2)$$

$$W_{max} = \frac{C_{dt} \cdot A_{tm} \cdot P}{\text{٣.٨٥}\sqrt{RT}} \qquad (3)$$

$$f_1 = (P_m/P)^{1/k}(3.85)\left[\frac{2k}{k-1}\left[1-\left[\frac{P_m}{P}\right]^{\frac{k}{k-1}}\right]\right]^{1/2}$$

$$\frac{P_m}{P} > .528 \qquad f_1 = 1 \,,\ \frac{P_m}{P} \leq .528 \qquad (4)$$

$$f_2 = \sin\alpha \qquad (5)$$

$$f_3 = \frac{P}{P_0} \qquad (6)$$

$$f_4 = \sqrt{\frac{T_0}{T}} \qquad (7)$$

In constructing the algebraic expression for volumetric efficiency some suggestive results are available in the literature. The function of the volumetric efficiency is [14-17] :

$$\eta_v = \eta_{vb} \cdot K_p \qquad (8)$$

where,

$$\eta_{vb} = 0.75 \qquad Z<1$$

$$\eta_{vb} = .75/Z \qquad Z \geq 1 \qquad (9)$$

The dimensionless parameter Z is identical with "Inlet-Valve Mach Index" adapted by C.F.Taylor, can be written as [15]:

$$Z = (b/d_c)2\,\frac{n_m}{C_i a} \qquad (10)$$

The functions k_p are given by;

$$k_p = \frac{k-\text{١}}{k} + \frac{\varepsilon - \left(\frac{P}{P_m}\right)}{k(\varepsilon-\text{١})} \qquad \frac{P}{P_m} \leq 1.4\,\varepsilon - .4$$

$$Kp = 0 \qquad \frac{P}{P_m} > 1.4\varepsilon - .4 \qquad (11)$$

Vehicle dynamic state equation In operation of the vehicle, the engine develops a torque depending on air flow, air-fuel ratio, spark advance and other conditions. This torque is transmitted to wheels through the torque converter, gear box and differential. When brakes are applied, the throttle is at its idle position and the brakes decelerate the vehicle dissipating the excess available energy in the form of heat. Physically the vehicle inertia equation is derived using Newton's law and can be expressed as:

$$dv / dt = (F_{in} - F_{out}) / M_{eq} \tag{12}$$

where,

$$M_{eq} = M + (I_{ee} + I_e) / R2_w \tag{13}$$

$$I_{ee} = I_e (GR_t) \tag{14}$$

$$F_{in} = T_e \xi_t \ R_t G / R_w \tag{15}$$

$$F_{out} = F_r + C_w v2 + F_b \tag{16}$$

where, T_e is a function of throttle angle, spark ignition timing and other engine parameters such as air-fuel ratio. The torque irregularity from cycle to cycle is expressed by:

$$\Delta \int_j^{j+1} T_e dt = -\delta_n (4\pi^2 I_e) \tag{17}$$

where δ_h , is the engine roughness. In this study we have simulated the engine roughness as a disturbance on engine torque. The characteristics of this signal is in good agreement with experimental results of reference [8].

MODELLING OF EXHAUST EMISSIONS

Engine operating variables and design parameters have significant influence on exhaust emissions. It is often difficult to isolate the effects of a single design variable or an operating parameter. Any variable such as air-fuel ratio, spark timing, speed, load, EGR, valve overlap, intake manifold pressure, compression ratio and valve timing have a significant influence on the exhaust emissions and fuel economy. Many studies have shown that among the mentioned variables only the air-fuel ratio, spark advance and EGR can be controlled. Some research workers have used the steady state data from experimental study and have reported the control strategies which sought to reduce emissions. Some approximate functional relationships between exhaust emissions, operating conditions and control variables can be established. Then these empirical relationships can be directly used for determining exhaust emission rates at any operating conditions. The following equations give the functional forms for the various emission rates. The functions f_1 , f_2 and f_3 are determined by using the empirical correlation techniques [18].

$$\dot{CO} = f_1 (W_{th} , AFR, \delta , EGR, T_e ,n) \tag{18}$$

$$\dot{HC} = f_2 (W_{th} , AFR, \delta , EGR, T_e ,n) \tag{19}$$

$$\dot{NO} = f_3 (W_{th} , AFR, \delta , EGR, T_e ,n) \tag{20}$$

Because steady state tests are used to obtain the experimental data, these equations are not valid for cold start and warm up periods.

SELF TUNING STRATEGY

Fuzzy control is an old control paradigm that has received much attention recently. While a controller is at work, uncertainties such as disturbance, parameter perturbation and unknown system dynamics always exist in the system. Conventional tuning techniques for PID controller usually produce an unsatisfactory control performance.

In modern control theories, after the first research on the fuzzy controller based on the methodology of fuzzy set presented by L.A. Zadeh, many enhancements and application articles about this controller have been presented. These studies show that the fuzzy controller exhibits superior applicability to the traditional PID controller and displays

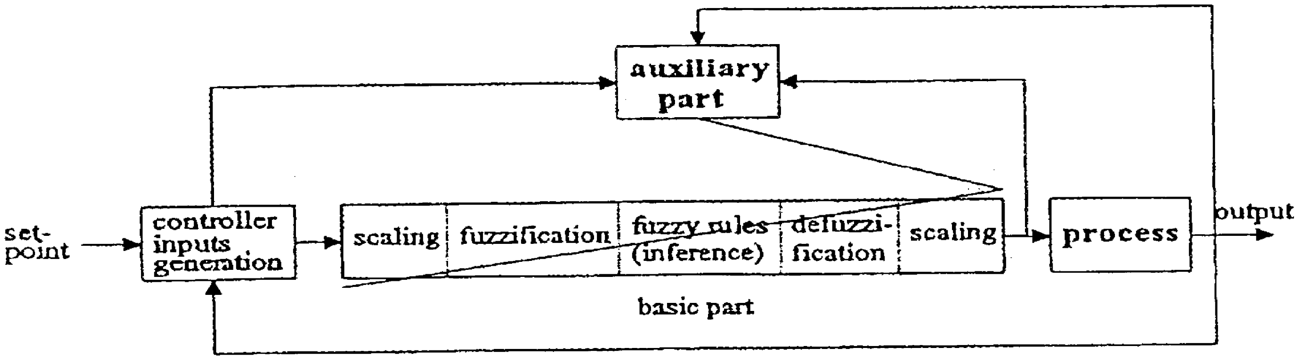

Figure 8. A learning fuzzy controller.

considerable robust effects. A fuzzy controller composed of control rules of conditional linguistic statements on the relationship between input and output variables has the enticing advantages to emulate the behavior of a human and to deal with model uncertainty. Similar to traditional adaptive controllers, adaptive fuzzy controllers can also be categorized into direct and indirect types.

A direct adaptive fuzzy controler can simply be represented by the block diagram as shown in Figure 8, in which the auxiliary part may be a reference model, an auxiliary controller, a monitor, or a parameter adjuster, and the diagonal line across the basic part means the tuning process upon the fuzzy controller.

The reference model is constructed by defining a desired dynamic equation or by using the relationship between change in input and change in output error.

In this study the objective of the fuzzy controller is to track the vehicle velocity and control the A/F ratio in lean operating conditions. So the desired A/F ratio is obtained by the exhaust emissions and engine roughness limited. In this paper we have used the fuzzy tuning method for adjusting the controller parameters. The controller only requires input and output data (does not require the plant's model). This method is based on servo controller tuning with fuzzy logic which is adapted by H.C Tseng and V.H. Hwang [19].

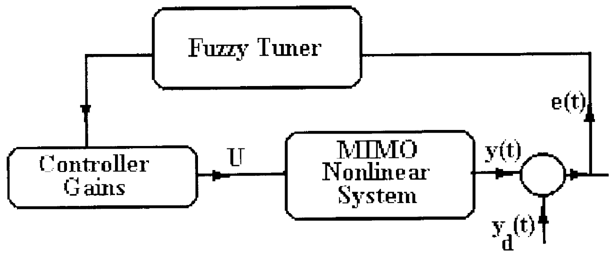

Figure 9. The block diagram of integrated controller.

Figure 9. shows block diagram of the self tuning fuzzy control. u is the plant input vector, y is the output vector, y**d**(t) is the desired output vector and e is the error vector.

The Process model is written by:

$$\dot{x}(t) = f(x(t), u(t), t) \tag{21}$$

$$y(t) = g(\dot{x}(t), x(t), t) \tag{22}$$

$$e(t) = y(t) - y_d(t) \text{ for tracking} \tag{23}$$

$$u(t) = Ke(t) \tag{24}$$

$$\text{where, } x \in R^n,\ u \in R^m,\ y \in R^r,\ e \in R^r. \tag{25}$$

The tracking tuner cases that:

$$e(t) \to 0 \tag{26}$$

FUZZY RULES

The control variable u(t) is proportional to error e(t) and is given by:

$$u(t) = K(t)\ e(t) \tag{27}$$

The objective of the fuzzy tuner is to evaluate the incremental changes of K. In other words, we have:

$$K(t_{k+1}) = K(t_k) + DK(t_{k+1}) \tag{28}$$

Based on the servo controller tuning whose complete technique can be found in [19], we have used the method for fuzzy rules generation based on the Lyapunov function as a performance index. Consider the following Lyapunov function candidate for deriving a stable fuzzy rule set [19,20]:

$$V(e,\dot{e}) = e^T e + \dot{e}^T \dot{e} \qquad (29)$$

The rate of change of V with respect to time, $\dot{V}$, can be approximated by:

$$\dot{V} \approx \frac{\Delta V}{\Delta t} \qquad (30)$$

where, $\Delta V(t_k) = V(t_k) - V(t_{k+1})$. In order to drive $y_{id}(t)$, we require $d|e_i|dt<0$. To satisfy the condition $d|e|/dt<0$ or $\Delta|e|<0$ by tuning K, one must find an expression that relates $\Delta|e|$ to the desired changes in K. This expression can be obtained through the introduction of a sensitivity function:

$$S_{ij} = \frac{\partial |e_i|}{\partial K_{ij}} \qquad (31)$$

which leads to:

$$\Delta |e_i| = \sum_{j=1} S_{ij} \Delta K_{ij} \qquad (32)$$

and for a stable control,

$$\Delta V \approx \Delta(e^T e + \dot{e}^T \dot{e}) \leq 0 \qquad (33)$$

The idea of deriving the fuzzy rules is to have $\Delta|e_i|$ such that it converges at a constant rate $\Delta|e_i| = -\eta_i$, where η_i is a positive defining the convergence rate. To guarantee this convergence condition, on can use the principle of superposition to determine the sign of each ΔK_{ij} ,such that $\Delta|e_i|$ is always negative.

One way to fuzzify the variables, Δe_i , ΔK_{ij} and S_{ij} is to introduce a linguistic term (membership function) for each of them in form of triangular as is shown in Figure 10. The complete fuzzy decision table for the matrix K is as shown in Table 1. In this paper the quadratic

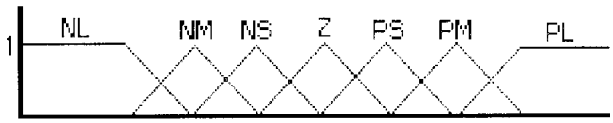

Figure 10. Membership function for the fuzzy term sets.

TABLE 1. Fuzzy Decision Table for Matrix ΔK.

$\Delta\|e\|$ / S_{ij}	NL	NM	NS	Z	PS	PM	PL
NL	PS	PS	PS	Z	PS	PM	PM
NM	PM	PM	PS	PS	PM	PM	PL
NS	PL	PM	PM	PS	PM	PL	PL
Z	Z	Z	Z	Z	Z	Z	Z
PS	NL	NM	NM	NM	NL	NL	NL
PM	NM	NM	NS	NS	NM	NM	NL
PL	NS	NS	NS	Z	NS	NS	NM

performance index is as follow:

$$J = e^T Q e + \dot{e}^T P \dot{e} \qquad (34)$$

where, e is output error vector and Q and P are weighting matrix for selecting the most important outputs.

RESULTS AND DISCUSSIONS

Computer simulations of the vehicle dynamic and SI engine with engine roughness simulation have been accomplished. The large non-linearities of the intake manifold state equation are preserved in this simulation. The results of the MIMO closed loop simulation are shown in Figures 11-14 . In order to check the performance of the algorithms, we have specified a torque disturbance with characteristics similar to experimental results. The results show that the vehicle velocity tracks the desired path excellently as shown in Figure 14 and the air-fuel ratio is approximately constant at a given value (about 20). But in regions where the vehicle is accelerated (time: 5-7.5 Sec.) by opening the Throttle as shown in Figure 13, the air fuel ratio is decreased and the

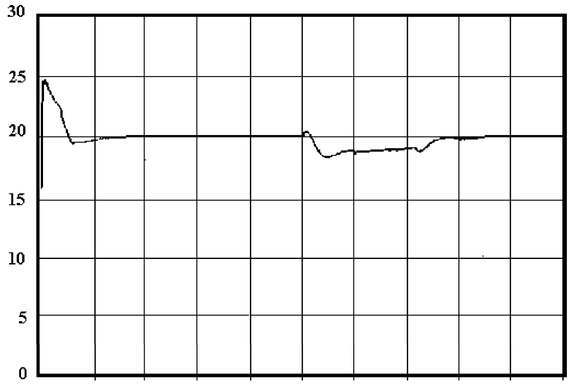

Figure 11. Air-fuel ratio versus time.

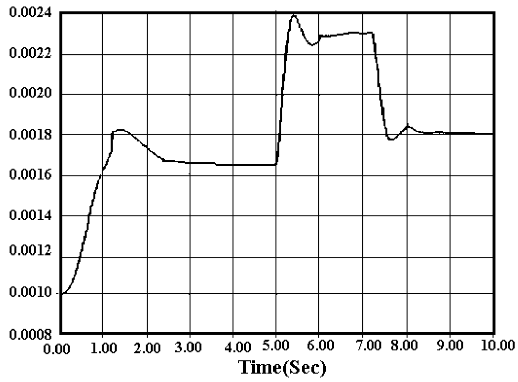

Figure 12. The engine fuel consumption rate.

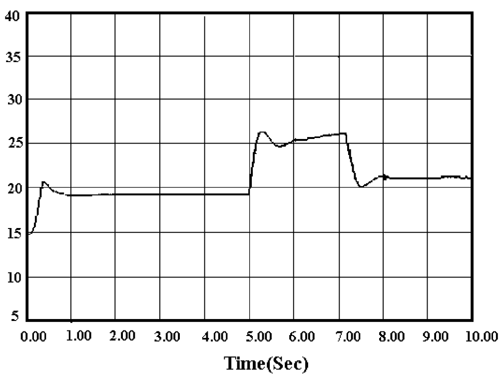

Figure 13. Throttle angle position versus time.

engine operated at rich mixture. Therefore, the requirements of the best economy and maximum power are satisfied.

CONCOLUSION

In this paper, it has been shown that it is possible to find the best engine adjustment compromise regarding emissions and engine roughness. A vehicle model has been used for simulation. The model contains non-linear elements of the engine, specially the engine torque irregularities. The control is done by regulating the multidimensional proportional fuzzy controller gains. These simulation results showed the applicability of self tuning fuzzy control for air-fuel ratio control to a lean burn engine.

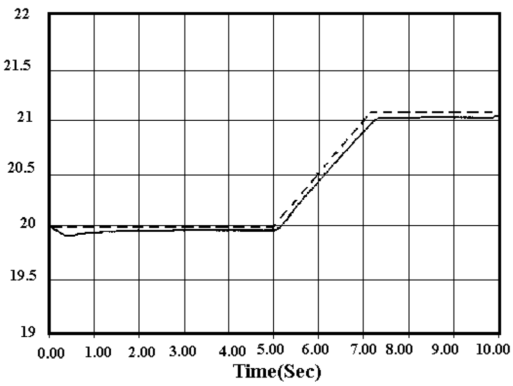

Figure 14. The tracking control of vehicle velocity.

NOMENCLUTURE

A_{tm}	Area of throttle body throat (m2)
AFR	Air to fuel ratio
a	Sound speed (m/sec)
b	Piston diameter (m)
C_{dt}	Flow coefficient of throttle body throat
C_i	Air flow coefficient in inlet valve
C_w	Drag coefficient
$\dot{C}o$	Co exhaust emission rate
d_c	Inlet valve diameter (m)
EGR	Exhaust gas recirculation
F_{in}	Positive force at the wheel (N)
F_{out}	Negative force at the wheel (N)
F_b	Braking force (N)
F_r	Friction force (N)
G	Axle ratio
$H\dot{c}$	Hc exhaust emission rate
I_e	Moment of inertia of rotary parts (kgm2)
I_{ee}	Equivalent moment of inertia (kgm2)
Ivc	Inlet valve closing angle (degree)
k	Ratio of specific heats
M	Vehicle mass (kg)
M_{eq}	Equivalent mass (kg)
n	Engine speed (rpm)

n_m	Average of piston speed (m/sec)
$N\dot{o}$	Nox ehaust emission rate
P	Exhaust pressure (kpa)
P_o	Standard pressure (kpa)
P_m	Intake manifold pressure (kpa)
R	Ideal gas constant (KJ/kg.K)
R_t	Transmission gear ratio
R_w	Wheel radius (m)
T	Exhaust temperature (K)
T_e	Engine output torque (Nm)
T_o	Standard temperature (K)
T_m	Intake manifold temperature (K)
V	Displaced volume of engine (m3)
V_m	Intake manifold volume (m3)
v	Vehicle speed (m/sec)
W_{th}	Throttle mass air flow (kg/sec)
W_{max}	Maximum mass air flow into intake manifold (kg/sec)
Z	inlet mach index
a	Throttle angle (degree)
e	Compression ratio
d	Spark advance (degree)
d_n	Engine roughness ($1/_{Sec}2$)
l	excess air coefficient
z_t	Mechanical efficiency of gear box
h_v	Volumetric efficiency

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