Journal of Simulation and Analysis of Novel Technologies in Mechanical Engineering
13 (4) (2021) 0053~0065
HTTP://JSME.IAUKHSH.AC.IR

Research article

Dynamic optimization of load step transient response of a turbocharged spark ignition engine focusing on valves timing

Mehdi Keshavarz*, Ahmad Keshavarzi

Department of Mechanical Engineering, Khomeinishahr Branch, Islamic Azad University, Isfahan, 84175-119, Iran

*keshavarz@iaukhsh.ac.ir

(Manuscript Received --- 07 Feb. 2022; Revised --- 06 Mar. 2022; Accepted --- 09 Mar. 2022)

Abstract

Demand increase for reduction of fuel consumption and emissions of internal combustion engines has caused technology development in the automobile industry. A successful solution is downsizing the internal combustion engines and adding turbochargers to them. One of the consequences of adding turbocharger to the engine is slower transient response of the engine in comparison to the naturally aspirated engines. In this paper, the engine simulation is done in one-dimensional software GT-POWER and the torque transient response is being focused. For optimization, the coupling between two software of GT-POWER and MATLAB SIMULINK is used to find a rapid way for calculation of a proper strategy in order to utilize variable valve timing (VVT) technology during the transient. Variable valve timing in this study refers to opening and closing timing of inlet and exhaust valves. The optimization target is to maximize the torque integral during time interval of the transient. The transient in this paper is the one in which the engine speed is constant and the load increases rapidly and suddenly to a special value (step increase of load). Improved genetic algorithm is used for optimization. The studied engine is 1.65 liters EF7-TC which is a spark ignition engine equipped with turbocharger.

Keywords: Turbocharged; VVT; Spark ignition engine; Optimization

1- Introduction

Internal combustion engines are still the main power generation sources in vehicles. However, internal combustion engines continuously need improvements to be able to meet customer requirements and government laws. Customer requirements are less fuel consumption, more output power and better driveability; government laws include reduction of environmental emissions. A technology which has become very prevalent during recent decades for decrease of fuel consumption in spark ignition internal combustion engines is downsizing the engine and equipping it with a turbocharger. Since turbocharger gets its power from the output gases, in the transient when the gas throttle opens suddenly and due to inertia of the turbocharger components, a time period takes for the engine to reach to the full load. This time interval is called the turbocharger lag. In order to make better the transient torque response and hence lag reduction, use of turbocharger with variable valve timing system is suggested. Increase of independent variables due to different technologies lead to complexity of equations and increase of actuators makes many problems in the engine transient. Also, with the increase of variables' degree of freedom, it is more probable for the engine to get away from its optimal state during the transient. Most of engine's working cycles happen in the transient and phenomenon of waves' interaction in the inlet and exhaust manifold, which is one of the main reasons of VVT technology appearance, has more effects on the engine performance and volumetric efficiency in the transient. Therefore, methodology of the engine transient control is an essential complementary for the engine steady state. Most researches in the field of VVT technology deal with valves timing optimization during steady state condition. Some of them are included in the references [1-10]. Some research are studied for analysis of the transient performance of spark ignition engines; however, less attention is paid to study of simultaneous effect of valve's four events variations on the transient response.

Turbocharger optimization for fast transient response is studied by Ericson et al. [11]. The problem includes simulation in a one dimensional modeling environment. In this study, a by-pass valve is considered after the compressor to model the additional energy which accelerates the turbine in the transient. This kind of modeling helps to simulate the transient in the steady state. In this study, the transient is divided into four different parts. The model states at the beginning of each time interval is saved for the optimization. Then, these states are simulated in the steady state and various VGT effects are analyzed. This study concludes that the regulations which maximize the product of volumetric efficiency and output pressure in the steady state result in good transient performance.

Bozza et al. [12] made a model of SI engine with direct injection and evaluated the effective factors on the transient performance with different turbochargers. They concluded that the size increase of turbine inlet chamber provides weaker transient boost pressure. Also, a bigger turbocharger provides weaker transient response while the boost pressure of this turbocharger is more.

Lefebvre and Guilain [13] made a one dimensional model of turbocharged engine in the GT-POWER software and compared the transient torque response with the engine measured values. In this study, it is showed that deviations from the equivalence ratio (λ) have major effects on the transient rotational torque response. They also presented methods to improve model capability for anticipating the transient torque response. The first proposed method is to associate modeling parameters of Wiebe function (combustion period and CA_{50}) to the engine speed and load. The second method is that the heat transfer model inside cylinder is a function of engine speed and load.

Kleeberg et al. [14] compared control strategies of variable valve timing for fast transient response of a direct injection turbocharged engine. In this study, two strategies are compared with each other. The first strategy has a large overlap and the second has a less one. The results show that larger overlap provides less torque in the beginning of the transient; however, it provides more torque at the end of the transient. It is also indicated that the input air temperature has major effect on the transient response. Modeling of a turbocharged SI variable geometry engine is done by Ericson et al. [15]. This modeling is based on the mean value engine model (MVEM) to analyze different characteristics of pressure boost of SI engine. In this simulation, optimization strategy of the transient response with an always completely closed variable geometry turbocharger is compared with a fuel consumption optimization strategy with a completely open variable geometry turbocharger. With this strategy, fuel consumption is about 16% more and the time response reduces about 1 second during NEDC. Response time is reduced about 0.3 second for the fuel consumption optimization controller with the help of electric power.

In [16], a simulation is done in the GT-POWER software in which valve overlap effects and pressure boost technologies on the transient rotational response are analyzed. The studied engine is a 1-liter turbocharged engine(Table 1). At first, valve overlap effects in the transient are studied with overlaps of 0, 10, 20, 30 and 40 degrees. The outcome of this study is that with the increase of valve overlap, the transient rotational torque response has become faster.

Table 1: Geometry characteristics of engine EF7-TC

Cylinder internal diameter (mm)	78.6
Stroke (mm)	85
IVO (aTDC)	343 degree
IVC (aTDC)	255 degree
EVO (aTDC)	125 degree
EVC (aTDC)	37 degree
Length of connecting rod (mm)	133.5
Compression ratio	9.81
Configuration and No. of cylinders	4 cylinders in-line

As it can be seen from the mentioned researches, few studies use optimization methods to reach proper transient performance. Most of them use feed-forward control designs to control actuators (like valve timing). Namely, most researches in the field of transient performance compare open loop controllers. Also, a limited number of previous researches have studied optimization of transient process, dynamically. Moreover, in previous studies related to the valve timing and transient, the valve overlap effect on the transient is only studied due to model complexity. However, in this study, all four events of the valve are optimized simultaneously for good transient performance.

In this paper, GT-POWER software is used due to its simulation ability and accuracy on the internal combustion engines researches. First, all the engine components are modeled in the software. Then, the model is verified based on some experimental results. After model verification and assuming all the engine geometry conditions to be constant, the built model in the GT-POWER software is coupled with MATLAB SIMULINK for control of inputs and outputs with improved genetic algorithm. The results demonstrate that the valves proper timing improves the transient response of turbocharged engine.

2- Combustion model

There are several ways to model the combustion in GT-power [17]. Generally, the combustion models existing in the GT-POWER software are modeled base on two theories: one-zone and two-zone. most of GT-POWER combustion models are of the two-zone kind. This section will provide details on the combustion calculations that occur during two-zone combustion. two-zone combustion is a combustion model with two distinct zones – unburned and burned. All spark ignition combustion models in GT-POWER are two-zone. The two zones are normally modeled with a separate temperature for each zone, but can optionally be specified to have the same temperature [18].

In two-zone combustion, combustion occurs in the following manner:

1. At the start of combustion (the spark in the SI engine, or the start of injection in the DI engine) the cylinder is divided into two zones: an unburned zone and a burned zone. All of the contents of the cylinder at that time start in the unburned zone, including residual gases from the previous cycle and EGR [17].
2. At each time step, a mixture of fuel and air is transferred from the unburned zone to the burned zone. The amount of fuel-air mixture that is transferred to the burned zone is defined by the burn rate. This burn rate is prescribed (or calculated by) the combustion model [17].
3. Once the unburned fuel and associated air has been transferred from the unburned zone to the burned zone in a given time step, a chemical equilibrium calculation is carried out for the entire "lumped" burned zone. This calculation takes into account all of the atoms of each species (C, H, O, N) present in the burned zone at that time, and obtains from these an equilibrium concentration of the 11 products of combustion species (N_2, O_2, H_2O, CO_2, CO, H_2, N, O, H, NO, OH). The equilibrium concentrations of the species depend strongly on the current burned zone temperature and to a lesser degree, the pressure [17].
4. Once the new composition of the burned zone has been obtained, the internal energy of each species is calculated. Then, the energy of the whole burned zone is obtained by summation over all of the species. Applying the principle that energy is conserved, the new unburned and burned zone temperatures and cylinder pressure are obtained [17].

In the two zone model, the following energy equations are solved separately for each time step in each zone:

Unburned Zone:

$$\frac{d(m_u e_u)}{dt} = -p\frac{dV_u}{dt} - Q_u + \left(\frac{dm_f}{dt}h_f + \frac{dm_a}{dt}h_a\right) + \frac{dm_{f,i}}{dt}h_{f,i} \quad (1)$$

where:

m_u=unburned zone mass
V_u=unburned zone volume
P=cylinder pressure
m_f=fuel mass
Q_u=unburned zone heat transfer rate
e_u=unburned zone energy
m_a=air mass
h_f=enthalpy of fuel mass
$m_{f,i}$=injected fuel mass
h_a=enthalpy of air mass
e_u=unburned zone energy
$h_{f,i}$=enthalpy of injected fuel mass

Burned Zone:

$$\frac{d(m_b e_b)}{dt} = -p\frac{dV_b}{dt} - Q_b - \left(\frac{dm_f}{dt}h_f + \frac{dm_a}{dt}h_a\right) \quad (2)$$

where subscript "b" denotes burned zone [17]. In above energy equation for the unburned zone, there are four terms on the right hand side of the equation. These terms handle pressure work, heat transfer, combustion, and addition of enthalpy from injected fuel, respectively. The third term (combustion) contains the instantaneous rate of fuel consumption or burn rate (dm_f/dt) [17].

The model used in this work is the spark ignition Wiebe model(Fig. 1), because it is the most widely used of all the spark ignition engine simulations. The spark-Ignition Wiebe model is part of two-zone models.

The spark-Ignition Wiebe model imposes the combustion rate using a Wiebe functions with three inputs. These inputs are the 50% burn point, the burn duration between the 10% and 90% burn points (Fig. 2) and a Wiebe function exponent to shift the other to inputs in respect to each other. This model also needs measured data to simulate the combustion process [19].

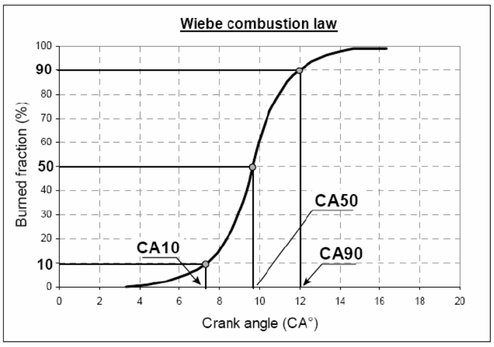

Fig. 1 Wiebe function parameters [19].

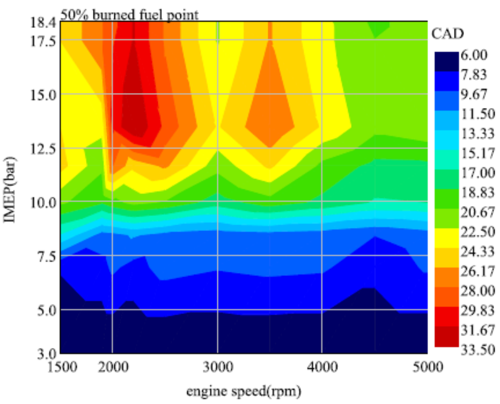

Fig. 2 50% burned fuel point map in CAD after TDC as a function of load and speed that generated by interpolating among experimental data linearly

The Wiebe equations are given as below:

$$BMC=-\ln(1-BM) \qquad (3)$$

$$BSC=-\ln(1-BS) \qquad (4)$$

$$BEC=-\ln(1-BE) \qquad (5)$$

$$WC=\left[\frac{D}{BEC^{\frac{1}{(1+E)}}-BSC^{\frac{1}{(1+E)}}}\right]^{-(1+E)} \qquad (6)$$

$$SOC=AA-\frac{(D)(BMC)^{\frac{1}{(1+E)}}}{BEC^{\frac{1}{(1+E)}}-BSC^{\frac{1}{(1+E)}}} \qquad (7)$$

$$Combustion(\theta)=(CE)\left[1-e^{-(WC)(\theta-SOC)^{(1+E)}}\right] \qquad (8)$$

where:

AA = Anchor Angle D = Duration
E = Wiebe Exponent (most of the time=2)
CE = Combustion Efficiency
BM = Burned Fuel Percentage at Anchor Angle (default = 50%)
BS = Burned Fuel Percentage at Duration Start (default = 10%)
BE = Burned Fuel Percentage at Duration End (default = 90%)
BMC= Burned Midpoint Constant BSC= Burned Start Constant BEC= Burned End Constant
WC= Wiebe Constant SOC= Start of Combustion Combustion(θ)= Fraction of Fuel Burned
θ= Instantaneous Crank Angle
[17, 20].

The spark ignition Wiebe model can be adopted to make transient simulations of the combustion process by introducing combustion maps as a function of engine load and speed in the model [19].

These maps are obtained as parameters of Wiebe function (50% burn point and 10-90% burn duration) for speeds of *1500*, *1900*, *2000*, *2200*, *2500* and *3000rpm* and loads of *2.5*, *7*, *8.5*, *10*, *11.7*, *13.5*, *17.5* and *18.*4bar exists based on experimental data, then these parameters for speeds and loads among these values are interpolated linearly and therefore the below maps obtained.

With replacing these maps in GT-POWER model, the combustion process simulated and calibrated for the transient.

3- Engine Model and Verification Process

The built model in the GT-POWER software is a one dimensional model with inputs including engine geometry parameters, initial and boundary conditions and engine working conditions. The engine geometry is derived from drawings of inlet and outlet manifolds and combustion chamber and also from geometry information of piston, crankshaft and connecting rod; engine working conditions are replaced based on laboratory conditions of Iran Khodro Powertrain Company (IPCO) Company.

Since accuracy of the transient model is highly dependent on input and output temperatures, the heat transfer is considered in inlet and outlet manifolds, cylinder wall, intercooler and turbine chamber as a function of engine speed and load. Also, a transient model is considered for injection and combustion processes such that the variations of spark timing, combustion duration, timing and duration of injection are taken into account regarding the variations of speed and load. However, performance maps of compressor and turbine are not usually available for speeds under *80000rpm* and for values under these speeds, the performance maps must be extrapolated. The reason is that at the beginning of the transient, the turbine and compressor performance lies in this region.

Engine model is calibrated for speeds of *1500*, *1900*, *2000*, *2200*, *2500* and *3000rpm* and loads of *2.5*, *7*, *8.5*, *10*, *11.7*, *13.5*, *17.5* and *18.4bar* and models of combustion, injection and heat transfer for speeds and loads among these values are interpolated linearly(Fig. 3).

To verify results of the simulated engine in the GT-POWER software, the obtained results from Iran Khodro Powertrain Company (IPCO) are used. These results are achieved from the test *749NGS4KFEF* which was done on December 2012 in test room number 6 of IPCO engine laboratory. Simulated and experimental results of IMEP720 in terms of time are demonstrated in Figs. 4 and 5 in the mentioned transient for speeds of *1500* and *2500rpm*. Before the beginning of the transient simulation, the model is run in a steady state for *2* seconds until the temperatures reach a steady state; then, the mentioned transient initiates. As it is seen in the following figures, the model behavior and laboratory information are similar.

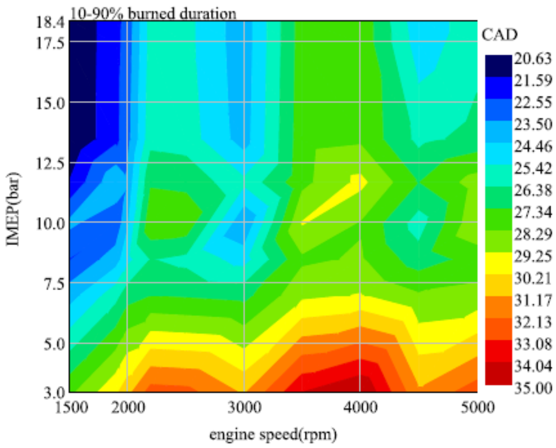

Fig. 3 10-90% burned fuel duration map in CAD as a function of load and speed that generated by interpolating among experimental data linearly

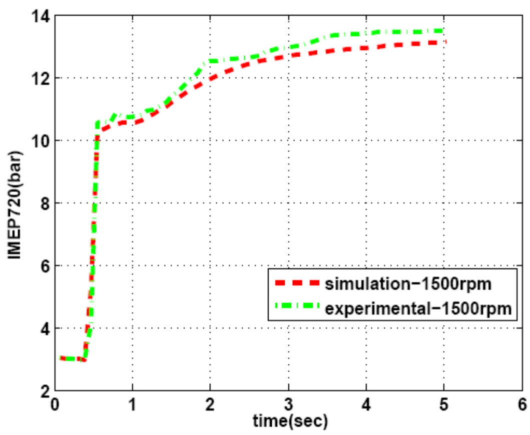

Fig. 4 simulation and experimental IMEP720 versus time in load step transient-1500 rpm

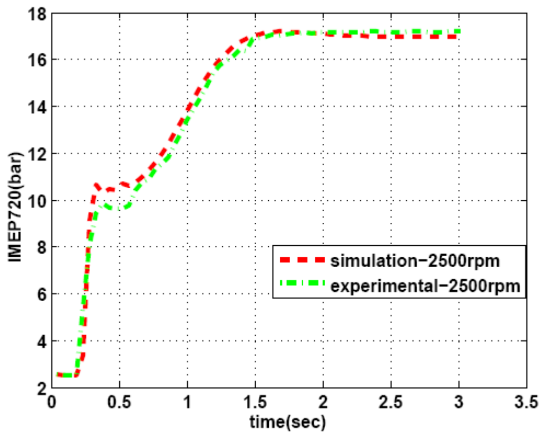

Fig. 5 simulation and experimental IMEP720 versus time in load step transient-2500 rpm

4- Research Method

The aim of this paper is to find a rapid calculation of a good strategy to use variable valve timing technology during the transient. The optimization target is to maximize the torque integral during time interval of the transient. The transient which is studied in this paper is the one in which the engine speed is constant and the load changes suddenly and rapidly from *2.6 bar* to the full load (step increase of load). In this transient, the engine mean load changes by sudden opening of throttle from the closed state to the wide open throttle (WOT) state.

As mentioned, main aim of this paper is regulation of valves timing such that the fastest transient load torque response is obtained in the constant speed. The considered torque is IMEP+PMEP which is showed with IMEP720 here and this measure equals to indicator torque during a cycle. The cost function in this optimization is the following integral:

MAX

$$\int_0^{t_{opt}} IMEP720(ivo(t), ivc(t), evo(t), e\imath \quad (9)$$

The optimization range should be selected such that a good trial and error is proposed between the time that the torque is limited by the throttle and input pressure dynamics and the time that the torque is limited by the turbocharger dynamics [22]. This time is considered *2* seconds in this paper. To decrease the number of optimization parameters, valve timing variables are optimized at specific times; linear interpolation is used to calculate variables between these time moments. Selected time moments are *t=[0 0.1 0.2 0.4 0.6 0.8 1 1.2 1.4 1.6 1.8 2]*. These time moments are after sudden opening of gas throttle. This optimization is done for two speeds of *1500* and *2500rpm*.

5- Genetic Algorithm

Math based optimization methods need to have an answer close to the optimal answer in order to initiate the problem solution. This may need some other methods to find a close answer to the optimal one. These algorithms have high rate of convergence, however, they easily get involved in local extermums. But in genetic algorithm and similar random search methods, the problem solution begins from random points in the search space and thus it is easy to anticipate the algorithm behavior. Convergence rate of these kinds of algorithms is less than math based algorithms; however, the possibility of involvement in local optimal points is also less and hence there is more hope to find the global optimal point [21].

6- Improved Genetic Algorithm

Since the fundamental of genetic algorithm is based on search, the measure of output function must be calculated for the problem inputs. This must be done for continuation of search action. In this study, the considered function is the simulated engine which is coupled with MATLAB software.

Time needed for calculation of any output load is *600* seconds. If the primary population of genetic algorithm is considered up to *200* and the problem is continued up to *100* generations, time of *100×200×600* seconds is needed for problem solution in any different speed [21].

In order to reduce time solution of the problem and also to increase the optimization accuracy, the genetic algorithm is applied as follows in this problem.

- To reduce time solution of the problem, the number of every new generation must be less than the previous one. Suppose p denotes the population number in every generation and n is a variable; then, the selection ways of *2* members from n members is:

$$1 + 2 + 3 + \cdots + (n-1) = \frac{\mathrm{n(n-1)}}{2} \tag{10}$$

since it is needed to reduce the number of new generation, it is supposed that $0.7p$ participate in the new generation. If N_p is the number of new generation, then:

$$N_\mathrm{p} = 0.7p \tag{11}$$

$$\frac{\mathrm{n(n-1)}}{2} = 0.7p \tag{12}$$

$$n^2 - n = 1.4\,p \tag{13}$$

$$n = 0.5 + \frac{1}{2}\sqrt{1 + 5.6p} \tag{14}$$

By selection of n from the above equation, N_p can be calculated in every step. For instance, if the number of first generation is *100* and it is required that*70* members participate in the second generation, n is calculated from the above equation to obtain the chosen members of the first generation so that their combination become *70* members [21].

- In order to increase the problem solution accuracy, interpolation must be done between input variables in every generation.
- Since during interpolation between two values, some good problem solutions may be destructed, 10% of the best members of the previous generation are entered directly to the new generation and are participated in this step [21].

7- Coupling GT-POWER and MATLAB for Optimization

In order to do this coupling, the simulated engine in the GT-POWER software must be defined as a block in MATLAB SIMULINK. By doing so, SIMULINK can identify the GT-POWER software. General process of problem solution in the coupled software is shown in Fig. 6. By running the written code in the MATLAB M-File, the MATLAB SIMULINK is called; during running of this program, the GT-POWER software which is coupled with SIMULINK is run and its results are imported to the M-File via SIMULINK and the rest of optimization continues. This algorithm chooses the best degree values for the valves in every specified time moment such that the area under IMEP720-t diagram is maximized in the considered time interval.

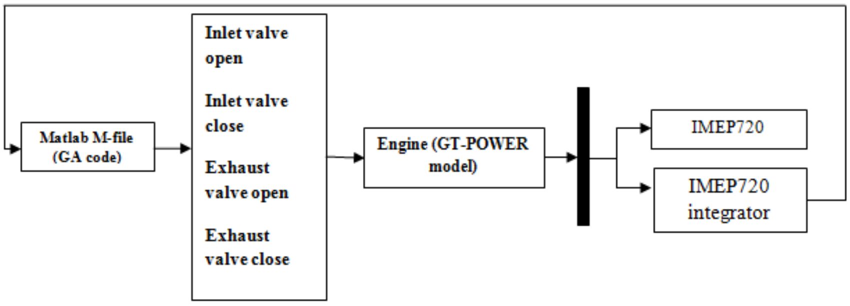

Fig. 6 coupling of GT-POWER with MATLAB M-file and MATLAM SIMULINK for optimization

8- Results

The results of IMEP720 are seen in terms of time in Figs. 7 and 8 for speeds of *1500* and *2500 rpm* for the based and optimized models. In these diagrams, the transient begins from *0.4* second for speed of *1500 rpm* and from *0.2* seconds for speed of *2500 rpm* and continues for *2* seconds.

The primary and optimized values of cost function, namely the IMEP720 integral and optimization percent of the cost function are presented in Table. 2.

Table 2: The initial and optimized values and optimization percent of the IMEP720 integral

	1500	2500
Initial IMEP720 integral	20.74	27.65
Optimized IMEP720 integral	23.12	29.91
Optimization percent	11.48	8.17

The optimized values for valves degrees are demonstrated versus time in Figs. 9-12. In these diagrams, the time zero is equal to beginning of the transient or beginning of the throttle opening. Also, at the beginning of the transient (time zero), the values of valves degrees are those of the based engine. It can be seen from diagrams of inlet and exhaust valves opening that at the most moments, the curve of speed *2500 rpm* lies under the curve of speed *1500 rpm* which means sooner opening of the inlet and exhaust valves for speed *2500 rpm*. Moreover, it is clear from diagrams of the inlet and exhaust valves closing that at all moments, the curve of speed *2500 rpm* lies over the curve of speed*1500 rpm* which means later closing of the inlet and exhaust valves for speed *2500 rpm*. The reason is the time lack compensation due to speed increase (*2500 rpm* in comparison to *1500 rpm*) and hence better filling of the cylinder.

As it can be seen in Figs. 9 and 10, the exhaust valve closes so near to the top dead center at the beginning of the transient and with continuation of the process, it closes later and at the end of the transient, the angle becomes constant. Since at the beginning of the transient, the engine load is low and there is no need to high power, the valve closes sooner to reduce the overlap value in order to prevent fresh air exit from the outlet valve port. With increase of the engine load, this valve closes later for the burnt gases to leave the cylinder properly; colliding of these gases to the turbine increases its velocity and hence increases the engine generated torque. At the end of the transient and with closing of engine load to its full load, the opening angle of this valve becomes constant, approximately.

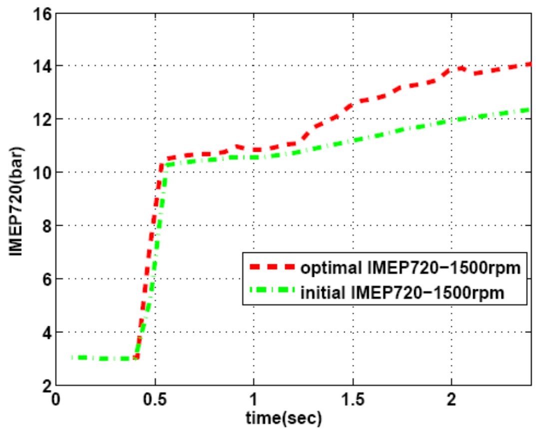

Fig. 7 initial and optimal IMEP720 in load step transient-1500rpm

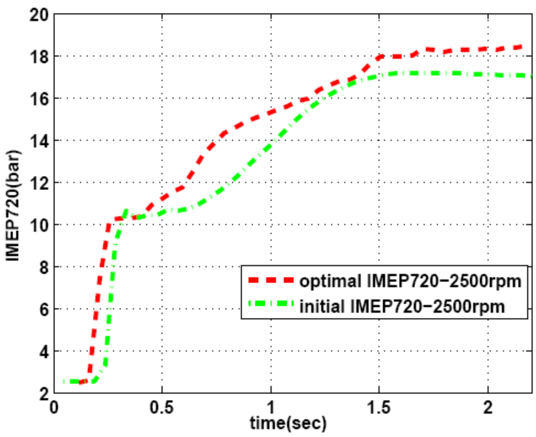

Fig. 8 initial and optimal IMEP720 in load step transient-2500rpm

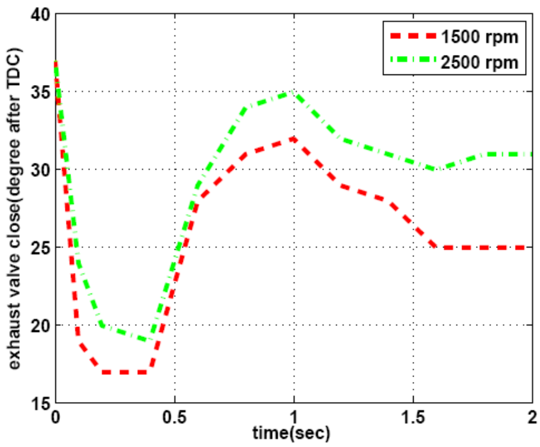

Fig. 9 optimal exhaust valve close angle in transient duration

As it can be observed from Fig. 10, the exhaust valve opens very close to the bottom dead center in the beginning of the transient; with process continuation, the valve opens sooner and at the end of the transient, the angle becomes constant. Since at the beginning of the transient, the combustion process velocity is low, due to low engine load and hence low flow disturbances of air-fuel mixture, the outlet valve opens later for the combustion process to be properly completed and the air-fuel mixture energy be released. With continuation of the transient and with increase of the engine load, this valve opens sooner for the combustion products to leave well the cylinder and let the fresh air to be replaced in the next cycle. At the end of the transient and with closing of engine load to its full load, the opening angle of this valve becomes constant, approximately. As it is clear from Fig. 11, the inlet valve closes very near to the bottom dead center at the beginning of the transient and with process continuation, the valve closes later and at the end of the transient, this angle becomes constant. Since at the beginning of the transient, the engine load is low and there is no need to high power, the valve closes sooner to prevent fresh air exit from the cylinder at the compression stage beginning. With continuation of the transient and with increase of the engine load and the engine need to more air, this valve closes later for the cylinder to be filled well. At the end of the transient and with closing of engine load to its full load, the opening angle of this valve becomes constant, approximately.

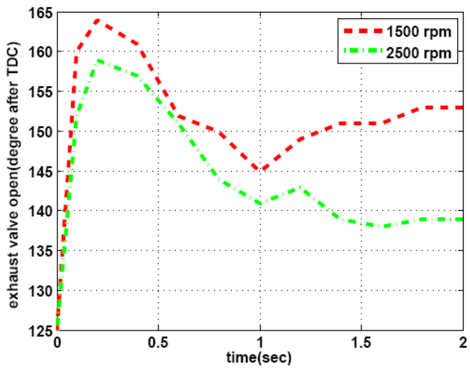

Fig. 10 optimal exhaust valve open angle in transient duration

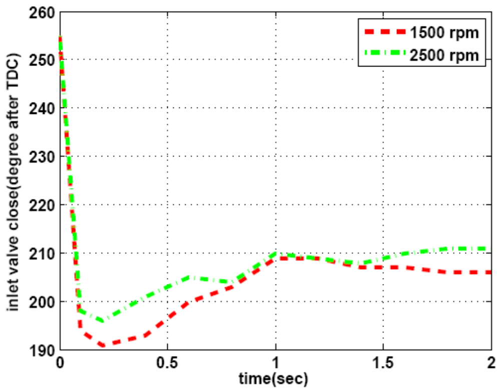

Fig. 11 optimal inlet valve close angle in transient duration

As it is seen from Fig. 12, the inlet valve opens very close to the top dead center at the beginning of the transient and with process continuation and with increase of the engine load and engine's need to more air, the valve opens sooner. At the end of the transient and with closing of the engine load to its full load, the opening angle of this valve becomes constant, approximately.

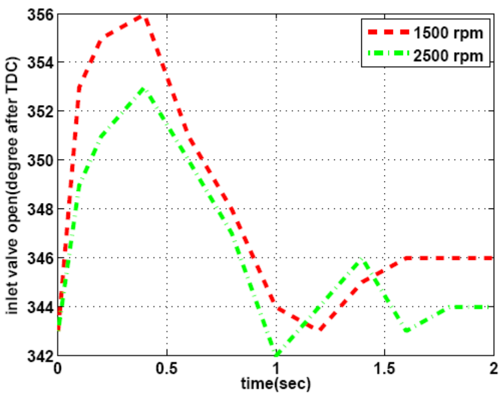

Fig. 12 optimal inlet valve open angle in transient duration

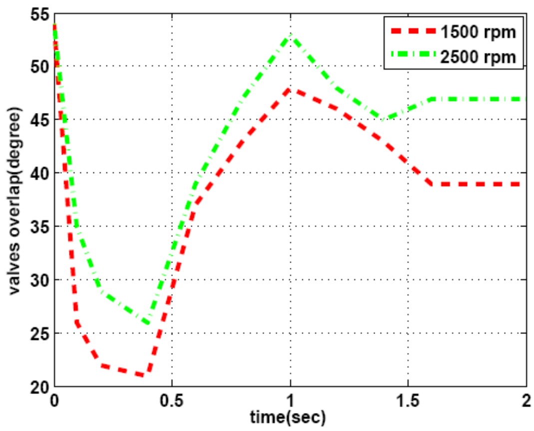

Fig. 13 optimal overlap of valves in transient duration

In Fig. 13, the optimal overlap of valves is plotted versus time during the transient. The time during which the inlet and exhaust valves are open simultaneously is called the valves overlap. As it is seen from the figure, since the engine load is low at the beginning of the transient, the overlap decreases and with continuation of the transient, the produced engine torque increases which leads to overlap increase that enhances the measure of entering air to the engine; at the end of the transient, the overlap is constant, approximately. This can be justified as follows: at the beginning of the transient, the ratio of engine overall pressure is not proper for sweeping output gases and hence with large overlap, large amount of gases remain in the cylinder which decreases the engine produced torque. Therefore, less overlap will be optimal at the beginning of the transient. With continuation of the transient, with overlap increase and hence output gas remaining in the cylinder, the energy of output gases increases which causes increase of the turbine velocity and the produced engine torque.

9- Conclusion

In this paper, a dynamic optimization is done by the improved genetic algorithm for valves timing in a turbocharged SI engine in a load step transient process. Dynamic optimization refers to the best selection of valves timing during time intervals of the transient such that the IMEP720 integral is maximized in any time interval. The optimization is done for two speeds of *1500* and *2500 rpm*. With this optimization, the IMEP720 integral is optimized about *11.48%* for speed *1500 rpm* and about *8.17%* for speed*2500 rpm*. Also, in order to achieve optimal values at the beginning of the transient, the inlet and outlet valves must be opened later and closed sooner. With continuation of the transient and with increase of the engine produced torque, inlet and outlet valves are opened sooner and closed later to increase properly the volumetric efficiency. At the end of transient and by closing of engine load to its full load, the valves degrees become constant, approximately

References

[1] Kakaee, A., & Keshavarz, M. (2012). Comparison the sensitivity analysis and conjugate gradient algorithms for optimization of opening and closing angles of valves to reduce fuel consumption in XU7/L3 engine.

[2] Shayler, P. J., & Alger, L. (2007). *Experimental investigations of intake and exhaust valve timing effects on charge dilution by residuals, fuel consumption and emissions at part load* (No. 2007-01-0478). SAE Technical Paper.

[3] NAGAO, F., NishiwaKI, K., & YOKOYAMA, F. (1969). Relation between Inlet Valve Closing Angle and Volumetric Efficiency of a Four-Stroke Engine. *Bulletin of JSME*, *12*(52), 894-901.

[4] Asmus, T. W. (1982). Valve events and engine operation. *SAE transactions*, 2520-2533.

[5] Li, L., Tao, J., Wang, Y., Su, Y., & Xiao, M. (2001). Effects of intake valve closing timing on gasoline engine performance and emissions. *SAE Transactions*, 2270-2276.

[6] Tuttle, J. H. (1982). Controlling engine load by means of early intake-valve closing. *SAE transactions*, 1648-1662.

[7] Mianzo, L., & Peng.H. (2000) Modeling and Control of Variable Valve Engine, Variable Valve Actuation. *American control conference Chicago*.

[8] Leroy, T., Chauvin, J., & Petit, N. (2009). Motion planning for experimental air path control of a variable-valve-timing spark ignition engine. *Control engineering practice*, *17*(12), 1432-1439.

[9] Wu, B., Prucka., R.G., Filipi, Z.S., Kramer, D.M., & Ohl, G.L. (2005). Cam-Phasing Optimization Using Artificial Neural Network as Surrogate Models-Maximizing Torque Output. *SAE paper, 01*, 3757. 2005

[10] Fontana, G., & Galloni, E. (2009). Variable valve timing for fuel economy improvement in a small spark-ignition engine. *Applied Energy*, *86*(1), 96-105.

[11] Ericsson, G., Angstrom, H. E., & Westin, F. (2010). Optimizing the transient of an SI-engine equipped with variable cam timing and variable turbine. *SAE International Journal of Engines*, *3*(1), 903-915.

[12] Bozza, F., Gimelli, A., Strazzullo, L., Torella, E., & Cascone, C. (2007). *Steady-state and transient operation simulation of a "downsized" turbocharged SI engine* (No. 2007-01-0381). SAE Technical Paper.

[13] Lefebvre, A., & Guilain, S. (2005). *Modelling and measurement of the transient response of a turbocharged SI engine* (No. 2005-01-0691). SAE Technical Paper.

[14] Kleeberg, H., Tomazic, D., Lang, O., & Habermann, K. (2006). *Future potential and development methods for high output turbocharged direct injected gasoline*

engines (No. 2006-01-0046). SAE Technical Paper.

[15] Eriksson, L., Lindell, T., Leufven, O., & Thomasson, A. (2012). Scalable component-based modeling for optimizing engines with supercharging, E-boost and turbocompound concepts. *SAE International Journal of Engines*, *5*(2), 579-595.

[16] Xu, X., Liu, J., Wang, Y., Zhao, Z., Xia, X., & Fu, J. (2011, April). A research of turbocharged gasoline transient response. In *2011 International Conference on Electric Information and Control Engineering* (pp. 4719-4722). IEEE.

[17] GT-Power v7.3 user's manual

[18] Heywood, J. B. (2018). *Internal combustion engine fundamentals*. McGraw-Hill Education.

[19] Bodin-Ek, E. (2008). Kalibrering av en transient GT-Power modell av en SI PFI turbo motor.

[20] Wiebe, I. (1964). Halbempirische formel durch die verbrennungsgeschwindigkeit. *Kraftstoffaufbereitung und Verbrennung bei Dieselmotoren, Spring-Verlag*.

[21] AKakee, A. H., Sharifipour, S., Mashadi, B., Keshavarz, M., & Paykani, A. (2015). Optimization of spark timing and air-fuel ratio of an SI engine with variable valve timing using genetic algorithm and steepest descend method. *UPB Sci Bull Ser D Mech Eng*, *77*(1), 61-76.

[22] Martensson, J., & Flardh, O. (2010). *Modeling the effect of variable cam phasing on volumetric efficiency, scavenging and torque generation* (No. 2010-01-1190). SAE Technical Paper.