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ABSTRACT

One of the most vital factors in combustion control is Air-to-Fuel Ratio (AFR) control and estimation. Various parameters such as intake air mass flow rate, fuel injection quantity and (Exhaust Gas Recirculation) EGR flow rate are affecting the AFR magnitude. In this study a detailed mathematical, nonlinear and control oriented model of dynamic processes of turbocharged diesel engines is presented for the purpose of AFR control. It is an attempt to describe the two main subsystems of compression ignition engines called intake manifold and fuel injection systems.

The intake model itself consists of compressor, intercooler, intake manifold, EGR circuit, valve section and cylinder subsystems. This model has been developed for study of the air intake system in a turbocharged diesel engine, using physical equations. Experimental data has also been used to adjust the model. The output variable of this submodel (air mass flow) is a significant parameter in calculations of engine power, emission and AFR control. Of the methods of assisted aspiration, the exhaust gas driven turbocharger is by far the most widely used. Varying the rate of gas flow through the turbine by means of Variable Geometry Turbine (VGT) is a method by which the exhaust gas flow through turbine can be limited at high engine speeds. Therefore, turbocompound in this model is considered to have VGT.

Common Rail Injection (CRI) that is a fully flexible fuel injection system in which quantity, timing and pressure of injection are controllable (based on electronic control) separately is chosen in this study. Fuel injection system is comprised of solenoid, working chamber and needle subsystems. A one-dimensional, transient and compressible flow model of CRI, based on Kirchhoff's law, mass and momentum conservation equations and the equilibrium of forces is derived.

AFR control of diesel engine is performed make use of fuzzy logic methodology. Fuzzy logic is a control structure which emulates the way humans arrive at decisions, given a certain set of circumstances. This method improves the control and simplifies the development process. The fuzzy logic controller achieves this by considerably reduced complexity and lack of tedious mathematical derivation associated with modern control theory methods.

All above-mentioned models are programmed in Matlab/Simulink software environment. The simulation results are then compared with available data to check the accuracy of the model.

KEYWORDS: AFR control, Common rail injection, Computer simulation, Fuzzy logic methodology, Intake system, Turbocharger.

INTRODUCTION

The drive to reduce emissions and fuel consumption while meeting improved performance objectives has led to several ameliorative diesel engine technology in recent years. EGR, Variable Geometry Turbocharger (VGT) and electronic fuel injection (EFI) systems are playing a crucial role in achieving these aims. Also today's diesel engines have great flexibility make use of advanced control systems. One important step is the control of the individual AFR control which is a good representation of the power produced by engine. AFR control culminates in high combustion efficiency of the engine, frugality in fuel consumption and meeting the established emission legislations. AFR is calculated from two main inputs: injection quantity and intake air mass flow quantity. For the purpose of AFR control, electronic fuel injection system is considered as the actuator. In other words, by varying the amount of injected fuel, AFR can be controlled and maintained in an acceptable frame-work.

The introduced models for air intake system and fuel injection system are nonlinear models. Linearization of this nonlinear engine model induces a model which is valid only in a small region around engine working point. So the use of nonlinear controllers is determinate. Fuzzy logic methodology is the plied nonlinear control method in this work.

Several design oriented and control oriented diesel engine models have been derived yet. Ouenou-Gamo et al. [1] presented a theoretical model for air intake process of a turbocharged diesel engine. This model describes the dynamics of air into the manifold and the intake phase by physical equations. Kolmanovsky et al. [2] summarized recent developments in turbocharged diesel engine models, equipped with Variable Geometry Turbine (VGT) and Exhaust Gas Recirculation (EGR). Jung [3] has investigated mean value modeling and robust control of the air path of a diesel engine equipped with VGT and EGR. The model is derived with a focus on the parameterization of the turbocharger.

Also a number of common rail injection systems have been developed and published over the last few years. Ahlin et al. [4] described the behavior of pressure in the common rail diesel injection system mathematically. This model was based on well known physical relations. Gullaksen et al. [5] developed a computer program that implemented the numerical simulation of equations for plunger, needle dynamics and transient flow in high pressure pipe-line. Furthermore, Lee et al. [6] represented a detailed mathematical model of all diesel fuel injection system components using principles of fluid dynamics.

Finally, dynamic system simulation software is an important tool for developing reliable engine control systems. All the equations and submodels of this work are simulated in Matlab/Simulink.

INTAKE SYSTEM

One of the major factors in calculating air-to-fuel ratio of a turbocharged diesel engine is air mass flow rate. For precise calculation of this parameter a detailed, analytical and mathematical model of air intake process is required. Figure 1 illustrates the components in a turbocharged diesel engine intake system: compressor, intercooler, EGR circuit and intake manifold.

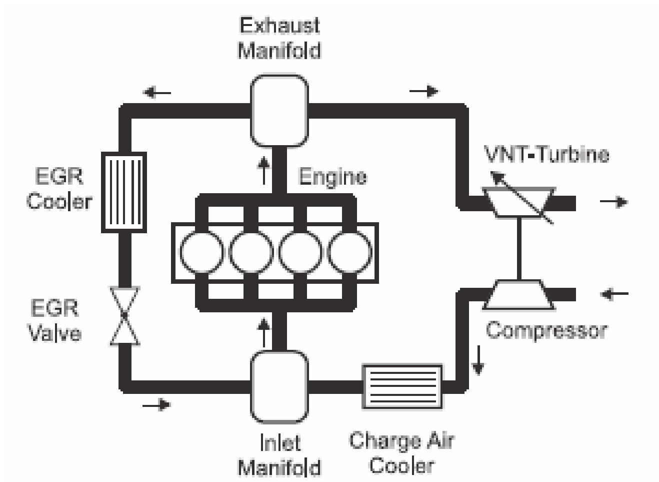

Figure1. Schematic diagram of a turbocharged diesel engine [7]

EGR is a well-established technique for reducing in-cylinder engine NOx production. In this method, a fraction of the exhaust gases is fed into the inlet manifold to mix with the incoming fresh air. The EGR flow is controlled by a valve between intake and exhaust manifolds.

Moreover, a fixed geometry turbocharger will provide a fixed pressure ratio in a given engine speed and load, while a Variable Geometry Turbocharger (VGT) provides a range of compressor pressure ratios in any given engine condition. Furthermore, adjustable nozzle vanes at the entry to the turbine alter the flow area into the turbine.

All the components of air intake system are modeled using physical relations such as ideal gas law, conservation of mass and energy equations and some experimental equations.

COMPRESSOR

Air characteristics after compressor, especially pressure and temperature are fundamental elements of intake mass flow rate. Work is done on compressor from an external source by turbocharger rotating shaft. The differential equation for calculating the downstream compressor pressure is as follows:

$$\frac{dp_c}{dt} = \frac{\eta_c}{\dot{m}_{comp} \cdot R \cdot T_0} \cdot p_0^{\frac{\gamma-1}{\gamma}} \cdot p_c^{\frac{1}{\gamma}} \cdot \dot{W}_c \tag{1}$$

The definition of compressor isentropic efficiency leads to a differential equation for downstream temperature:

$$\frac{dT_c}{dt} = \frac{1}{\dot{m}_{comp} \cdot c_{p_a}} \cdot \dot{W}_c \tag{2}$$

Compressor efficiency is assumed to be constant and is equal to 0.51. The power is transmitted from turbine to compressor by a rotating shaft. The mathematical modeling of the shaft is available in [8]. It should be noted that compressor mass flow rate modeling is done based on assumptions made in [9].

INTERCOOLER

The downstream temperature of intercooler is calculated using the effectiveness of a heat exchanger and an appropriate coolant temperature [3]:

$$T_{c,ic} = \eta_{he} T_{cool} + (1 - \eta_{he}) T_c \tag{3}$$

Pressure drop across the intercooler is neglected.

INTAKE MANIFOLD

Make use of mass and energy balance equations for the intake manifold volume, the following differential equations are derived for pressure and mass of air inside the intake manifold:

$$\frac{dM_{im}}{dt} = \dot{m}_{comp} + \dot{m}_{egr} - \dot{m}_{cyl} \tag{4}$$

$$M_{im} c_{p,air} \frac{dT_{im}}{dt} = -\dot{m}_{comp} c_{p,air} T_c + \dot{m}_{egr} c_{p,egr} T_{eg} + \dot{m}_{cyl} c_{p,mix} T_{im} \tag{5}$$

Temperature of exhaust gases is computed from an experimental equation. Having pressure and mass of air in intake manifold, the pressure is computed by applying the ideal gas law:

$$P_{im} = \frac{R \cdot T_{im} \cdot M_{im}}{V_{im}} \tag{6}$$

VALVE SECTION

The air intake flow into cylinder depends on its density, its velocity through the valve section and cross sectional area of the input port:

$$\dot{m}_{cyl} = \rho_{csa} . u_{csa} . A_{csa} \tag{7}$$

The air density through the valve section is calculated by assuming isentropic condition and considering an adapted neck:

$$\rho_{csa} = \frac{P_{im}}{RT_{im}} \left(\frac{P_{cyl}}{P_{im}} \right)^{\frac{1}{\gamma}} \tag{8}$$

Air velocity is computed by writing the energy conservation equation of the air between the intake manifold and the valve section:

$$u_{csa} = \sqrt{\frac{2\gamma}{\gamma - 1} \left(\left(RT_{im} - \frac{P_{cyl}}{\rho_{csa}} \right) + R(T_{im} - T_0) \right)} \tag{9}$$

For calculating the flow cross sectional area, the following equations are used [10]:

$$A_{csa} = \pi x_p \cos\beta \left(D_v - 2w + \frac{x_p}{2} \sin 2\beta \right) \tag{10}$$

If: $0 < x_p < \frac{w}{\sin\beta \cos\beta}$.

$$A_{csa} = \pi D_m \left[(x_p - w\tan\beta)^2 + w^2 \right]^{\frac{1}{2}} \tag{11}$$

If: $\frac{w}{\sin\beta\cos\beta} < x_p < \left[\left(\frac{D_p^2 - D_s^2}{4D_m} \right) - w^2 \right]^{\frac{1}{2}} + w\tan\beta$

$$A_{csa} = \frac{\pi}{4} \left(D_p^2 - D_s^2 \right) \tag{12}$$

If: $x_p > \left[\left(\frac{D_p^2 - D_s^2}{4D_m} \right) - w^2 \right]^{\frac{1}{2}} + w\tan\beta$

In above equations, β is valve seat angle, D_v is valve head diameter, w is valve seat width, D_p is input port diameter, D_s is valve stem diameter and $D_m = D_v - w$. And according to the camshaft elliptical form, the valve lift expression is given by [1]:

$$x_p = b \left(1 - \cos^2 \varphi_{cs} \left(1 - \frac{b^2}{a^2} \right) \right)^{-0.5} - b \tag{13}$$

Where a and b are the large and the short axis of an ellipse respectively and φ_{cs} is the camshaft angular position.

EGR CIRCUIT

It is assumed that no mass is accumulated in the EGR system. So it can be modeled with static equations rather than with differential equations. The flow through the EGR valve is determined by standard orifice flow equation [10]:

$$\dot{m}_{egr} = \frac{A_{egr} P_{em}}{\sqrt{RT_{em}}} \sqrt{\frac{2\gamma}{\gamma - 1} \left(P_r^{\frac{2}{\gamma}} - P_r^{\frac{\gamma+1}{\gamma}} \right)} \tag{14}$$

With the following definition for pressure ratio:

$$P_r = \max \left(\frac{P_{im}}{P_{em}}, \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma+1}} \right) \tag{15}$$

Effective area of EGR valve is computed from a second order polynomial function according to EGR valve normalized position. EGR valve position is changing from 0 (completely closed) to 1 (completely open). EGR temperature is assumed to be constant. Its value is set to 700 K.

CYLINDER DYNAMICS

The volume in the cylinder during admission phase depends on the piston stroke and the cylinder section. Variations of cylinder volume are expressed by the following differential equation:

$$\frac{dV_{cyl}}{dt} = A_{cyl} r \dot{\varphi} \sin\varphi \left(1 + \frac{\cos\varphi}{\sqrt{\lambda^2 - \sin^2\varphi}} \right) \quad (16)$$

Furthermore, derivative form of the cylinder temperature is given below:

$$\frac{dT_{cyl}}{dt} = \frac{1}{m_{cyl}} \left(\gamma \frac{p_{cyl}}{RP_{csa}} - T_{cyl} \right) \dot{m}_{cyl} + \frac{1}{c_v m_{cyl}} \dot{W}_{ad} \quad (17)$$

And cylinder pressure is calculated by differentiating the ideal gas law:

$$\frac{dP_{cyl}}{dt} = \frac{1}{V_{cyl}} \left(RT_{cyl} \dot{m}_{cyl} + Rm_{cyl} \dot{T}_{cyl} - P_{cyl} \dot{V}_{cyl} \right) \quad (18)$$

EXHAUST MANIFOLD

Variations of exhaust manifold parameters are described by the filling and emptying model as well as by the first law of thermodynamics. Exhaust manifold temperature is coupled to EGR temperature and to exhaust gases temperature:

$$M_{em} c_{p,e,} \frac{dT_{em}}{dt} = (\dot{m}_{fuel} + \dot{m}_{cyl}) c_{p,eg} T_{eg} - \dot{m}_{egr} c_{p,egr} T_{em} - \dot{m}_{em} c_{p,em} T_{em} \quad (19)$$

Air flow dynamics in exhaust manifold can be described as a first order lag:

$$\frac{dM_{em}}{dt} = \dot{m}_{fuel} + \dot{m}_{cyl} - \dot{m}_{egr} - \dot{m}_{em} \quad (20)$$

Again the pressure of exhaust manifold is calculated from the relation of ideal gas law:

$$P_{em} = \frac{R \cdot T_{em} \cdot M_{em}}{V_{em}} \quad (21)$$

TURBINE

For obtaining mass flow circulating through the variable geometry turbine and its outlet pressure experimental equations are used [11]:

$$\dot{m}_t = A \left(1 - e^{-B\left(\frac{P_{em}}{P_t}\right)^C} \right) \quad (22)$$

A, B and C parameters are polynomial functions of guide vane position of turbine. Guide vane position is changing from 0 (completely closed) to 10 (completely open). Pressure of gases after turbine is also computed from an experimental relation:

$$P_t = 10^5 \left(9.10^{-7} \dot{m}_t^{\,2} + 0.0002 \dot{m}_t + 0.9981 \right) \quad (23)$$

The definition of turbine isentropic efficiency yields the following expression for downstream temperature of turbine:

$$T_t = T_{em} \left(1 - \eta_t \left(1 - \left(\frac{P_{em}}{P_t} \right)^{\frac{1-\gamma}{\gamma}} \right) \right) \quad (24)$$

It is assumed that the turbine efficiency is constant during the intake phase and is equal to 0.76. The mechanical power produced by the turbine can be calculated from the following equation:

$$W_t = c_p \left(\dot{m}_c + \dot{m}_{fuel} \right) \left(T_{em} - T_t \right) \quad (25)$$

This power is used for calculation of compressor power.

Finally the output parameter of intake system model is air mass flow rate into the cylinder.

It should be noted that turbocharger characteristic maps are not used in the modeling because of two main reasons. A), These maps were not readily available and B), we are analyzing the parameters of air intake system only in the intake phase (180 degrees of crank angle) and the variations of turbocharger characteristics are not considerable during short time of intake process.

FUEL INJECTION SYSTEM

In this section the description of Common Rail Injection (CRI) system is presented. The turning point in achieving better combustion characteristics is to provide higher injection pressures by the application of common rail injectors.

In these systems an electronic control unit calculates and sends an output signal which opens and closes a solenoid valve to obtain the required amount of fuel to be injected at the desired instant of time. Figure 2 shows a section area of a Bosch diesel injector.

In this system the injector is connected to the high pressure fuel accumulator. The fuel pressure is controlled by the pump and remains stable at a high pressure of approximately 160 MPa. In order to control the opening and closing time of needle a working chamber is located on top of the needle. When the solenoid valve is open, it creates a pressure drop in working chamber and causes a negative force which overcomes the force of needle spring and then the injection initiates.

SOLENOID VALVE

It is assumed that the solenoid is extremely fast. Based on standard magnetic circuit principles [12], the voltage equation and the consideration of back Electromotive Force (e.m.f) voltage, the current of the solenoid circuit is determined from the following differential equation:

$$\frac{di}{dt} = \frac{1}{L}\left(e - R \cdot i - \frac{F(x,i)}{i} \cdot \frac{dx}{dt}\right) \tag{26}$$

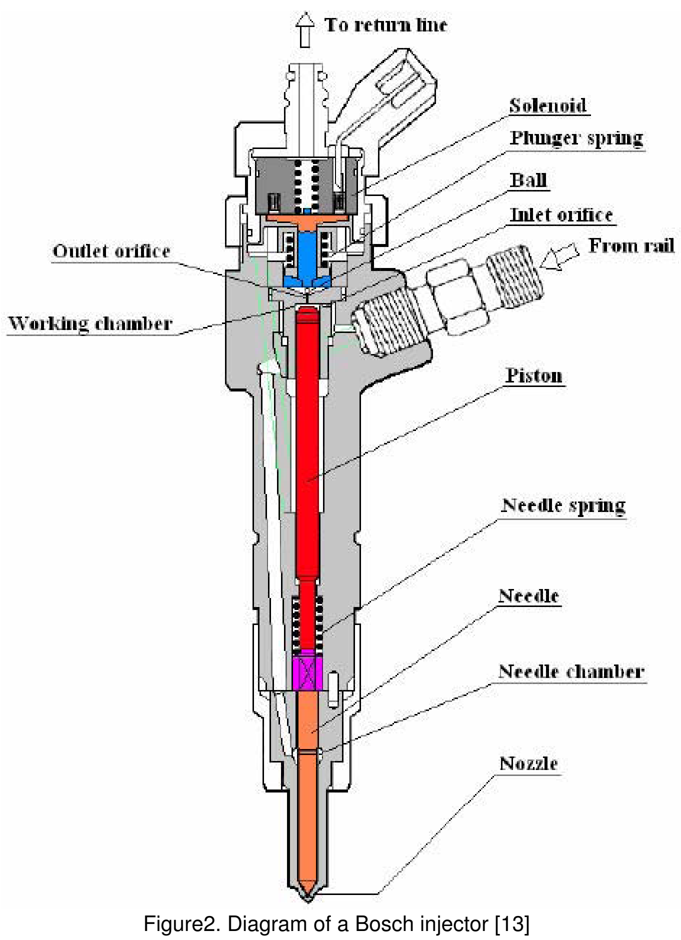

Figure2. Diagram of a Bosch injector [13]

And the movement of the solenoid plunger in injector is modeled by an equation obtained from the equilibrium of forces [14]:

$$m_1 \frac{d^2 x}{dt^2} = K_f i + P_L A_1 - m_1 g - K_1 (x + x_0) - f_1 \frac{dx}{dt} \tag{27}$$

In this section, it is assumed that coil inductance is constant and also any possible friction due to the bulging of solenoid spring is ignored.

HYDRAULIC COMPONENTS

For modeling of hydraulic parts it is considered that pressure propagation speed is infinite. Fuel leakage between the components is neglected and pipe wall expansion is not taken into calculations. Then a one-dimensional, transient and compressible flow model of CRI is built base on mass conservation and equilibrium of forces. The mass conservation relation for working chamber can be expressed as follows [15]:

$$A_2 \frac{dy}{dt} - \frac{V_{wc}}{\beta}\frac{dP_W}{dt} = K_1 A_1 \sqrt{\frac{2(P_L - P_W)}{\rho}} - K_0 A_0 \frac{x}{X_{\max}} \sqrt{\frac{2(P_W - P_R)}{\rho}} \tag{28}$$

By applying the Newton's second law to the injector piston, the piston system motion equation will be:

$$m_2 \frac{d^2 y}{dt^2} = P_L A_N - P_W A_2 - K_2 (y + y_0) - m_2 g - f_2 \frac{dy}{dt} \tag{29}$$

And the theory of "flow through an orifice" leads to the next equation for injection flow rate [6]:

$$q_I = K_I A_I \frac{y}{Y_{\max}} \sqrt{\frac{2(P_L - P_W)}{\rho}} \tag{30}$$

Lastly, the injection quantity is the integral of the injection flow rate:

$$Q = \int_0^t q_I dt \tag{31}$$

In the above-mentioned equations, fuel bulk modulus and fuel density are dependent only on fuel pressure. They are also dependent on temperature but in a smaller scale. Polynomial equations are used to define these dependencies [16].

Finally, the output of fuel injection model is fuel injection quantity.

AFR CONTROL

Air mass flow rate into cylinders and fuel injection quantity are complex non-linear functions of several interactive variables, which together present a highly coupled plant. AFR of an engine can be controlled by three different controllers. Two of these controllers are located in the air intake system and the third one is located in injection system. EGR and VGT valves as controllers play a significant role in emissions, fuel consumption and drivability of a vehicle. The settings of VGT vane and EGR valve alter the engine pumping work and cylinder charge composition, affecting fuel consumption and emissions [17]. By changing the positions of EGR and VGT valves, air flow rate into cylinders will be changed. In this case μ Analysis tools can be applied to yield good controller performance [3].

Electronic controls, which allow greater flexibility in setting of individual parameters, make electronic fuel injection an appropriate system for precisely control of injection quantity and timing. By application of the ECU, the injection is no longer dependent on the position of crankshaft. Besides, high speed solenoid valves have the capability to adjust the amount of injected fuel and therefore air-to-fuel ratio. The high fuel pressure is supplied by a pump while an ECU calculates and sends an output signal (voltage of solenoid circuit) to injector which opens and closes a solenoid valve [14]. Consequently this kind of fuel injection system can be used for control of AFR.

In this work the positions of EGR and VGT valves are set manually for different engine speeds. However the controller for AFR control is a solenoid valve in fuel injection system. In other words, the fuzzy controller calculates a modified value for bandwidth of solenoid voltage signal. This value works as a new input for the fuel injection subsystem. Modified amount of fuel injection quantity is obtained make use of this new bandwidth value.

In order to eliminate the steady state error of the system a fuzzy integrator is used. The equation for this integrator is:

$$BW_{new} = BW_{old} - C\left(AFR_{ref} - AFR_{est}\right) \qquad (32)$$

In this relation, C is the output of fuzzy controller; AFR_{ref} is the desired air-to-fuel ratio and AFR_{est} is the calculated air-to-fuel ratio. BW is representing the value of solenoid voltage bandwidth (the control signal).

CONTROL STRATEGY

A complete description of the methodology of fuzzy logic is beyond the scope of this paper, however the interested reader should refer to [18, 19] for a deeper insight. The controller used in this work was designed using the Matlab fuzzy logic toolbox, employing a graphical interface to simplify the task.

The first stage of the fuzzy controller design is to decide on a structure. From the analysis of the problem it is clear that the inputs to the controller need to be AFR error (difference between desired and actual AFR) signal and the derivation of the AFR error signal. Knowing AFR error and the AFR error derivation signals, one can decide for a new bandwidth value. The output of the controller is the C factor in the equation of fuzzy logic integrator.

The next step is defining membership functions for inputs and outputs. They are illustrated in figure 3. Inputs and the output are covered by 7 triangular membership functions. The first input (AFR error) is varying from -1 to +1. The second input is also varying from -1 to +1. The output is also changing from -1 to +1 too [20]. It should be noted that we have used normalized data base. Two gains are applied to alter the real variation bounds of AFR error and its derivation to normalized range which is between -1 and +1.

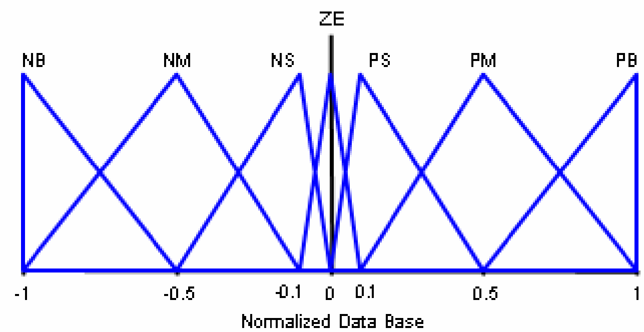

Figure3. Membership Function shape for inputs and output

The final step is to define the rules for the controller. They are simply an embodiment of the control action one wishes to be achieved. 49 rules are defined for this controller.

When the error is negative it shows that actual AFR is greater than the desired AFR. It means that injected fuel quantity is small. So injection quantity should be increased in order to reduce the AFR magnitude. Consequently, control signal bandwidth needs to be risen. On the contrary, when the error is positive it indicates that actual AFR is smaller than the desired AFR. And it tells us that injected fuel quantity is high and it should be decreased in order to raise the AFR magnitude. Therefore, control signal bandwidth needs to be declined. These statements are the fundamentals of defining the base rule. Based on the defined membership functions for inputs and output and the notation which is used according to figure 3, the following rules are derived for the purpose of AFR control. The rules that are used are expressed in table 1.

Table.1. Rule base used for the purpose of AFR control

$\dot{e}$ \ e	NB	NM	NS	ZE	PS	PM	PB
NB	PB	PB	PB	PB	PM	PM	PS
NM	PB	PM	PM	PM	PS	PS	ZE
NS	PM	PM	PS	PS	PS	ZE	ZE
ZE	PS	PS	PS	ZE	NS	NS	NM
PS	ZE	ZE	NS	NS	NM	NM	NM
PM	ZE	NS	NM	NM	NM	NB	NB
PB	NM	NB	NB	NB	NB	NB	NB

The inference mechanism for this controller is mamdani inference rule [19]. For “fuzzy and” method minimum, for “implication” method product, for “aggregation” method sum and for “deffuzification” method centroid of area defuzzifier (centroid) is chosen. Center average defuzzifier has two main advantages comparing with other methods of defuzzification: justifiability and continuity.

MODEL VALIDATION

Output of the air intake model is measured in different engine speeds and the results of this model are compared with available experimental data [3]. The comparison between simulation results and experimental data is shown in figure 4.

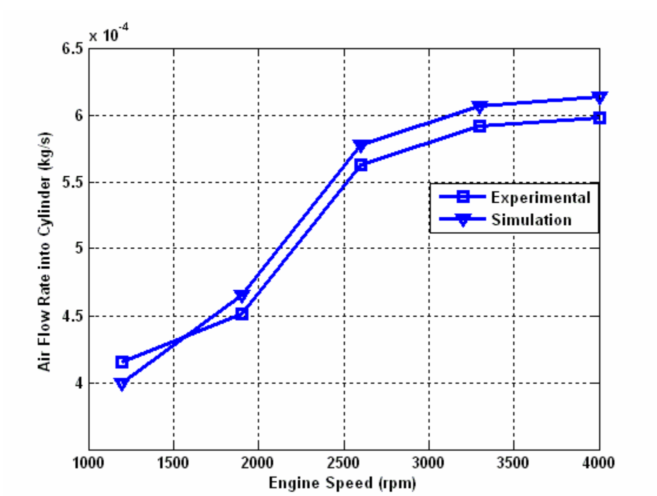

Figure4. Comparison of simulated and experimental data for air intake system

Measurements have also performed for various solenoid control pulses with durations of 4, 6, 8 and 10 milliseconds. The results of measurements together with experimental data [21] are shown in table 1. These figures show a very good match between the experimental and simulation results.

Table.2. Comparison of simulated and experimental injection quantities per injection

Duration of Pulses (ms)	Measured quantity of injection (mg)	Simulated quantity of injection (mg)	Error (%)
4	114	115.89	1.6
6	172	169.017	-1.7
8	227	222.402	-2
10	284	275.825	-2.8

The magnitude of voltage during injection is assumed to be constant (15 V). Generally speaking, there is a good consistency between simulation results and experimental data.

SIMULATION RESULTS

All the above-mentioned nonlinear and coupled differential and ordinary equations are coded in Matlab/Simulink software. The basic engine parameters, which are used in simulation, are listed in the following table.

Table.3. Basic engine parameters

Cylinder bore	0.0725 (m)
Compression ratio	17:1
Crank radius	0.05 (m)
Connecting rod length	0.12 (m)
Displacement volume	$4.2*10^{-4}$ (m^3)
Piston stroke	0.1 (m)

Intake manifold pressure and temperature diagrams are shown in figures 5 and 6. According to these graphs, both temperature and pressure have increased. This rise is because of turbocharger existence and also augmentation of a portion of exhaust gases through EGR system into the intake manifold. These figures show a 13 degrees increase in temperature and 10 kPa increase in pressure during the intake phase in intake manifold.

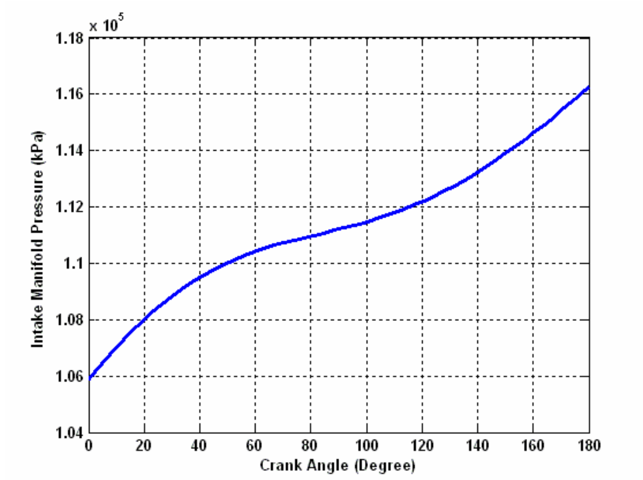

Figure5. Intake manifold pressure

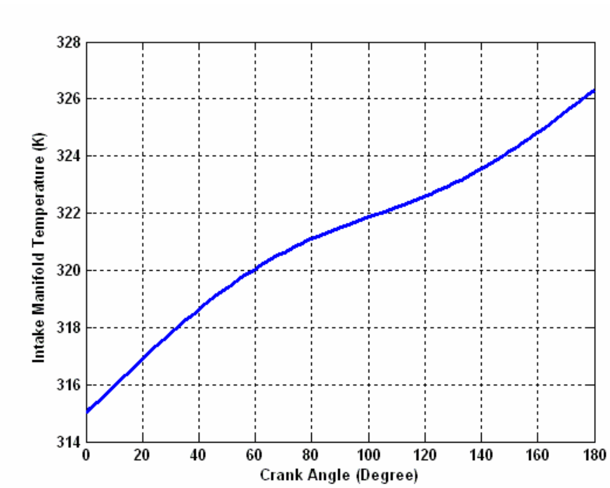

Figure6. Intake manifold temperature

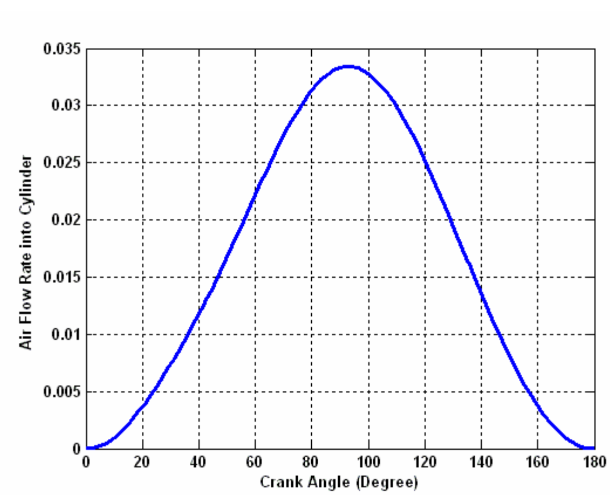

Figure7. Air flow rate into cylinder

Mass of air entering the cylinder is one of the two main parameters which are needed for computing and controlling the AFR. Figures 7 and 8 show the profile of cylinder air mass flow rate and cylinder charge during the intake phase.

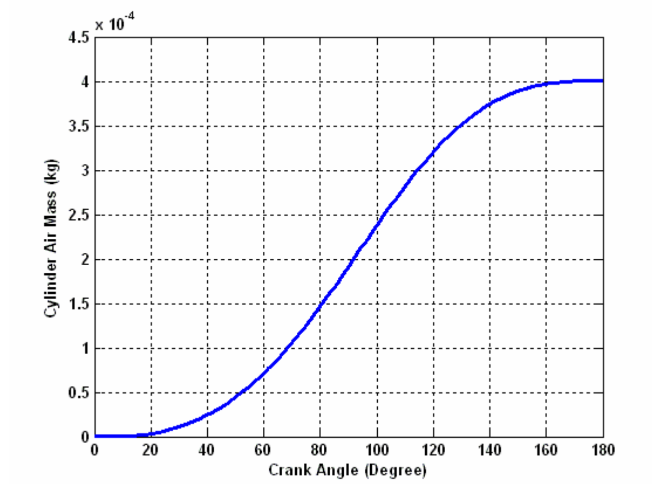

Figure8. Mass of air in the cylinder

Figure 7 shows the standard profile for opening and closing of air intake valve. Increase in valve lift results in increase in air flow rate into cylinder. When the valve reaches its maximum lift, air flow rate has its greatest magnitude. Integrating from air flow rate into cylinder we can calculate the amount of air entered the cylinder.

At the end of exhaust phase in previous engine cycle, the temperature of exhaust gases was high (around 700 K). As the air intake valve opens, Fresh air enters the cylinder and temperature decreases. At the end of intake phase the temperature in the cylinder reaches the value of 315 K (near to intake manifold temperature). Cylinder temperature variation is shown in figure 9.

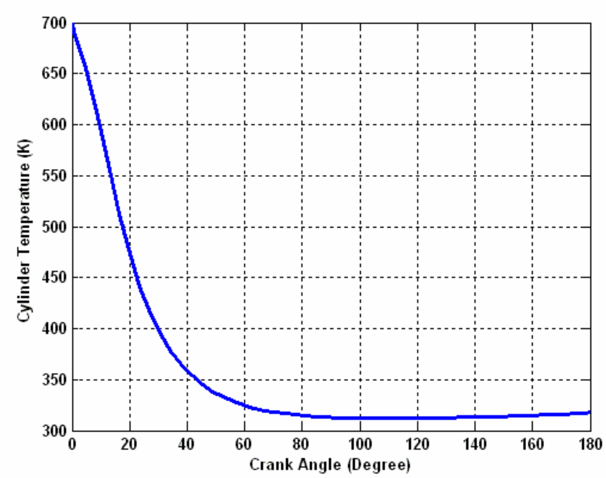

Figure9. Cylinder temperature variations

According to figure 10, cylinder pressure increases nominally at the beginning of intake phase. It has two main reasons: A), intake manifold pressure is high and B), air intake valve is approximately closed at the very beginning of intake process. Because of accelerating downward movement of piston and more opening of intake valve, pressure declines and again by entering more air into cylinder and by completing the filling process cylinder pressure rises and approaches the intake manifold pressure at the end of intake phase.

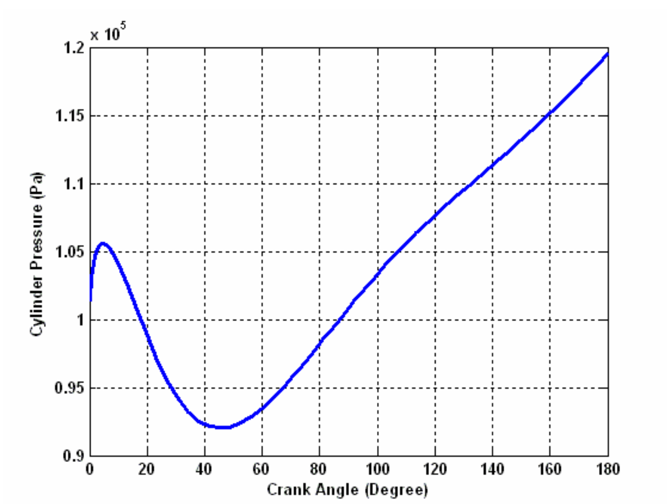

Figure10. Cylinder pressure variations

Engine speed in above figures is 1200 rpm. The values of turbine valve position and EGR valve position are set to 1 and 0.35 respectively. It should be noted that EGR and VGT valve positions are altered appropriately in assorted engine speeds. It is also assumed that intake process starts precisely at TDC and ends exactly at BDC. So, the duration of intake phase is 180 crank angle degrees.

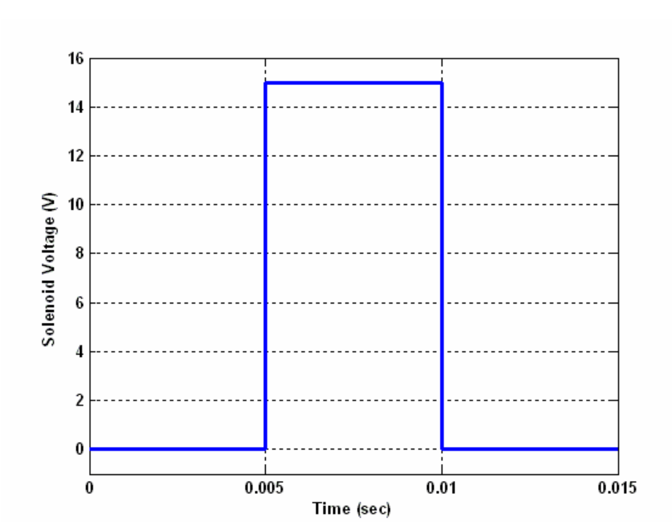

Figure11. Control signal (voltage) of solenoid

A control pulse of 15 V DC with the duration of 5 milliseconds is applied to solenoid circuit in order to obtain the quantity of injection. In computing the quantity of injected fuel, the time of injection is not important. So the simulation results are shown versus a sample time rather than crank angle. 5 millisecond duration of injection mean that injection is occurred in 36 degrees of crank angle, no matter when it happens. The shape of this signal is seen in figure 11.

The graph of needle movement is also illustrated in figure 12. Solenoid plunger movement and needle movement both comply with the shape of applied voltage to the solenoid circuit. According to figure 12, needle movement and applied voltage are not simultaneous and the needle moves with a short delay in comparison with voltage signal.

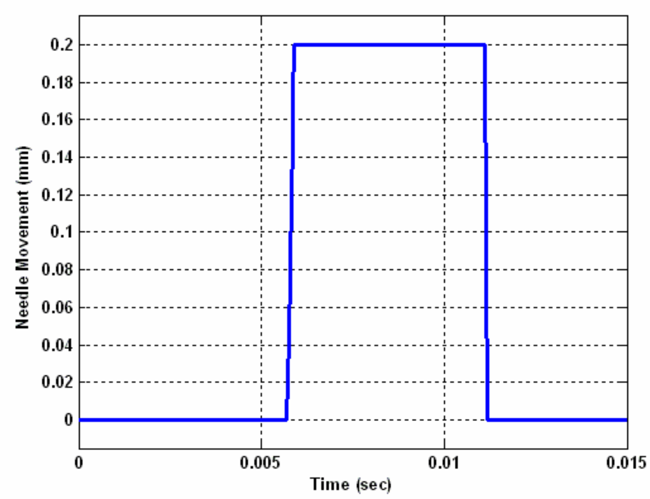

Figure12. Needle movement

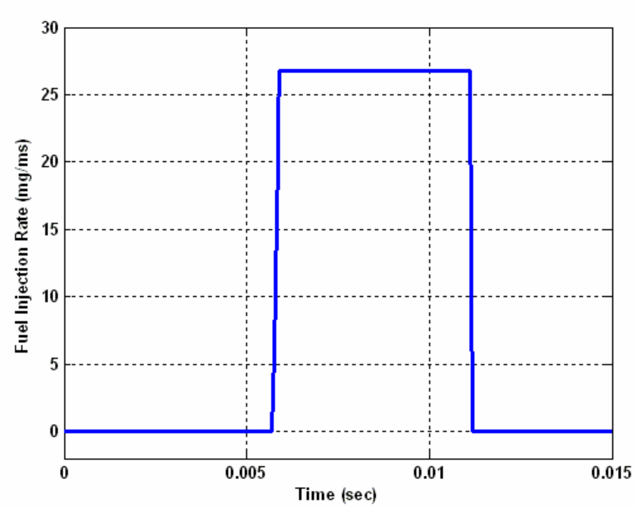

Figure13.Rate of injection

The profile of injection rate with a six nozzle injector during injection phase is shown in figure 13. The shape of Injection rate is following the shape of needle movement. As the needle lifts from its seat injection initiates and injection rate rises as the needle lifts more. When the needle moves back to its first position injection ceases.

By integrating the injection rate in one injection duration we can obtain the quantity of injection in each time during injection period. The shape of injection quantity variation is shown in figure 14.

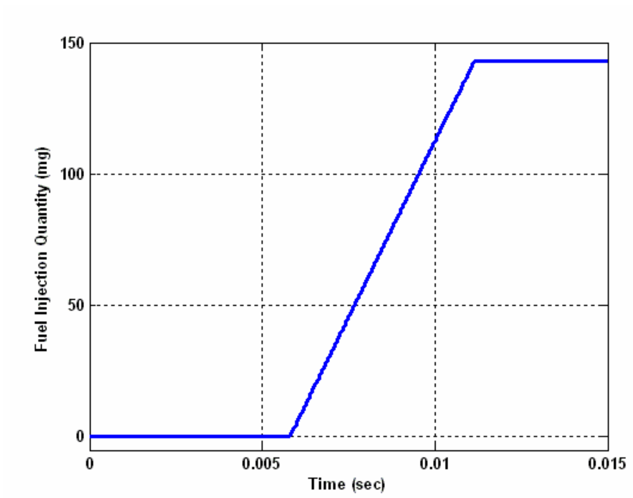

Figure14. Injection quantity in one injection duration

Figure 15 illustrates the behavior of designed fuzzy logic controller. Fuzzy logic controller reaches the desired value of AFR (in this case 45) in about 0.8 seconds. This figure shows the performance of controller when the engine speed is 1200 rpm. So this controller is capable to adjust the AFR in less than 8 complete cycles.

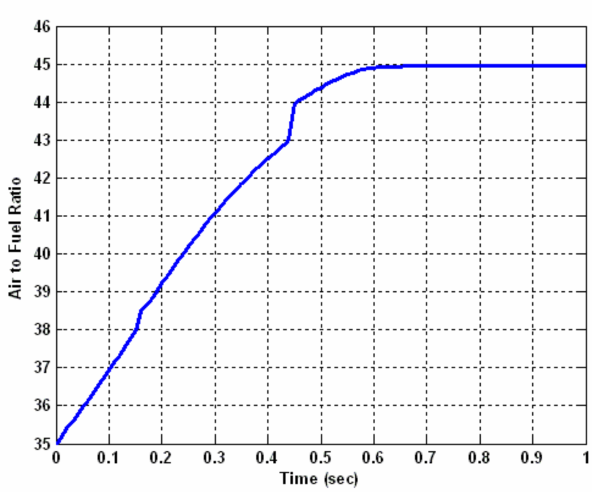

Figure15. Behavior of fuzzy logic controller

Each engine complete thermodynamical cycle at 1200 rpm speed is 0.1 seconds.

Figure 16 shows the variations of error signal. At first the error is about 10 and gradually after 0.8 seconds, error declines and reaches to zero. It means that the controller has reached the value of AFR to desired value exactly. The value of AFR was at first 30.

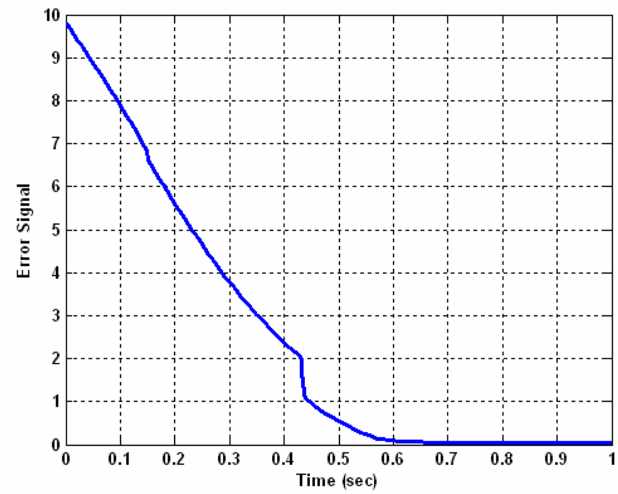

Figure16. Variation of error signal magnitude

Figure 17 represents the profile of control signal variations. The actual AFR is less than the desired AFR. So the AFR should be increased. Decreasing the value of solenoid voltage bandwidth, declines the amount of injected fuel and this will culminates in raised AFR. As it is seen in figure 17 the signal bandwidth has fallen from 0.00343 seconds to 0.00257 seconds during a 0.8 second period.

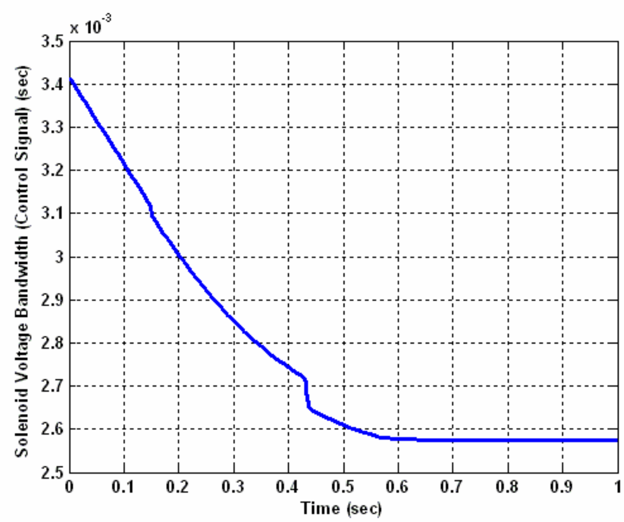

Figure17. Variations of solenoid voltage signal bandwidth

The controller performance as it is observed in figures 15 to 17 is quite fast and it reaches a plateau at the end where the AFR is equal to the desired value.

CONCLUSION

In this study, detailed theoretical models for air intake system and electronic fuel injection system of a turbocharged diesel engine based on physical relations are proposed. These mathematical models are programmed in Matlab/Simulink software environment. The objective has been to calculate the amount of air entering the cylinder during intake process and injected fuel quantity during one injection period and eventually AFR control. A fuzzy controller based on fuzzy logic methodology is designed for the purpose of air-to-fuel ratio control of the engine.

The main conclusions are as follows:

- The simulation results for air intake model agree reasonably well with available experimental data.
- There is a good consistency between simulated and measured data for electronic fuel injection system.
- Air-to-fuel ratio control is performed make use of a fuzzy controller with 49 rules.
- Fuzzy controller performance is quite fast. It corrects the value of AFR and adjusts its quantity to the desired value in less than 8 engine complete cycle by varying the fuel injected quantity through changing the magnitude of solenoid voltage signal bandwidth.

Furthermore, there are still more issues for broader research and dissection. The further steps to improve or continue the existing work are as follows:

- By combustion modeling the properties of exhaust gases such as pressure and temperature can be calculated more precisely. This task helps to obtain a more accurate input data for the turbine model.
- Augmentation of other diesel processes in a cycle (compression, combustion and expansion) culminates in a full cycle diesel engine model that can be used for simulation and control objectives.
- Pilot injection and its effects on combustion and engine performance parameters can be investigated make use of introduced model for fuel injection system.
- Other parameters such as engine speed and torque can be controlled simultaneously with air-to-fuel ratio control. Moreover, the presented model can be calibrated and plied in heavy duty diesel engines.

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NOMENCLATURE

A	:	Cross sectional area
C_p	:	Specific heat at constant pressure
C_v	:	Specific heat at constant volume
e	:	Voltage
F	:	Force
K	:	Spring coefficient
m	:	Mass flow rate
M	:	Mass
P	:	Pressure
r	:	Crank radius
R	:	Gas constant, Resistant
T	:	Temperature
u	:	Velocity
V	:	Volume
W	:	Power, Work
Φ	:	Crankshaft angular position
γ	:	Air specific heat ratio
η_c	:	Compressor isentropic efficiency
ρ	:	Density

Subscripts

0	:	Ambient conditions, Inlet orifice
1	:	Outlet orifice
2	:	Piston
ad	:	Admission
c	:	Compressor
c,ic	:	Downstream of intercooler
cool	:	coolant
csa	:	Valve section
cyl	:	Cylinder
eg	:	Exhaust gases
egr	:	Exhaust gas recirculation
L	:	Fuel line
he	:	Heat exchanger
im	:	Intake manifold
em	:	Exhaust manifold
t	:	Turbine
W	:	Working chamber