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Modified Mean Value SI Engine Modeling (EGR Included)

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ABSTRACT

One way of applied modeling for dynamic systems, especially complex and multi input – multi output (MIMO) systems, is using the Mean Value Method which in itself is simple and useful. One way of using the Mean Value Method is the automotive engine. In most articles, the Mean Value Method is accompanied with the assumption of constant manifold temperature which in some cases, for example, Exhaust Gas Recirculation (EGR) is insufficient. In this article, a Modified Mean Value model is presented. This model, with the assumption of variable manifold temperature, is used on an SI engine with EGR. It is simulated in a SIMULINK atmosphere. Finally, the model output is validated with the data of a real engine.

KEYWORDS: Mean Value Engine Modeling, EGR, SI Engine, Manifold Air Temperature.

INTRODUCTION

One of the applied methods for dynamic systems, especially complicated and multi input – multi output (MIMO) ones, is using the mean value method. This method applies the mean value of a parameter through 10000 cycles duration which includes the dynamic behaviour of the system. One of the applied fields of this method is the SI engine. Mean Value modeling for the case of engine is briefly called MVEM, which is used because of its simplicity and usefulness. This method is used for the most of dynamic systems. MVEM was developed for the first time in the Denmark University of Technology in 1990 [1]. After that, this method was developed more and more. Some basic and important examples of the applied MVEM are references [2], [3] and [4]. The interesting fact in all of these references is the assumption of constant manifold air temperature. Development of engine equipment, devices and necessity of system monitoring, causes this assumption to be insufficient. This case especially will be important when automobile is equipped by Exhaust Gas Recirculation that hereafter will be called as EGR system. With applying EGR on the system, manifold air temperature (MAT) will significantly be affected. Applying EGR system on the SI engine is one of the most important examples of the weakness of MAT constancy assumption. In this article, a modified mean value model has been applied to an SI engine. It is equipped with EGR system which compared to a similar model, presents more advantages. This model is simulated in SIMULINK atmosphere and is validated with a real engine.

MODELING PHASE

This section is about using principles of mean value modeling which concerns with the SI angine.
SI engine dynamic modeling is categorized to three subsections as mentioned below:

- Engine Revolution Dynamics
- Fuel Dynamics
- Manifold Air Dynamics

In the following subsections, each one will be modeled through MVEM.

Engine Revolution Dynamics

Here, used relations are on the base of Euler's equations of the rotational dynamics (Eq. 1).

$$\sum_i M_i = I\alpha \tag{1}$$

However, regarded engine torque components are as below:

- gross produced torque M_g
- frictional torque M_{fric}

each of these components are formulated through equations 2 and 3.

$$M_g = (1-E)\frac{\eta_{fc}\, Q_{HV}\, \dot{m}_f}{\frac{2\pi}{60} n} \tag{2}$$

$$M_{fric} = (1-E)\left[a_0 + a_1 n + a_2 n^2 + (a_3 + a_4 n)\, p_i\right] \tag{3}$$

As it is seen, in modeling the frictional torque, a polynomial has been used. Descriptions for the terms and their units is presented in nomenclature. It is clear that:

$$M = I_t \frac{2\pi}{60}\dot{n} \tag{4}$$

However, equation regards to the engine rotational dynamics will be developed.

$$\dot{n}=(1-E)\frac{60}{2\pi I_t}\left\{\frac{\eta_{fc}\,Q_{HV}\,\dot{m}_f}{\frac{2\pi}{60}n}-\left[a_0+a_1 n+a_2 n^2+(a_3+a_4 n)\,p_i\right]\right\} \quad (5)$$

Fuel Dynamics

Used relations are founded on Mass Conservation Law. Thus, fuel flow dynamics can be formulated as follows. Note that, equivalence ratio assumed to be one. Thus, air to fuel ratio is assumed constant and equal to 14.6.

$$\dot{m}_f=(1-X)\dot{m}_{fi}+\dot{m}_{ff}=\frac{\dot{m}_{ap}}{14.6} \quad (6)$$

$$\ddot{m}_{ff}=\frac{1}{\tau_{ff}}\left(-\dot{m}_{ff}+X(1-Y)\dot{m}_{fi}\right) \quad (7)$$

Notes and descriptions are listed in nomenclature.

Manifold Air Dynamics

Similar to the past section, Mass Conservation Law is the foundation of the modeling process. The most important difference between the traditional MVEM and the method used here, is mentioned in this section. According to the perivious sections, most of the papers which use mean value modeling, apply manifold air temperature constancy assumption. In [5], for modeling the SI engine which includes EGR, air flows in the manifold modeled based on the adiabatic assumption. Differences between results in the adiabatic solution and the isothermal one in case of manifold air pressure, is not negligible. It is the main driver for preparing this modified mean value engine model. Both isothermal and adiabatic assumptions are used, while the former is used for the air presseure dynamics and the latter, for the air temperature dynamics.

Manifold Air Pressure Dynamics

With the isothermal assumption that was introduced in [1] manifold air temperature can be modeled as follows:

$$\dot{m}_i=\dot{m}_{at}-\dot{m}_{ap}+\dot{m}_{EGR} \quad (8)$$

$$\dot{p}_i=\frac{R T_i}{V_i}\dot{m}_i \quad (9)$$

From equations 8 and 9, it is resulted that:

$$\dot{p}_i=\frac{R T_i}{V_i}\left(\dot{m}_{at}-\dot{m}_{ap}+\dot{m}_{EGR}\right) \quad (10)$$

Each of the mas flow rate components can be formulated as below:

$$\dot{m}_{at}=\dot{m}_{at0}+\dot{m}_{at1}\,\beta_1(\alpha)\beta_2(p_i) \quad (11)$$

$$\beta_1(\alpha)=1-\cos(\alpha-\alpha_0) \quad (12)$$

$$\beta_2(p_i)=sign\left(\frac{p_i}{p_a}-\frac{p_a}{p_i}\right)\times$$

$$\begin{cases}\sqrt{(p_r)^{\frac{2}{\kappa}}-(p_r)^{\frac{\kappa+1}{\kappa}}}, & (p_r)\geq\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}\\ \sqrt{\frac{\kappa-1}{\kappa+1}\left(\frac{2}{\kappa+1}\right)^{\frac{2}{\kappa-1}}}, & otherwise\end{cases} \quad (13)$$

$$p_r=\min\left(\frac{p_i}{p_a},\frac{p_a}{p_i}\right)$$

$$\dot{m}_{ap}=\frac{V_d\,\eta_v}{120\,R}\frac{n\,p_i}{T_i} \quad (14)$$

$$\eta_v=\eta_{v0}+\eta_{vn1}\,n+\eta_{vn2}\,n^2+\eta_{vp1}\,p_i$$

$$\dot{m}_{EGR}=E\,\dot{m}_{at} \quad (15)$$

In the above equations, $\dot{m}_{at}$, is the air mass flow over the throttle valve, $\dot{m}_{ap}$, the air mass flow that enters the combustion chamber and $\dot{m}_{EGR}$, the EGR mass flow rate. As can be seen in the equation 15, $\dot{m}_{EGR}$ is modeled as a percentile of the $\dot{m}_{at}$ and in addition, is subtracted from $\dot{m}_{at}$. On the other hand, it is assumed that no oxygen is in the EGR. Another problem that is encountered, is the modeling of the manifold inlet gas temperature. With the assumption of constant EGR temperature, the manifold inlet gas temperature hereafter will be called the gas mean temperature which is modeled as follows:

$$T_m=(1-E)T_a+E\,T_{EGR} \quad (16)$$

Thus, it is resulted that:

$$\dot{p}_i=\frac{R(1-E)}{V_i}\times\left\{\left[\dot{m}_{at0}+\dot{m}_{at1}\,\beta_1(\alpha)\beta_2(p_i)\right]T_m-\frac{V_d\,\eta_v\,n\,p_i}{120\,R}\right\} \quad (17)$$

Notes and descriptions are listed in nomenclature.

Manifold Air Temperature Dynamics

Cosidering that isothermal assumption naturally never can model the manifold air temperature, the adiabatic solution for temperature dynamics is applied here.

From the thermodynamic first law, it is exerted that:

$$\dot{m}_{at}\,h_a+\dot{m}_{EGR}\,h_{EGR}-\dot{m}_{ap}\,h_i=\frac{d(mu)}{dt}=\dot{m}_i\,c_v\,T_i+m_i\,c_v\,\dot{T}_i \quad (18)$$

Assuming air as a perfect gas, enthalpy and constant volume, air heating capacity will be rewrited as below:

$$h_x=\frac{T_x\,R\kappa}{\kappa-1} \quad (19)$$

$$c_v=\frac{R}{\kappa-1} \quad (20)$$

In addition, perfect gas law says:

$$m_i = \frac{p_i V_i}{R T_i} \tag{21}$$

Thus, it is resulted that:

$$\dot{T}_i = \frac{\dot{m}_{at} T_a R \kappa T_i}{p_i V_i} + \frac{\dot{m}_{EGR} T_{EGR} R \kappa T_i}{p_i V_i} - \frac{\dot{m}_{ap} R \kappa T_i^2}{p_i V_i} - \frac{\dot{m}_i R T_i^2}{p_i V_i} \tag{22}$$

After simplifying the equations and applying EGR mass flow rate as some $\dot{m}_{at}$ percentile and finally use gas mean temperature, the final equation, equation 23, will be developed:

$$\dot{T}_i = \frac{R T_i}{p_i V_i}(1-E) \times \left\{ \left[\dot{m}_{at0} + \dot{m}_{at1} \beta_1(\alpha) \beta_2(p_i)\right](T_m \kappa - T_i) - \frac{V_d \eta_v n p_i}{120 R}(\kappa - 1) \right\} \tag{23}$$

Notes and descriptions about the parameters and constants used here, are listed in nomenclature.

MODEL SIMULATION PHASE

This modified model resulted in, will be simulated in SIMULINK toolbox of MATLAB and model outputs reviewed here. Figure 1, shows a schematic view of the inputs, outputs and links among them in SIMULINK atmosphere. As can be seen in figure 1, the model inputs are throttle angle, EGR ratio. On the other hand, outputs are engine revolution, manifold air pressure and manifold air temperature. Because of the use of air to fuel constant mass flow rate ratio (14.6), fuel dynamics in this project have no state variable.

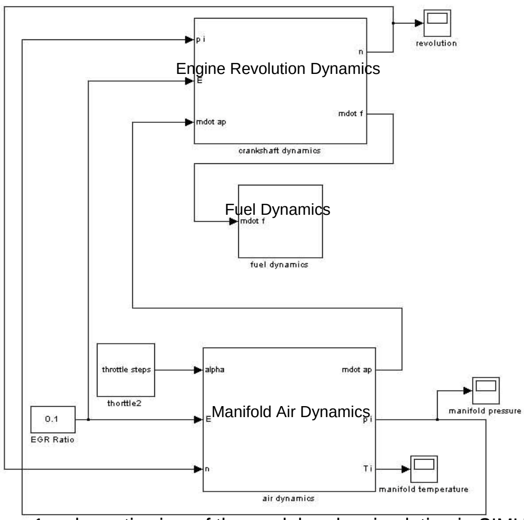

Figure 1. schematic view of the model under simulation in SIMULINK

Figure 2, shows behavior of the throttle angle through time. Concerning EGR ratio, simulation will be based on two situations; without EGR and with 10% EGR. EGR tmeperature is assumed to be constant and equal to 448 K.

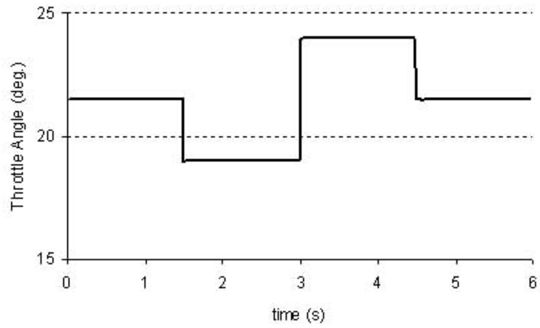

Figure 2. behavior of throttle angle as input through the time

MODEL VALIDATION

Achieved results will be compared and validated by an experimental results of the 1275 cc Leyland engine under similar situations of [5]. Naturally, similar to simulation, validation is based on two case studies: without EGR and with 10% EGR. Sections 4.1 and 4.2 show validation in both case studies respectively as follow.

First Case Study: Engine without EGR

In the first case study, engine is modeled without EGR. Figure 3, shows the behavior of engine revolution vs. time. In this figure and all the others, the solid lines represent model results and the dashed lines show the experimental ones.

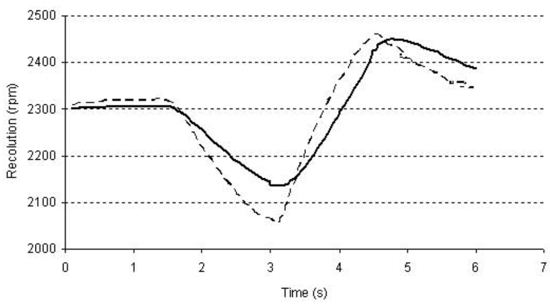

Figure 3. engine revolution behavior through time without EGR

It is clear that the starting of the simulation ,2000 rpm, is neither idle speed nor overrun situation. The maximum error of the model in this case is about 5 percent. Manifold air pressure output has been modeled in figure 4.

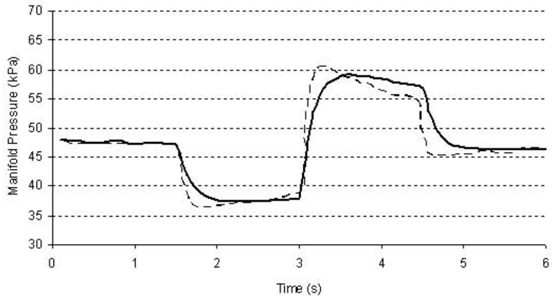

Figure 4. manifold air pressure behavior through time without EGR

As it can be seen the error is too low while the maximum error in this case is about 3 percent. Figure 5, shows manifold air temperature vs. time.

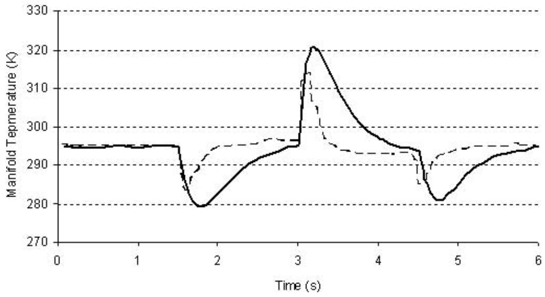

Figure 5. manifold air temperature behavior through time without EGR

In figure 5 and comparing with other figures, the error occurred in manifold air temperature is more than the others; while the maximum error is about 8 percent. In all of these last four figures, significant differences have been occurred in t = 1.5 sec, t = 3 sec and t = 4.5 sec which relate to sudden changes in input regime. Further achievements of these results will be discussed in section 5. It is clear that, if the isothermal assumption is used, the results of the figure 5, show a straight line and therefore, these sudden changes in imput regimes won't affect the results. This is the main advantage of this model .

Second Case Study: Simulation with 10% EGR

In the second case, 10% EGR has been applied in the model and the EGR effect on the state variables has been reviewed. Figure 6, shows behavior of the engine revolution vs. time.

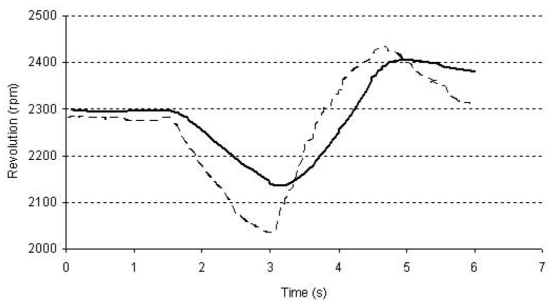

Figure 6. engine revolution behavior through time with 10% EGR

As it can be seen, the maximum error is about 5 percent. In addition, EGR will result in not significant but little loss of revolution. Figure 7, shows manifold air pressure vs. time .

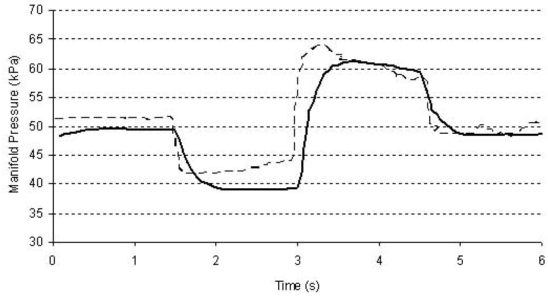

Figure 7. manifiold air pressure behavior through time with 10% EGR

The maximum error has been obtained about 8 percent. A comparison between figures 4 and 7, show that the manifold air pressure with increasing EGR ratio, increases. Figure 8, shows manifold air temperature behavior through time with 10 percent EGR.

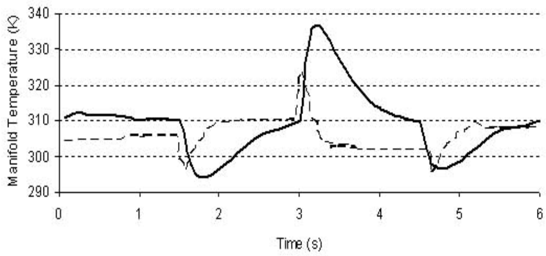

Figure 8. manifold air temperature through time with 10% EGR

The maximum error occurred in t = 3 sec is in about 10 percent. It is clear that in both case studies, manifold air temperaure has more error vs. other variables. The major reason for this fact is introducing the gas mean temperature and effect of EGR ratio on volumetric efficieny. A comparison of figures 5 and 8, with increasing EGR ratio, manifold air temperature increases. In all of these last four figures, significant differences have occurred in t = 1.5 sec, t = 3 sec and t = 4.5 sec which relate to sudden changes in input regime. Further achivements of these results will be discussed in section 5.

Conclusion

The modified model used here in comparison with the traditional mean value engine models (isothermal models), is applicable for the wider range of engine performance. The most significant error in both case studies occurred in manifold air temperature. For solving this problem, developing more precise manifold inlet gas mean temperture and volumetric efficiency will be useful. Results achieved by this model show that neglecting the instances that signify increase or decrease of input regime in the manifold air temperature is approximately constant. The main advantage of this article is applicability of the sudden changes of inputs regimes. In comparison of this article and [5], deficiency of the paradox that occurred in manifold air pressure dynamics without EGR has been eliminated.

NOMENCLATURE

n	engine revolution (rpm)
M	torque (Nm)
Q	heating value
$\dot{m}$	mass flow rate (kg/s)
p	pressure (kPa)
T	temprature (K)
X	part of the fuel which accumulated on the wall
V	volume (m^3)
E	EGR ratio
p_r	manifold to ambient pressure ratio

Greek Letter

α	throttle angle (deg)
η	efficiency
κ	gas atomicity coefficient
τ	time coefficient (sec)

Subscripts

t	total
g	gross
fric	frictional
f	fuel
a	ambient
at	past over the throttle plate
ap	past over the intake valve
i	intake manifold
fc	fuel conversion
v	volumetric
ff	fuel film
fi	fuel injected
d	displacement
EGR	related to EGR
m	mean
n	engine speed
p	intake anifold pressure

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