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Research of the Intake Valve Deactivation on Engine Performance

S. Binbin, G. Song*, L. Bo

School of Transportation and Vehicle Engineering, Shandong University of Technology, Zibo, China

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ABSTRACT

In this paper, the effect of the Intake Valve Deactivation (IVDA) on engine performance is investigated in detail. Based on an optimization platform with Genetic Algorithm (GA) and engine thermodynamic model, the characteristics of the engine volumetric efficiency and pumping loss were studied under the cam-drive, Single Intake Valve (SIV) and Dual Intake Valves (DIV) operating modes, and the effect of the IVDA on the engine fuel economy was revealed with taking the power consumption of the Electromagnetic Actuated Valvetrain (EAVT) system into consideration. Then, switch rules for the SIV and DIV mode was proposed, and the switching boundary conditions between them were confirmed. Finally, the optimal intake valve close timings for the EAVT system were obtained. Results show that, under the low speed conditions, the SIV mode has little influence on the engine volumetric efficiency, while within the high speed conditions the effect of the IVDA on the volumetric efficiency is significant; compared with the traditional cam-drive valvetrain, the pumping loss of the EAVT engine decreases significantly and shows unique characteristics due to the use of the EIVC strategy; with the use of the IVDA scheme, the energy consumption of the EAVT system reduces, but the engine pumping loss increases in the meantime, both balance their influence on the engine fuel economy. In general, the IVDA scheme is preferred if the engine volumetric efficiency can be ensured, otherwise, the DIV mode takes priority over the SIV mode to maintain the engine power performance.

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NOMENCLATURE

EAVT	Electromagnetic Actuated Valvetrain	EIVC	Early Intake Valve Closing
SIV	Single Intake Valve	BSFC	Brake Specific Fuel Consumption
DIV	Dual Intake Valves	GA	Genetic Algorithm
IVDA	Intake Valve Deactivation	PMEP	Pumping Mean Effective Pressure

1. INTRODUCTION

Variable valvetrain offers great flexibility and potential for the improvement of engine economy, power and emissions [1-4]. This is especially true for the camless valvetrain [5-8], since each individual valve can be actuated independently according to engine operating conditions. Particularly, for the engines with dual intake valves per cylinder, the reasonable intake valve operating mode should achieve not only for the improvement of the engine volumetric efficiency under the full load conditions, but also for the promotion of the thermodynamic cycle efficiency within the partial load operations [9-11].

Based on a controlled auto-ignition engine, Kim et al. [12] revealed the positive effect of intake valve timing and lift on the flow and mixing characteristics of the mixture. Moore et al. [13] proved that, with the use of the IVDA under the low load operations, the in-cylinder charge motion and combustion stability can be improved, while the engine volumetric efficiency is nearly not affected, especially within the low speed conditions. Based on a 4-cylinder gasoline engine with variable valve actuation, Chavithra et al. [14] confirmed that the IVDA and cylinder deactivation could reduce the fuel consumption substantially in comparison to the same engine with the fixed valve timings. Tian et al. [15], using computational fluid dynamics method, proved that the in-cylinder charge movement intensity, kinetic

*Corresponding Author's Email: *gaos546@126.com* (G. Song)

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energy and turbulent kinetic energy can be improved with the use of the IVDA.

Generally, as for the effect of the IVDA on engine performance, current studies mainly concentrate on the in-cylinder charge motion and combustion stability, while neglecting the characteristics of the volumetric efficiency and pumping loss under various intake valve operating modes. Moreover, the influence of the EAVT's energy consumption on engine fuel economy is still unknown. Based on a self-developed EAVT system [16], this paper therefore presents systematic studies on the engine volumetric efficiency, pumping loss and fuel economy for different intake valve operating modes.

2. MODELLING OF THE ENGINE WITH EAVT

2. 1. Prototype Engine With EAVT As shown in Figure 1, the prototype engine with the EAVT system has been developed [16, 17]. In the EAVT, a laser sensor is used to provide the feedback signal of the valve position for the controller; a moving-coil linear actuator is deposed within the air-gap magnetic field generated by the Halbach-array permanent magnet. By controlling the direct current in the linear actuator, Lorentz force is generated to actuate the valve.

2. 2. Modelling of Engine Working Process As it is unpractical to describe every sub-model of the EAVT engine, three of the key models relating to the IVDA are expressed as follows:

(1) Modelling of the camless valve movement curve

Different from the traditional cam-drive valve lift generated from the cam curve, the valve movement curves of the EAVT engine are mainly determined by the transition time and valve timings of the EAVT, and both change with engine speed and load. As shown in Figure 2, the measured transition time of the EAVT is about 4 ms, which can be translated into crank angle.

$$\varphi_v = {}^{6Nt}\!/_{1000} \tag{1}$$

where φ_v is the crank angle, N means the engine speed, t represents the transition time.

In addition, if the engine operates under the high speed or low-load conditions, due to the constraint of the transition time, the EAVT is incapable of achieving the maximum design lift (8 mm). Consequently, it is necessary to correct the actual valve lift based on the transition time and valve timings as follows:

i) when $V_{ct} - V_{ot} \geq 2\varphi_v$:

$$\begin{aligned} V_{ohs} &= V_{ot} + \varphi_v \\ V_{ohe} &= V_{ct} - \varphi_v \\ h_{\max} &= 8mm \end{aligned} \tag{2}$$

ii) when $V_{ct} - V_{ot} < 2\varphi_v$:

$$\begin{aligned} V_{ohs} &= V_{ohe} = V_{ot}\left(V_{ct} - 0.5V_{ot}\right) + V_{ot} \\ h_{\max} &= 8 - {}^{8\left(V_{ot} + \varphi_v - V_{ohs}\right)}\!/_{\varphi_v} \end{aligned} \tag{3}$$

where h_{max} is the maximum design lift of the EAVT system, V_{ot} and V_{ct} are the valve open and close timing, respectively. V_{ohs} is the time when valve starts to hold open and V_{ohe} is the valve close time

Given the above analysis, V_{ot} and V_{ct} are the key parameters for determining the valve movement curves. Consequently, both of them are designed as optimization variables as shown in Figure 3. However, one thing should be noted that, for the 4-cylinder engine with the fixed firing order (1-3-4-2), it is difficult to determine the ranges of the variables for the four cylinders. For example, the intake top dead point of the first cylinder is 360 °CA, if the optimization variable (intake valve close timing) is set within [500 °CA, 580 °CA], the other three cylinders will not function properly. Consequently, universal variables are proposed as follow:

$$\begin{aligned} V_{ot}(i) &= V_{ob} + V_o(i) \\ V_{ct}(i) &= V_{cb} + V_c(i) \end{aligned} \tag{4}$$

where V_{ob} and V_{cb} are the given valve open and close timing, and both change with the cylinder number.

For example, for cylinder 1, 2, 3 and 4, the V_{ob} of the intake valve are 360°CA, 180°CA, 540°CA and 0°CA, respectively. $V_o(i)$ and $V_c(i)$ are the universal optimization variables designed to represent valve open and close timing, respectively.

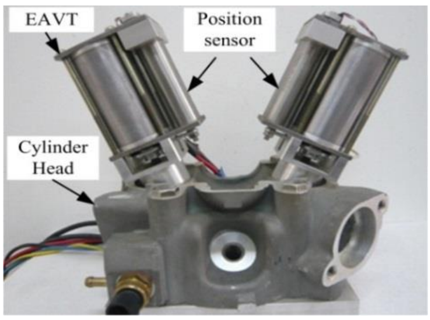


Figure 1. The prototype engine with the EAVT

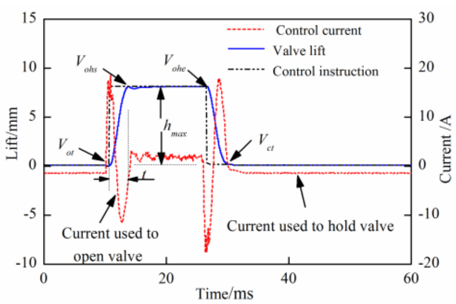


Figure 2. The measured valve lift of the EAVT

For the intake valve, the ranges of the both are set within [-40°CA, 40°CA] and [-10°CA, 10°CA]. i is the number i GA population.

(2) Fractal combustion model

As the IVDA scheme will influence the turbulence level and macroscopic charge motion of the in-cylinder charge, and both will affect the engine combustion, fractal combustion model is suggested in this paper to predict the rate of heat release [17].

$$dm_b/dt = \rho_u A_T S_L \tag{5}$$

where dm_b/dt is the mass burning rate, ρ_u is the density of the unburned gas-fuel and S_L means the laminar burning speed relating to residual gas coefficient [17]. A_T is the flame front area propagating within the turbulent flow field, which is affected by the turbulence density of the in-cylinder charge.

$$A_T = \left(I_{max}/I_{min} \right)^{D_3-2} A_L \tag{6}$$

where A_L is the laminar flame area confirmed by the geometric parameters of the cylinder. I_{min} and I_{max} are the minimum and maximum fluctuation factor relating to the Reynolds number as follow:

$$I_{max}/I_{min} = Re^{3/4} = \left(u' c_l H / \nu_\mu \right)^{3/4} \tag{7}$$

where R_e is the Reynolds number, u', c_l and ν_μ are the turbulent dynamic energy density, turbulent dissipation coefficient and kinematic viscosity of the unburned mixture determined by the K-k turbulent model [18]. H is the instantaneous clearance height. Furthermore, D_3 shown in Equation (6) is the fractal dimension of the flame and reflects the distortion of the flame as follows:

$$\begin{aligned} D_3 &= \frac{D_{3,\max} u' + 2.05 S_L}{u' + S_L} \\ D_{3,\max} &= 2.05(1-w_l) + 2.35 w_l \\ w_l &= 1 - exp(-nr_f / n_{ref} r_{f\_ref}) \end{aligned} \tag{8}$$

where r_{f_ref} is the reference radius of the spark, n_{ref} represents the reference speed, n is the actual speed of the engine, r_f is the dynamic radius of the spark.

(3) BSFC model

For the EAVT engine, the power loss model of the valvetrain system should be revised due to the use of the EAVT system instead of the original camshaft system. When the EAVT engine operates under the SIV mode, the corrected BSFC is as follow:

$$b_{es} = \frac{B}{P_i - (P_m - P_{cam}) + 4(P_o - P_c) \times 10^{-3}} \tag{9}$$

For the DIV operating mode, the corrected BSFC is:

$$b_{ed} = \frac{B}{P_i - (P_m - P_{cam}) + 8P_o \times 10^{-3}} \tag{10}$$

where b_{es} and b_{ed} represent the BSFC of EAVT engine operating under the SIV and DIV mode. B is the fuel consumption, P_i is the indicated power, P_m is the mechanical power losses, P_o means the power loss of each working EAVT. P_c represents the power loss of each deactivated EAVT, which also needs current to keep valve closed as shown in Figure 2. Both P_o and P_c can be measured by the test bench shown in Figure 12. P_{cam} means the power loss of camshaft valvetrain system, which is estimated by the empirical formula [19].

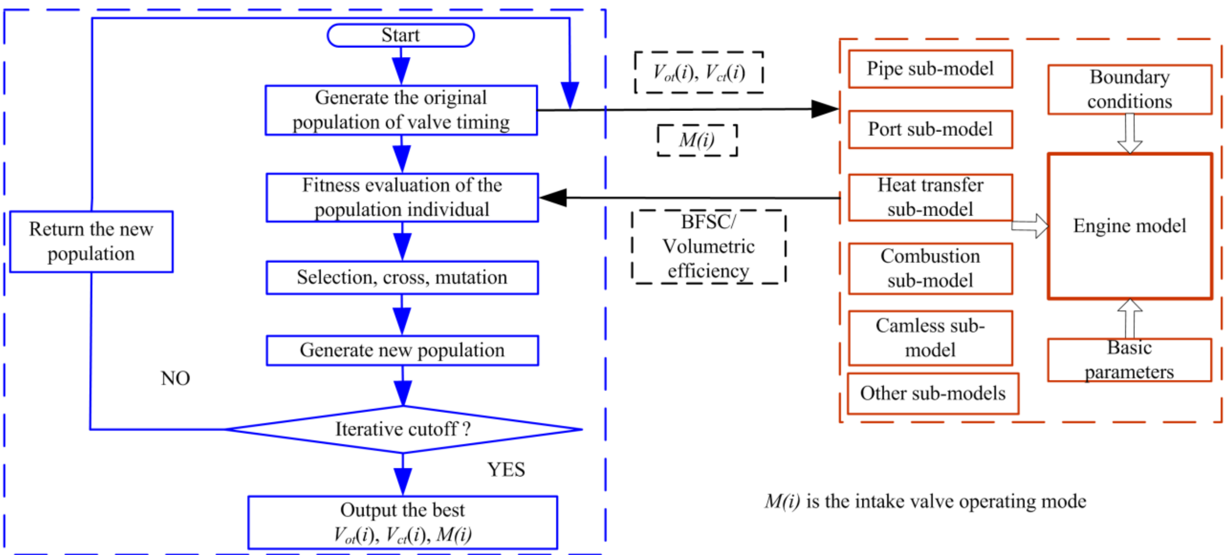


Figure 3. The united optimization platform with GA and engine model

$$P_{cam} = c_{vb}\left(\frac{Nn_b}{B_c^{2}S_c n_c}\right) + c_{vo} + c_{vh}\left(\frac{L_v^{1.5}N^{0.5}n_v}{B_c S_c n_c}\right) + c_{vm}\left(1+\frac{10^3}{N}\right)\frac{L_v n_v}{S_c n_c} + c_{vr}\left(\frac{Nn_v}{S_c n_c}\right) \qquad (11)$$

where c_{vb}, c_{vo}, c_{vh}, c_{vm} and c_{vr} are the empirical constants. B_c and S_c are cylinder bore and stroke. n_c, n_b and n_v represent the quantity of cylinders, bearings and valves. L_v means the maximal valve lift of the cam drive valve.

(4) GA optimization model

To obtain the optimum valve timings and operating modes of the EAVT under various operating conditions, a united optimization platform including GA and engine thermodynamic model is established as shown in Figure 3. According to the feedbacks of the BSFC or volumetric efficiency from the engine model, the GA module optimizes the factors of valve timings and operating modes, to achieve the optimal volumetric efficiency under the full load conditions and the least BSFC under the partial load conditions. Base on the optimization model, the optimal intake valve timing of the first cylinder is obtained as shown in Figure 4.

3. EFFECT OF THE IVDA ON VOLUMETRIC EFFICIENCY

Based on the united optimization platform, the optimal volumetric efficiency under the full load conditions are obtained as shown in Figure 5.

For the low speed conditions (N<2500r/min), the difference of the optimum volumetric efficiency between the SID and DIV mode is rather small, while within the middle to high speed conditions, the difference between them increases with the engine speed significantly.

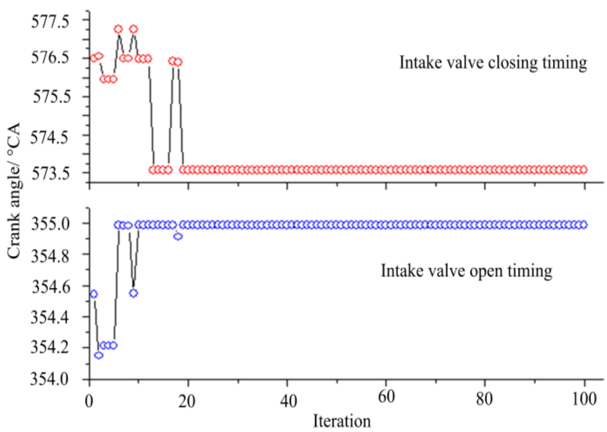


Figure 4. Iterative process of the intake valve timing

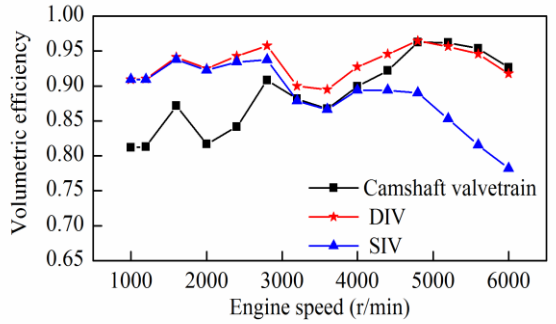
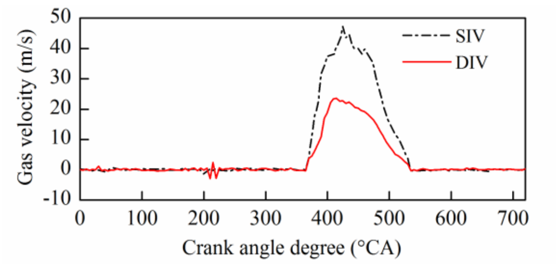


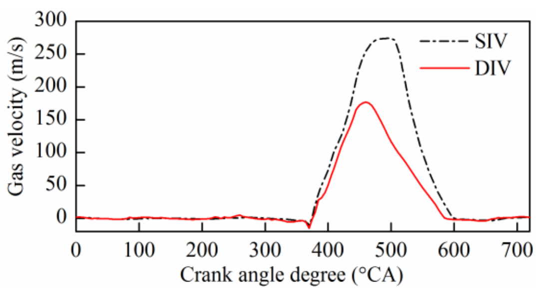
Figure 5. Volumetric efficiency under various modes

The comparisons of the gas velocity in the intake manifold between the SIV and DIV mode are presented in Figure 6. With the use of the IVDA, the gas velocity during the intake process increases owing to the decrease of the effective flow area at the intake valve seat. Furthermore, as shown in Figure 6(a), under the low speed conditions, the reduction of the volumetric efficiency caused by the smaller effective flow area can be covered by the enhancement of the gas velocity, but this rule cannot be achieved within the high speed conditions, as shown in Figure 6(b).

In addition, the intake-valve close timing of the SIV mode delays to take full advantages of the inflow inertia under high speed conditions. In comparison to the camshaft valvetrain with the fixed valve timing, owing to the flexibly controllable valve events, the EAVT system can make full use of the airflow resonance effect and the inflow inertia during the early and terminal stage of the intake stroke.



(a) 1000r/min



(b) 5600r/min

Figure 6. Gas velocity under different engine speeds

As shown in Figure 7(a), when the engine speed is low, with the help of the EAVT system, the reverse flow occurring in the early and terminal stage of the intake stroke are eliminated.

However, limited by the transition time of the EAVT system, the superiority of the EAVT system (preventing reverse flow) weakens gradually as the engine speed increases. Specifically, the smallest transition time of the EAVT is about 3.8 ms, and the shortest period of the intake valve opening is about 118.56 °CA under the speed of 5200 r/min. Consequently, to prevent the inverse flow in the terminal stage of the intake stroke, the EAVT controller have to shift the intake valve from the opening actuation to the closing operation immediately, which restricts the actual maximal lift of the EAVT, and further influences the volumetric efficiency.

In addition, as shown in Figure 7(b) and (c), the inverse flow can be slightly reduced by delaying the intake valve open timing. But on the other hand, this method also shortens the intake valve open duration and the maximum lift, and both will cause greater negative effect on the volumetric efficiency.

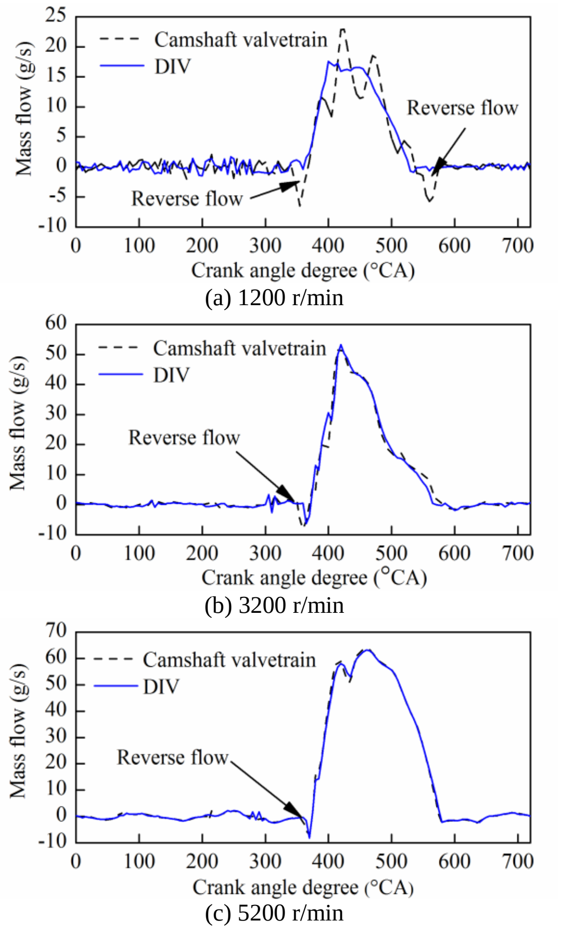


Figure 7. Mass flow under various engine operations

4. EFFECT OF THE IVDA ON PUMPING LOSS

As shown in Figure 8 and Figure 10, in contrast to the traditional engine, where pumping loss is mainly caused by the throttle, the pumping loss of the EAVT engine decreases significantly and presents unique features due to the use of the EIVC. When the camless engine works under the low speed conditions (N<2400 r/min), within the high load region, the pumping loss decreases as the engine load increases. But within the low to medium load conditions, the pumping loss presents a contrary changing law as shown in Figure 8. Moreover, the IVDA scheme has a slight influence on the pumping loss.

To reveal the reasons that cause the characteristics in Figure 8, the Pressure-Volume (P-V) diagrams of the EAVT engine at the speed of 1200 r/min are obtained as shown in Figure 9. First of all, as shown in Figure 9 (a), when the load of the EAVT engine is high, due to the later intake valve close angle after the bottom dead centre of the intake stroke, a special phenomenon that the pressure of the inlet gas is higher than the exhaust gas occurs in the region B. Consequently, under the load condition, the pumping loss of the EAVT engine is the area difference between region A and B.

Furthermore, as shown in Figure 9 (b), if engine load decreases to the middle conditions, the intake valve close angle needs to be advanced before the bottom dead centre of the intake stroke to reduce the inlet mixture, which not only leads to the expansion of the inlet mixture during the later stage of the intake stroke, but also causes the leftward shift of the compression curve. Consequently, the area difference between region A and B (the pumping loss) increases as the area of region B reduces. Finally, as for the low load conditions, due to the earlier intake valve closed angle, the area of region B drops to zero. But as the area of region A is small, the overall pumping loss is smaller than other conditions.

As shown in Figure 10, different from the variation laws of the pumping loss shown in Figure 8, for the middle and high engine speed conditions ($N \geq$ 2400 r/min), the pumping loss is directly proportional to the engine load, which is also opposite of the cam-drive valvetrain system.

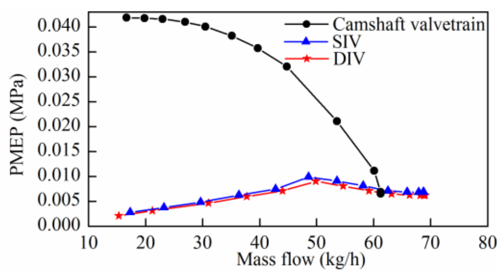
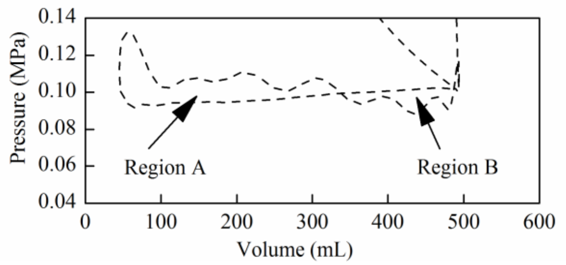
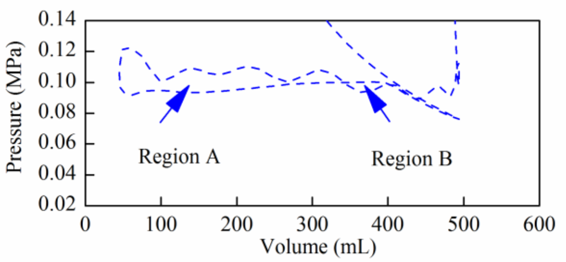


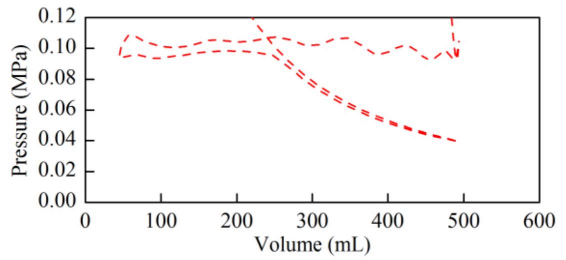
Figure 8. PMEP under the speed of 1200 r/min



(a) High load



(b) Middle load



(c) Low load

Figure 9. P-V diagram at the speed of 1200 r/min

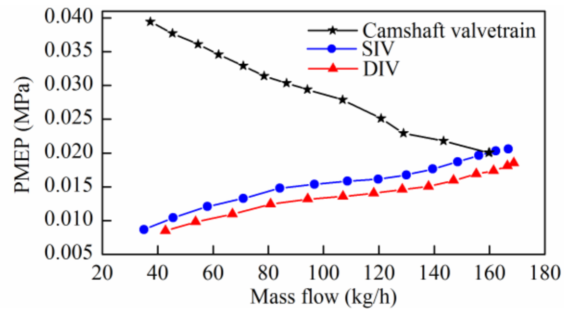
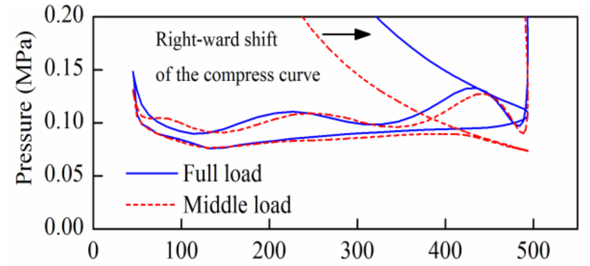
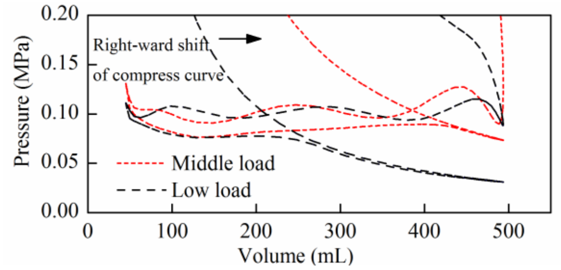


Figure 10. Pumping loss under the speed of 2800 r/min

To explain the reasons, the P-V diagram of the EAVT engine at the speed of 2800 r/min is presented as shown in Figure 11. In comparison to the low speed conditions, as the gas pressures of the combustion and exhaust stroke increase, the phenomenon that the pressure of the inlet gas is higher than exhaust gas no longer exists. Furthermore, as the engine load enlarges, the intake valve close angle needs to be delayed, and thus the pumping loss increases with the rightward shift of the compression curve.



(a) Full and middle load conditions



(b) Middle and low load conditions

Figure 11. P-V diagram under the speed of 2800 r/min

5. EFFECT OF THE IVDA ON ENGINE ECONOMY

The loss of the EAVT can be divided into three basic categories, which are the copper, core and mechanical loss. The copper loss is the loss produced by the electrical current in the liner actuator and generates heat in the moving-coil. The core loss consists of the hysteresis and eddy current loss as a result of the changing magnetization in the core material. The mechanical loss is defined as the loss associated with mechanical effect, and mainly consists of the friction and windage loss.

As shown in Figure 2, to obtain a shorter transition time, larger current is always adopted to accelerate the valve during the first half of the stroke, while a larger reverse current is applied to decelerate the valve to achieve low seating velocity during the latter half of the stroke. On the contrary, the current used to keep the valve open or close is relatively small. In general, most of the energy is consumed during the transition process. Based on the measured current in the liner actuator, the input electrical power P_{in} can be obtained as follows:

$$P_{in} = UI \tag{12}$$

As the EAVT engine is naturally aspirated with throttle-free load control mode, the back pressure around the intake valve seat can be viewed as equal to an atmospheric pressure approximately. Consequently, the test platform of the EAVT system shown in Figure 12 is designed to measure the power consumption of the EAVT system. Figure 13 presents the experimental energy and power consumption of the EAVT used to actuate the intake valve.

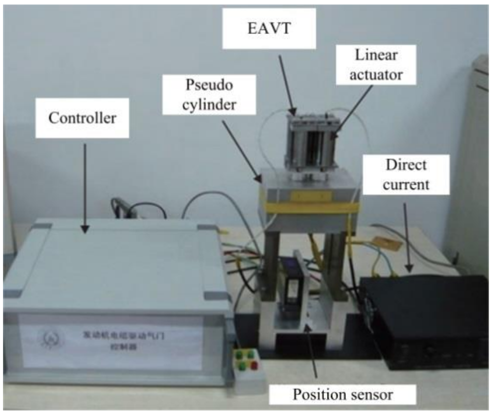


Figure 12. Power consumption test of the EAVT system

As mentioned above, transition time is the key factor that influences the energy consumption of the EAVT. Since a fixed transition time (4 ms) is used in the EAVT, the difference of the energy consumption under various speed conditions is relatively small. Furthermore, for the low speed operations, as the optimized event duration and engine speed have an inverse relationship, the energy consumption decreases as engine speed increases. For the high speed operations, as larger current is used to shift the EAVT from the opening action to the closing movement, the energy consumption increases.

Based on the revised BSFC model discussed above, the BSFC of the EAVT engine is obtained as shown in Figure 14. Compared with the cam-drive valvetrain system, owing to the throttle-free load control of the EAVT system, the pumping loss of the EAVT engine reduces significantly, and thus the engine fuel economy is improved substantially under the partial load conditions. Furthermore, although, the IVDA scheme increases the engine pumping loss, the BSFC of the SIV mode is still superior to that of the DIV mode, due to the energy consumption reduction of the EAVT system and the improvement of the in-cylinder charge motion.

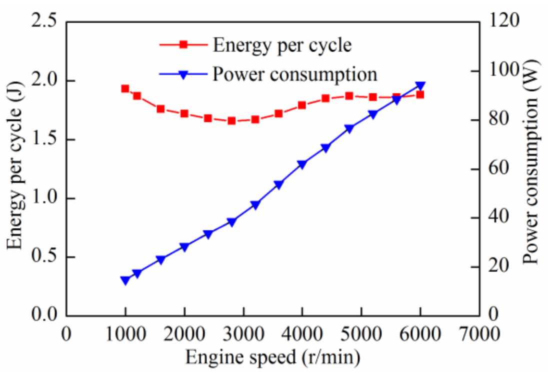


Figure 13. Energy and power consumption of the EAVT

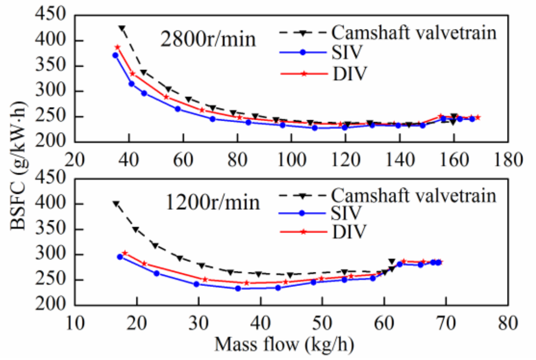


Figure 14. Corrected fuel economy

6. SWITCH SCHEDULE FOR THE IVDA

Given the above analysis, a switch rule for the IVDA is proposed. The SIV strategy takes priority over the DIV if the volumetric efficiency can be ensured; otherwise, the DIV mode is the preferred method to maintain the engine power performance. Based on the rule proposed, the optimal switch boundaries for the SIV and DIV mode are obtained as shown in Figure 15.

For the low speed conditions, where the volumetric efficiency can be ensured, the SIV mode is the preferred operating mode. But within the high speed and load conditions, the DIV operation takes priority over the SIV mode to maintain the engine power performance. Furthermore, as for the uncommon use regions, the DIV mode is preferred to facilitate system control.

Finally, the optimal intake valve close timings, as the function of engine speed and torque, are obtained as shown in Figure 16. In general, the intake valve closes timings delay as the engine speed and torque increase. Within the speed conditions from 0 to 3200 r/min, where the EAVT engine operates under the SIV mode, later intake valve closing timing is preferred to make full use of the high inflow inertia. But when the engine is higher than 3200 r/min, the inflow inertia decreases due to the use of DIV mode, which means earlier intake valve closing timing is required to avoid the reverse flow. Consequently, a local maximum is obtained in region A.

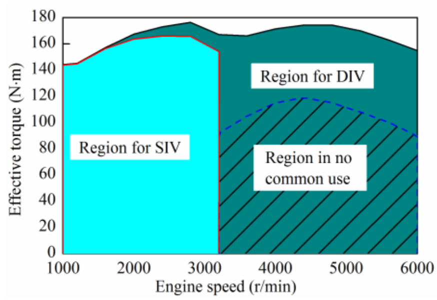


Figure 15. Operating regions for the SIV and DIV mode

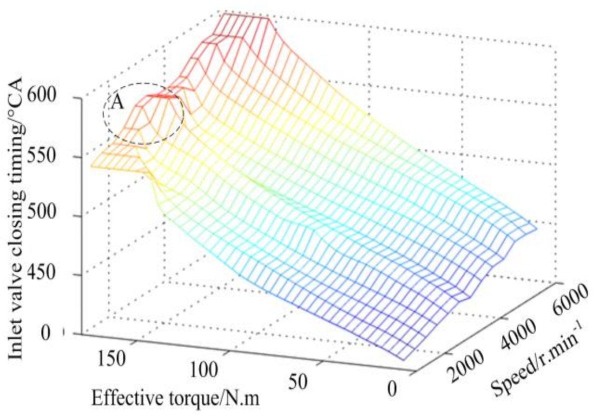


Figure 16. The optimal intake valve closing timings

7. CONCLUSIONS

In this study, an EAVT system that can achieve IVDA was introduced, and a united optimization platform with GA and engine thermodynamics model was established to study the effects of the IVDA on engine performance. Specifically, the obtained conclusions are as follows:

1. From the point of the volumetric efficiency, under the low speed conditions, for the IVDA scheme, the loss of the volumetric efficiency caused by the smaller effective flow area can be covered by the enhancement of gas velocity, but this rule is invalid within the high speed conditions.
2. As for the pumping loss, first of all, with the use of the IVDA, the pumping loss shows a slight increase in comparation to the DIV mode; secondly, the pumping loss can be significantly reduced owing to the throttle-free load control; moreover, the EAVT engine shows unique characteristics of the pumping loss. Specifically, first of all, for the low speed conditions, the pumping loss decreases with the engine load increases within the high load conditions, while within the low to medium loads, the relationship between them presents a contrary changing trend; secondly, for the middle to high speed conditions, the pumping loss is proportional to the engine load.
3. From the point of the engine economy, the BSFC of the EAVT engine operating under the SIV mode is better than that under the DIV mode, owing to the advantages of the energy consumption reduction and the improvement of the in-cylinder charge motion.
4. On the level of the switch rules for the SIV and DIV mode, the IVDA is preferred under most of the low to medium speed conditions, where the volumetric efficiency can be ensured. However, as for the high speed and load conditions, the DIV mode takes priority over the SIV to maintain the engine power performance.

8. ACKNOWLEDGMENT

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Research of the Intake Valve Deactivation on Engine Performance

S. Binbin, G. Song, L. Bo

School of Transportation and Vehicle Engineering, Shandong University of Technology, Zibo, China

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چکیده

در این مقاله، اثر غیرفعال کردن شیر ورودی (IVDA) بر عملکرد موتور با جزئیات مورد بررسی قرار گرفته است. بر اساس پلت فرم بهینه سازی با الگوریتم ژنتیک (GA) و مدل ترمودینامیکی موتور، ویژگی های بازده حجمی موتور و اتلاف پمپ تحت حالت های رانش میله، سوپاپ مکش تنها (SIV) و مصرف دو سوپاپ (DIV) مورد بررسی قرار گرفت، و اثر IVDA بر اقتصاد سوخت موتور با در نظر گرفتن مصرف برق سیستم های الکترومغناطیسی محرک مجموعه سوپاپ (EAVT) بررسی شد. سپس، قوانین سوئیچ برای حالت SIV وDIV مطرح شد و تعویض شرایط مرزی بین آنها تایید شد. در نهایت، زمان بهینه بستن شیر ورودی برای سیستم EAVT به دست آمد. نتایج نشان می دهد که تحت شرایط سرعت کم ، حالت SIV تاثیر کمی بر روی راندمان حجمی موتور دارد، در حالی که در شرایط سرعت بالا اثر IVDA در راندمان حجمی قابل توجه است. در مقایسه با مجموعه سوپاپ رانش میله سنتی، اتلاف پمپ موتور EAVT به طور قابل توجهی کاهش می یابد و ویژگی های منحصر به فردی را با توجه به استفاده از استراتژی EIVC نشان می دهد؛ با استفاده از این طرح IVDA، مصرف انرژی سیستم EAVT کاهش می یابد، اما اتلاف موتور پمپاژ افزایش می یابد و هر دو اثر خود را روی اقتصاد سوخت موتور متعادل می کنند. به طور کلی، طرح IVDA در صورتی که راندمان حجمی موتور تضمین شود ترجیح داده می شود، در غیر این صورت، حالت DIV اولویت بیشتری نسبت به حالت SIV برای حفظ عملکرد قدرت موتوردارد.

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