



[biblio.ugent.be](http://biblio.ugent.be)

The UGent Institutional Repository is the electronic archiving and dissemination platform for all UGent research publications. Ghent University has implemented a mandate stipulating that all academic publications of UGent researchers should be deposited and archived in this repository. Except for items where current copyright restrictions apply, these papers are available in Open Access.

This item is the archived peer-reviewed author-version of:

Impact of variable valve timing on power, emissions and backfire of a bi-fuel hydrogen/gasoline engine

S. Verhelst, J. Demuynck, R. Sierens and P. Huyskens

In: International Journal of Hydrogen Energy, Volume 35, Issue 9, May 2010, Pages 4399-440

<http://dx.doi.org/10.1016/j.ijhydene.2010.02.022>

**To refer to or to cite this work, please use the citation to the published version:**

Verhelst, S., Demuynck, J., Sierens, R. & Huyskens, P. (2010). Impact of variable valve timing on power, emissions and backfire of a bi-fuel hydrogen/gasoline engine. *INTERNATIONAL JOURNAL OF HYDROGEN ENERGY* 35(9). 4399-440  
<http://dx.doi.org/10.1016/j.ijhydene.2010.02.022>.

## **Impact of variable valve timing on power, emissions and backfire of a bi-fuel hydrogen/gasoline engine**

S. Verhelst\*, J. Demuyne and R. Sierens

Department of Flow, Heat and Combustion Mechanics

Ghent University

Sint-Pietersnieuwstraat 41, B-9000 Gent, Belgium

P. Huyskens

Karel de Grote University College, Department of Applied Engineering,

Salesianenlaan 30, B-2660, Antwerp, Belgium

\*corresponding author: T +32 9 264 3306; F +32 9 264 3590; E-mail [Sebastian.Verhelst@UGent.be](mailto:Sebastian.Verhelst@UGent.be)

### **Abstract**

Hydrogen-fueled internal combustion engines are a possible solution to make transportation more ecological. Apart from difficulties in production and storage of hydrogen, there are three major bottlenecks in the operation of hydrogen powered engines: reaching a high power output, reducing NO<sub>x</sub> emissions at high loads and avoiding backfire. This paper presents an experimental study of the influence of continuously variable valve timing of the intake valves on these bottlenecks. Measurements were performed on a four-cylinder engine that can run on gasoline as well as on hydrogen. The measurements on hydrogen are compared to those on gasoline. For hydrogen, the effects of the cam phasing were investigated at wide open throttle, where load is controlled by the quality of the mixture (equivalence ratio) as well as in throttled mode, where load is defined by the quantity of mixture.

Results show that it is possible to optimize the applied control strategy by using variable valve timing as a means to increase the range of both the qualitative and quantitative load control methods, which contributes to an easier switch between both methods.

*Keywords:* Hydrogen ICE; variable valve timing; bi-fuel; emissions; backfire

## Nomenclature

### Symbols

$n$	engine speed	[rpm]
$Q_{air}$	volumetric air flow	[m <sup>3</sup> /h]
$Q_{H_2}$	volumetric hydrogen flow	[m <sup>3</sup> /h]
$V_s$	swept volume	[m <sup>3</sup> ]

### Greek Symbols

$\eta_{vol}$	volumetric efficiency	[-]
$\Phi$	fuel to air equivalence ratio ( $\Phi = 1/\lambda$ )	[-]
$\lambda$	air to fuel equivalence ratio	[-]

### Abbreviations

BDC	bottom dead center
BTDC	before top dead center
ca	crank angle
CVVT	continuously variable valve timing
EGR	exhaust gas recirculation
H <sub>2</sub> ICE	hydrogen powered internal combustion engine
ICE	internal combustion engine
IT	ignition timing
IVO	intake valve opening time
LNT	lean NO <sub>x</sub> trap
MAF	mass air flow
MBT	minimum spark advance for best torque
NO <sub>x</sub>	nitrogen oxides
PFI	port fuel injection
TDC	top dead center
TP	throttle position
TWC	three way catalyst
VOP	valve overlap period
WOT	wide open throttle

# 1 Introduction

The globally increasing awareness about the environment, the therefore increasingly strict environmental guidelines (e.g. EURO-legislation, Kyoto protocol, limited fleet average CO<sub>2</sub> emissions [1],...) and the finiteness of the fossil fuel reserves drive us to the use of alternative fuels. Hydrogen is one possible way to come to ecological, (near) zero emission transport. Hydrogen is most commonly associated with fuel cells. Thus far it seems that affordable fuel cell vehicles will not be available for mass production anytime soon [2]. An appealing alternative is using hydrogen in internal combustion engines (H<sub>2</sub>ICEs). The relatively cheap H<sub>2</sub>ICEs permit the use of a normal spark ignition engine, a technology known to man for more than a century. Another advantage can be found in the fact that a hydrogen fueled ICE can easily be used as a bi- or flex-fuel engine and can thus be used to avoid the well known “chicken or the egg” dilemma. These benefits and experimental research on hydrogen-fueled ICEs have previously been reviewed by some of the authors [3,4,5].

Important progress has been made in the field of hydrogen fueled ICEs during the last decade. Nevertheless, some problems still remain. The three major bottlenecks are abnormal combustion, performance and NO<sub>x</sub> output. Although measures can be taken to avoid abnormal combustion (surface ignition, backfire, knock, [3,4,6]), there is no given set of measures with guaranteed success. Backfire (the ignition of the hydrogen-air mixture during the intake process) can be particularly difficult to predict and is thus the most difficult to control [7].

For what concerns the other two bottlenecks, they mostly have to be looked at as one, since there is a very strong power output versus NO<sub>x</sub> emissions trade-off for H<sub>2</sub>ICEs. The wide flammability limits of hydrogen-air mixtures allow lean mixtures resulting in lower combustion temperatures and thus low NO<sub>x</sub> output. Wide open throttle (WOT) operation can be applied, regulating power output through varying the equivalence ratio. This avoids throttle losses and increases engine efficiency. However, using this approach means richer mixtures (lean of stoichiometric) as the load increases, and correspondingly, increasing combustion temperatures. When going richer than a certain threshold equivalence ratio, peak cylinder temperatures exceed the NO<sub>x</sub> formation temperature and NO<sub>x</sub> emissions quickly rise to a maximum, which can be higher than that for gasoline operation. This is due to the higher adiabatic flame temperature and rapid combustion of hydrogen mixtures. Thus, hydrogen engines can be used at high efficiencies while producing hardly any NO<sub>x</sub>, but then the engine load is restricted. For the NO<sub>x</sub> output not to exceed 100 ppm a certain threshold equivalence ratio should not be crossed. This threshold equivalence ratio depends on the engine and varies between about  $\lambda = 1.8$  ( $\Phi = 0.56$ ) and  $\lambda = 2.2$  ( $\Phi = 0.45$ ), with  $\lambda$  the air to fuel equivalence ratio and  $\Phi$  the fuel to air equivalence ratio ( $\Phi = 1/\lambda$ ). For loads above the WOT threshold equivalence ratio condition, an approach using stoichiometric mixtures and throttling is required. Although engine efficiency suffers, exhaust gas aftertreatment is simple using a conventional three-way catalyst (TWC).

An alternative to the “stoichiometric + throttling”-strategy is to regulate the power output using stoichiometric mixtures and exhaust gas recirculation (EGR) [8]. Power output is then increased by decreasing the EGR percentage. This way it is possible to keep controlling the energy content of the mixture in the cylinder (qualitative control) without the need of a throttle valve (quantitative control) and thus reducing the pumping losses. In short, this strategy -made possible by the high EGR tolerance of hydrogen mixtures- increases

efficiency compared to the “stoichiometric + throttling”-strategy [9]. Nonetheless, the increasing combustion instability for high EGR percentages can be a practical limit [10].

The easiest way to introduce hydrogen ICEs is by converting a gasoline engine to bi-fuel operation, by mounting a hydrogen fuel system and gas injectors. Starting from a gasoline engine implies port fuel injection (PFI) of the hydrogen. Therefore, this paper concentrates on hydrogen PFI engines, although dedicated hydrogen engines would benefit greatly from direct injection [3, 4, 11,12] and novel concepts for mixture formation have been reported [13]. Also, work has been published on compression ignition operation on H<sub>2</sub> [14].

A PFI hydrogen engine operating stoichiometric at WOT, has a theoretical power deficit of about 18% compared to a gasoline engine, due to the lower volumetric energy density. In practice, the power deficit can be even higher if the equivalence ratio has to be limited to avoid abnormal combustion. This can be circumvented by using variable valve timing, which has been used successfully to enable hydrogen engines to run stoichiometric without backfire [9], through better scavenging of hot exhaust gases. Huynh T.C. et al [15] found that backfire is especially affected by the valve overlap period (VOP). By varying the VOP and supercharging they achieved to get backfireless operation at equal or higher performance than on gasoline. In [8] the authors demonstrate that supercharging is an appropriate means to increase the power output of H<sub>2</sub> PFI engines.

## 2 Experimental set-up

### 2.1 Engine

A Volvo four cylinder sixteen valve gasoline engine with a total swept volume of 1783cc and a compression ratio of 10.3:1 was converted to bi-fuel operation by mounting an additional fuel rail supplying gaseous fuel to 8 (2 per cylinder) Teleflex GSI gas injectors, mounted on the intake manifold [16]. The intake manifold was modified to avoid any damage if backfire would occur during the hydrogen measurements: a T-type branch pipe was mounted on the intake manifold with the ‘straight ahead’ branch closed by a foam plug and the other branch leading to the air filter – mass air flow (MAF) sensor – throttle valve assembly, see Fig. 1. Any pressure rise in the intake manifold due to the occurrence of backfire results in the foam plug being blown out instead of damaging other components such as the MAF sensor.

The engine has continuously variable valve timing (CVVT) on the intake camshaft, allowing up to 40 degrees crank angle (°ca) advance of the intake valve opening and closing time. A MoTeC M800 engine control unit is used to control ignition timing, start of injection, injection duration and intake valve timing. Cylinder pressure measurements were possible using a spark plug pressure sensor, a piezo-electric Kistler type 6117BFG16. A piezo-resistive Kistler type 4075A20 sensor placed in the intake manifold close to one of the inlet valves was used for pegging the cylinder pressure. The crank angle was recorded using a Kistler crank angle encoder type 2613B1.

## **2.2 Fueling system**

In the beginning of the experiments the accommodation of the lab did not permit the storage of a large volume of hydrogen. The limited supply of hydrogen made it impossible to do extensive testing at high loads and engine speeds. An adapted fueling system, shown in Fig. 2, with hydrogen storage outside of the lab made it possible to extend tests to higher RPMs, because of the increased available volume of hydrogen.

The adapted fuel system, shown in Fig. 2 uses three different pressure regulators to reduce and regulate the pressure. The first stage is located outside where the hydrogen is stored. This regulator diminishes the pressure from up to 200 bar to 40 bar. Close to the engine -at the second stage- the pressure is decreased to 4.5 bar. The last pressure regulator (brand: Fairchild, type: 4500A) is electro-pneumatically operated to ensure a constant hydrogen injection pressure (2 barg) for a wide range of fuel flows. Just before the fuel rail the hydrogen flow is measured (brand: Bronkhorst, type: F-116AI-FD-00-V). At the beginning of the fuel rail an electronically operated valve closes the fuel supply when the engine is turned off or when the emergency system is activated. The manometer on the other end of the fuel rail can give information on any leaking injectors if pressure drops when the engine is turned off. Furthermore several mechanical valves are implemented in the system, to avoid air getting into the system while work is in progress on the engine or when the hydrogen tanks get changed.

## **2.3 Emissions**

The exhaust gas components O<sub>2</sub>, CO, CO<sub>2</sub>, NO, NO<sub>x</sub> and H<sub>2</sub> were measured (O<sub>2</sub>: Maihak Oxor-P S710, paramagnetic; CO-CO<sub>2</sub>-NO-NO<sub>2</sub>: Maihak Multor 610, non-dispersive infra red; H<sub>2</sub>: Maihak Thermor 615, thermal conductivity). A direct reading of the air to fuel equivalence ratio  $\lambda$  is given by a Bosch wide band sensor and digital air/fuel ratio meter with calibrations for hydrogen and gasoline.

## **2.4 Safety**

Due to the operating principle of the electro pneumatically controlled pressure regulator -which uses pressurized air to regulate the pressure so that it is independent of the hydrogen flow rate- it is possible that some hydrogen has to be blown off to avoid a pressure increase downstream (e.g. going from a high load high speed point to idling). A tube was connected to this blow off opening and led to the tapered roof above the engine test bench, where a Buveco Bucom ST600EX hydrogen sensor was installed at the highest point. In case of any leakage an alarm is activated around 1 vol% of hydrogen in air (this is equivalent to 25% of the lower flammability limit.).

## 3 Control strategy used for hydrogen operation

### 3.1 Strategy

Wide open throttle operation ensures the highest volumetric efficiency, but is only useable at low loads, as for achieving higher loads it is necessary to use a richer mixture, which causes the NO<sub>x</sub> output to exceed the acceptable rate of 100 ppm, as discussed in the introduction. At stoichiometric operation with throttling, NO<sub>x</sub> aftertreatment with a TWC is straightforward. However, the region between the threshold equivalence ratio, where NO<sub>x</sub> emissions get too high, and stoichiometric is to be avoided, as exhaust gas aftertreatment (to reduce NO<sub>x</sub> emissions) in this interval is complex because of the excess of oxygen in the exhaust gas. Lowering the amount of NO<sub>x</sub> in the exhaust gas would require lean NO<sub>x</sub> aftertreatment, such as a Lean NO<sub>x</sub> Trap (LNT) [17,18].

Hence, to put a H<sub>2</sub>ICE to optimum use for efficiency and performance, the control strategy has to include a switch from lean burn ( $\lambda > 2$ ) to stoichiometric operation. The practical implementation of the combination of quantitative and qualitative mixture control for the test engine is made clear in Fig. 3. Here, it is straightforward that when load increases lambda has to decrease (at the left hand side of the graph), but at a certain point ( $\lambda \approx 2$ ) the NO<sub>x</sub> level transcends the acceptable threshold level of 100 ppm. At this point the engine control strategy has to leap from WOT qualitative to quantitative throttled (at the right hand side of the graph), where a catalytic converter can bring the NO<sub>x</sub> emissions down to an acceptable level. The use of this control strategy was successfully implemented by BMW in the Hydrogen 7 as described in [19,20]. To get a smooth transition from one mode to another, throttle position has to be changed very fast to avoid big disparities in power output. This causes technical difficulties. During the change between the two modes variable valve timing might help to smoothen out the transition.

### 3.2 CVVT

The continuously variable valve timing on the engine enables a continuous time phasing of the intake valves. The valve lift and the total opening time remain fixed. Accordingly, valve timing is a compromise, as the ram effect which depends on the intake valve closing time will be adverse to the dynamic effects of the exhaust gases leaving the cylinder, which are dependent on the opening time of the intake valve. For an easier understanding, Fig. 4 shows the intake and exhaust cam profiles of the test engine and the effects of an advancement of the intake valve opening time (IVO). The valve timing is also adjustable with the MoTeC programmable motor management system. The intake valve opening time is adjustable as a function of the load and the speed of the engine.

The valve timing can be used to influence the power output as a means to get a softer transition between the two engine operation modes discussed above.

In the results given below, the influence of the CVVT system on, among others, the volumetric efficiency  $\eta_{vol}$  will be discussed. For gaseous fuels it is important to properly define volumetric efficiency. Here, the following definition is used:

$$\eta_{vol} = \frac{Q_{air} + Q_{H_2}}{V_s \cdot \frac{n}{60} \cdot \frac{1}{2}}$$

With  $Q_{air}$  and  $Q_{H_2}$  the measured air and hydrogen volumetric flow rates, respectively,  $V_s$  the swept volume and  $n$  the engine speed.

## 4 Results

### 4.1 Gasoline

The influences of the CVVT system were first examined for gasoline to determine a base set to compare the hydrogen results with. All measurements are for MBT ignition timing, closed loop lambda control ( $\lambda = \Phi = 1$ ) and quantitative (throttled) load control. Results of CVVT on gasoline are given qualitatively in the following.

When advancing the intake valve opening time at low loads (small throttle openings or throttle position TP), the resulting increased valve overlap (see Fig. 4) leads to a higher amount of internal EGR. This is caused by exhaust gases being pushed back into the inlet manifold when the intake valves open, due to the low gas momentum in the intake and exhaust manifolds at low loads. During the first part of the intake stroke, the descending piston draws a mixture of fresh charge and exhaust gases into the cylinder. The higher amount of internal EGR leads to decreased  $NO_x$  emissions as peak temperatures are lowered. This is confirmed by the results, shown in Fig. 5, of  $NO_x$  and brake torque at 1500 and 2500 rpm. However, at very low IVO advancement, ignition timing has to be delayed in such a way that its diminishing effect on the  $NO_x$  production more than compensates for the effect of the lesser internal EGR, see Fig. 5: 1500 rpm at 4°ca IVO advance. The engine torque did not vary significantly with intake valve timing, so volumetric efficiency is hardly affected. This is due to the conflicting trends between higher internal EGR with advanced IVO and, for these low loads, more fresh charge being pushed back in the intake manifold at the end of the intake phase when IVO is delayed (delaying the closing of the intake valves due to the fixed opening duration, see Fig. 4). At low engine speeds, there is a critical amount of EGR above which combustion becomes instable. This poses a limit to the IVO advancement (and thus the valve overlap). This can be seen in Fig. 5 from the decreased torque at 1500 rpm at an IVO advance of 40°ca.

At medium loads and low to medium engine speeds, the higher valve overlap (advanced IVO) leads to better cylinder filling (thus, a higher volumetric efficiency) due to the inertia of the exhaust gases flowing from the cylinder to the exhaust manifold. The opposing effects of  $NO_x$  increase at lowered IVO advance and  $NO_x$  decrease due to the lower ignition advance, which was also visible for low loads and engine speed, are shown in Fig. 6 for 2500 rpm.

At high loads and low to medium engine speeds the results of the higher volumetric efficiency due to advanced IVO are even more distinct. This can be seen on the brake torque curve at 2500 rpm in Fig. 7. In this operating range internal EGR is absent and  $\text{NO}_x$  emissions are not a function of the IVO time.

At higher engine speeds, a compromise is necessary for an optimal volumetric efficiency: early intake valve opening and thus large valve overlap to use exhaust gas inertia, versus late intake valve closing to use the inertia of the fresh charge for a better filling at the end of the intake phase (ram effect). This trade-off can be seen on the brake torque curve at 3500 rpm (Fig. 7) that shows a maximum at an IVO advance of  $28^\circ\text{ca}$ .

## 4.2 Hydrogen

As mentioned before, measurements on hydrogen were limited to 1500rpm at the beginning of the tests (due to limitations of the hydrogen storage in the lab). The results for these tests were already partly discussed elsewhere [21].

The adapted fueling system made more extensive testing possible, so measurements were extended to higher engine speeds. Given that changes have been made to the test setup since the low engine speed experiments, results will be given and discussed separately to avoid misinterpretation. All measurements reported here are for MBT ignition timing. Both qualitative load control (varying the equivalence ratio at WOT) and quantitative load control (throttling at stoichiometric) have been investigated.

### 4.2.1 Wide open throttle, qualitative control:

Figure 8 shows results at wide open throttle, for a fixed engine speed of 1500 rpm. Here, the air to fuel equivalence ratio is used to regulate load, with the brake torque increasing as more hydrogen is injected. Brake torque and  $\text{NO}_x$  emissions are shown as a function of the air to fuel equivalence ratio, for different intake cam phasing. Advancing IVO at a fixed equivalence ratio leads to a higher brake torque. As injection durations had to increase to keep the equivalence ratio constant, this means the volumetric efficiency increases with advancing IVO. This is caused by the low momentum of the fresh charge at low engine speed, meaning it is best to close the intake valves early so no fresh charge is pushed back in the intake manifold at the end of the intake phase (see Fig. 4).

$\text{NO}_x$  emissions have no clear trend with the intake cam phasing. These results are qualitatively similar to the results at high loads (large throttle openings) and medium speeds on gasoline, see Fig. 7: brake torque increases with advanced IVO while  $\text{NO}_x$  emissions are relatively unaffected. The threshold equivalence ratio for this engine is  $\lambda \approx 2$  ( $\Phi \approx 0.5$ ). One would expect the  $\text{NO}_x$  emissions to peak around  $\lambda = 1.2$  ( $\Phi = 0.83$ ) [4]. This peak is not visible in the figure as the measurements were done at 0.25 increments in air to fuel equivalence ratios.

An important result to note from Fig. 8 is that it was not possible to run at stoichiometric for the most advanced intake cam timings. At these timings, backfire occurred. It is not clear what causes this. There is an increased contact between fresh charge and exhaust gas due to the longer valve overlap, but the injection duration of hydrogen for these low speeds is such that there is more than enough time to delay the start of injection until after the exhaust valve closes and still have an end of injection before BDC. Contrary to our findings, Tang et al.

at Ford [19] did not find an influence of the intake valve timing on the richest mixture possible for WOT on their 2.0 liter Zetec engine. In their case, the mixture richness was limited (to  $\lambda \approx 1.35 / \Phi \approx 0.74$ ) due to the occurrence of pre-ignition events.

The gain in torque output with the richer mixtures that are possible with the small intake cam advance, far outweighs the effects of the intake cam phasing. For example, at  $\lambda = 1.5$ , which is the richest point possible using the maximum intake cam advance of  $40^\circ\text{ca}$ , the brake torque is 83.2 Nm, compared to 70 Nm at  $4^\circ\text{ca}$  IVO advance. However, up until  $16^\circ\text{ca}$  IVO advance, stoichiometric operation is possible resulting in a torque output of 97 Nm (compared to about 120 Nm on gasoline).

#### **4.2.2 Quantitative, throttled engine control:**

From Fig. 8 it is clear that the region between stoichiometric mixtures and the threshold equivalence ratio is to be avoided due to excessive  $\text{NO}_x$  emissions and (oxygen-rich) conditions that prohibit efficient aftertreatment. When the load demand is higher than that reachable (from the viewpoint of acceptable  $\text{NO}_x$ ) with the WOT qualitative control strategy, one has to switch to a throttled strategy at stoichiometric mixtures as Fig. 3 illustrates.

Figure 9 shows results for stoichiometric mixtures as a function of the throttle position, for a fixed engine speed of 1500 rpm. Again, an advanced IVO increases the brake torque. This effect is strongest for the larger throttle openings, as the momentum of the gas (fresh charge or exhaust gas) in the manifolds is higher. The mechanism by which torque increases is again that less fresh charge is being pushed back in the intake when closing the intake valve shortly after BDC. For the smaller throttle openings, an advanced IVO leads to higher internal EGR due to the depression in the intake. At these throttle openings, the conflicting trends between internal EGR and volumetric efficiency result in a negligible effect of intake cam phasing on torque output. This is analogous to the low load results on gasoline, see Fig. 5.

Looking at the trends found for gasoline, the effects on  $\text{NO}_x$  emissions for throttled hydrogen operation are expected to be a decrease in  $\text{NO}_x$  with advancing IVO at the smaller throttle openings, and a negligible effect for the larger throttle openings. This is confirmed by the results in Fig. 9, where the maximum  $\text{NO}_x$  output is also reached at  $16^\circ\text{ca}$ . The explanation for this phenomenon is the same as was the case at gasoline operation. It can be seen in Fig. 9 that no results are shown for advanced IVO phasing at the larger throttle openings. This is because the maximum throttle opening for these IVO timings is limited due to the occurrence of backfire. This is similar to the limitations on the richest equivalence ratio for advanced IVO in Fig. 8. However, for the throttled case the benefit of IVO advance on torque is greater than going to a less advanced IVO and WOT (the highest torque output of 100 Nm is reached at an IVO advance of  $28^\circ\text{ca}$  and 80% open throttle, compared to 97 Nm at  $16^\circ\text{ca}$  and WOT).

#### **4.2.3 Extension to higher engine speeds**

The experiments on the effects of variable intake valve timing were extended to higher engine speeds. Measurements have been taken at WOT and MBT timing for 2500 rpm and 3500 rpm. For these measurements the air to fuel equivalence ratio was set as low as possible for maximum torque (around  $\lambda = 2$ ) and acceptable  $\text{NO}_x$  output.

At 2500 rpm and increasing IVO advancement the injected amount of hydrogen needed to be lowered for the NO<sub>x</sub> emissions not to exceed 100 ppm. At WOT operation and medium to high engine speeds, the inertia of the exhaust gases is relatively high. During the valve overlap, the inertia of the exhaust gases creates a vacuum that sucks fresh charge into the cylinder. Thus, a longer valve overlap period leads to a better cylinder filling. This increase in volumetric efficiency is confirmed by Figs. 10 and 11 and is analogous to the trends seen in gasoline operation at medium to high loads (Fig. 6 and 7).

Figures 12 and 13 show the effect of intake valve timing on the in-cylinder peak pressure at 2500 rpm. Peak pressures shown are the average of 100 cycles. One can clearly see on Fig. 13 that for this engine speed the engine peak pressure reaches a maximum at 28°ca advancement, a 25% increase compared to the peak pressure at 4°ca advancement. This optimum is caused by the necessity to dilute the mixture at higher IVO advancement, as stated above, since otherwise the increasing peak pressures would make the engine temperature rise above the NO<sub>x</sub> formation temperature. This decrease in energy density of the mixtures at higher IVO advances is responsible for the lower peak pressures under these working conditions and also accounts for the clearly visible optimum in torque around 28°ca IVO advancement seen in Fig. 10. So, although torque rises due to increasing volumetric efficiency for higher IVO advancements, at a certain point the decreasing energy density of the mixture in the cylinder, due to the higher dilution of the mixture, makes the torque decrease again.

Since the definition of volumetric efficiency (see Section 3.2) takes into account the flow rate of air as well as the flow rate of hydrogen, a change of the air to fuel ratio does not translate into a direct change of volumetric efficiency, as an increase of one implies a decrease of the other.

Because the spark plug accommodated with the pressure sensor, was not adapted for hydrogen use (with the heat range being too high), operation conditions were restricted and the comparison has been limited to  $\lambda \geq 2$ . The unadapted spark plug caused misfires which leads to unburned hydrogen remaining in the cylinder causing backfire in the following cycle. Figure 14 shows a clear case of misfire in cycle number 9. Figure 15 shows the occurrence of backfire on a plot of intake manifold pressure versus crank angle. The cycle where backfire occurs is plotted in bold.

## 5 Discussion and conclusion

At low engine speeds the advantage of a varying intake valve timing at WOT (qualitative control) is limited, but allows an extension of the qualitative control range (the range on the lean side of the threshold equivalence ratio) to slightly higher engine loads. This is advantageous as the brake thermal efficiency is significantly higher for this load control strategy (with the brake thermal efficiency reaching a maximum of about 35% around  $\lambda = 2$  for the four cylinder engine [16]). CVVT had no (measured) effect on the threshold equivalence ratio, implicating that the region between stoichiometric and the equivalence ratio, which has to be avoided due to complex NO<sub>x</sub> aftertreatment, remains alike.

When higher loads are demanded than can be delivered with WOT operation at the threshold equivalence ratio, one has to switch to stoichiometric operation. Throttling can then be used to regulate the power output. The jump from qualitative to quantitative operation is not straightforward since one needs to be able to change the TP very fast for smooth engine behavior. Since Fig. 3 shows that only minor steps are necessary to permit the leap between the two operation modes, variable valve timing can help smooth out the transition. Additionally, the ignition timing can be adjusted [18].

Results for medium to high engine speeds show that at wide open throttle operation, CVVT can improve volumetric efficiency by advancing the intake valve opening time. However, at 2500 rpm and 28°C IVO advance a clear optimum in torque was found, which can be explained by the high in-cylinder pressures, causing high temperatures at advanced IVO timings, leading to NO<sub>x</sub> formation, which can only be prevented by diluting the mixture, causing the torque to decrease.

## Acknowledgements

The authors thank the following MSc students for performing measurements during their end-of-year thesis, contributing to this work: C. Billiow, J. De Landtsheere, F. De Smet, P. Maesschalck, S. Martin, N. Rombaut, A. Trenson and M. Vermeir, The support by the technical staff, K. Chielens, P. De Pue and R. Janssens is also greatly appreciated.

## References

- [1] Leading by CO<sub>2</sub> example. Editorial, Automotive Engineering International, April 2007.
- [2] Fuel Cells. Cover story, Engine Technology International, June 2007.
- [3] S. Verhelst, S. Verstraeten, R. Sierens, A critical review of experimental research on hydrogen fueled SI engines, SAE technical paper nr 2006-01-0430, 2006. Also in SAE 2006 Transactions, Journal of Engines, pp264-74.
- [4] S. Verhelst, T. Wallner, Hydrogen-fueled internal combustion engines, Prog Energy Combust Sci 2009;35:490-527.
- [5] S. Verhelst, R. Sierens, Recent Progress in Hydrogen-Fueled Internal Combustion Engines, in: S. Demidov, J. Bonnet (Eds.), Traffic Related Air Pollution and Internal Combustion Engines, Nova Science Publishers Inc., New York, 2009, pp. 211-49.
- [6] S. Szwaja, K.R. Bhandary, J.D. Naber, Comparisons of hydrogen and gasoline combustion knock in a spark ignition engine, Int J Hydrogen Energy 2007;32:5076-87.
- [7] X. Liu, F. Liu, L. Zhou, B. Sun, H.J. Schock, Backfire prediction in a manifold injection hydrogen internal combustion engine, Int J Hydrogen Energy 2008;33:3847-55.
- [8] S. Verhelst, P. Maesschalck, N. Rombaut, R. Sierens, Increasing the power output of hydrogen internal combustion engines by means of supercharging and exhaust gas recirculation, , Int J Hydrogen Energy 2009;34:4406-12.
- [9] M. Berckmüller, H. Rottengruber, A. Eder, N. Brehm, G. Elsässer, G. Müller-Alander, C. Schwarz, Potentials of a charged SI-hydrogen engine, SAE technical paper nr 2003-01-3210, 2003.
- [10] S. Verhelst, R. Sierens, Combustion studies for PFI hydrogen IC engines, SAE technical paper nr 2007-01-3610, 2007.
- [11] C.M. White, R.R. Steeper, A.E Lutz., The hydrogen-fueled internal combustion engine: a technical review, Int J Hydrogen Energy 2006;31:1292-305.
- [12] V. Knop, A. Benkenida, S. Jay, O. Colin, Modelling of combustion and nitrogen oxide formation in hydrogen-fuelled internal combustion engines within a 3D CFD code, Int J Hydrogen Energy 2008;33:5083-97.

- [13] A.A. Boretti, H.C. Watson, The lean burn direct injection jet ignition gas engine, *Int J Hydrogen Energy* 2009;34:7835-41.
- [14] J.M. Gomes Antunes, R. Mikalsen, A.P. Roskilly, An experimental study of a direct injection compression ignition hydrogen engine, *Int J Hydrogen Energy* 2009;34:6516-22.
- [15] T.C. Huynh, J.K. Kang, K.C. Noh, J.T. Lee, J.A. Caton, Controlling Backfire for a Hydrogen-Fueled Engine Using External Mixture Injection, *J Eng Gas Turbines Power* 2008;130:062804-1-8.
- [16] S. Verhelst, P. Maesschalck, N. Rombaut, R. Sierens, Efficiency comparison between hydrogen and gasoline, on a bi-fuel hydrogen/gasoline engine, *Int J Hydrogen Energy* 2009;34:2504-10.
- [17] M. Touns, P. Menegazzi, P. Rouchon, NOx Trap Model For Lean Burn Engine Control, SAE technical paper nr 2003-01-2292, 2004.
- [18] T. Fujita, S. Ozawa, K. Yamane, Y. Takagi, Y. Goto, M. Odaka, Performance of NOx absorption 3-way catalysis applied to a hydrogen fueled engine, 15th World Hydrogen Energy Conference, Yokohama, Japan; 2004.
- [19] T. Wallner, H. Lohse-Busch, S. Gurski, M. Duoba, W. Thiel, D. Martin, T. Korn, Fuel economy and emission evaluation of BMW Hydrogen 7 Mono-Fuel demonstration vehicles, *Int J Hydrogen Energy* 2008;33:7608-18.
- [20] G. Kiesgen, M. Klüttingen, C. Block, H. Fischer, The New 12 Cylinder Hydrogen Engine in the 7 Series: The H2 ICE Age Has Begun, SAE technical paper nr 2006-01-0431, 2006.
- [21] S. Verhelst, J. De Landtsheere, F. De Smet, C. Billiouw, A. Trenson, S. Sierens, Effects of Supercharging, EGR and Variable Valve Timing on Power and Emissions of Hydrogen Internal Combustion Engines, SAE technical paper nr 2008-01-1033, 2008. Also in *SAE International Journal of Engines* Vol. 1(1), pp. 647-656, 2009.

### Figure captions

Figure 1. Modified intake of the engine.

Figure 2. The adapted hydrogen storage and fueling system.

Figure 3. The combined qualitative and quantitative control strategy of the hydrogen fueled test engine.

Figure 4. Intake and exhaust valve profiles. Advancing the intake valve timing results in a larger valve overlap.

Figure 5. Brake torque and raw NO<sub>x</sub> emissions on gasoline, for low loads at low and medium engine speeds, as a function of intake cam phasing.

Figure 6. Brake torque and raw NO<sub>x</sub> emissions on gasoline, for medium loads at low and medium engine speeds, as a function of intake cam phasing.

Figure 7. Brake torque and raw NO<sub>x</sub> emissions on gasoline, for high loads at medium to high engine speeds, as a function of intake cam phasing.

Figure 8. Brake torque and NO<sub>x</sub> emissions on hydrogen, at 1500 rpm and wide open throttle, for different intake cam phasing, as a function of the equivalence ratio.

Figure 9. Brake torque and NO<sub>x</sub> emissions on hydrogen, at 1500 rpm,  $\lambda=1$ , for different intake cam phasing, as a function of throttle opening.

Figure 10. Brake torque, air to fuel equivalence ratio ( $\lambda$ ) and volumetric efficiency on hydrogen (WOT) as function of IVO advance at 2500 rpm.

Figure 11. Brake torque and volumetric efficiency on hydrogen (WOT) as a function of IVO advance at 3500 rpm.

Figure 12. The influence of IVO advancement on in-cylinder peak pressure at 2500 rpm (on hydrogen, WOT).

Figure 13. Detail of Figure 12 around the cylinder peak pressure.

Figure 14. Cyclic variation in in-cylinder pressure and the effect of misfires due to a failing spark plug for hydrogen at WOT.

Figure 15. Backfire as a result of misfires for hydrogen at WOT. Intake manifold pressure measurements of 7 consecutive cycles. Backfiring cycle shown in bold.

Figure1  
[Click here to download high resolution image](#)

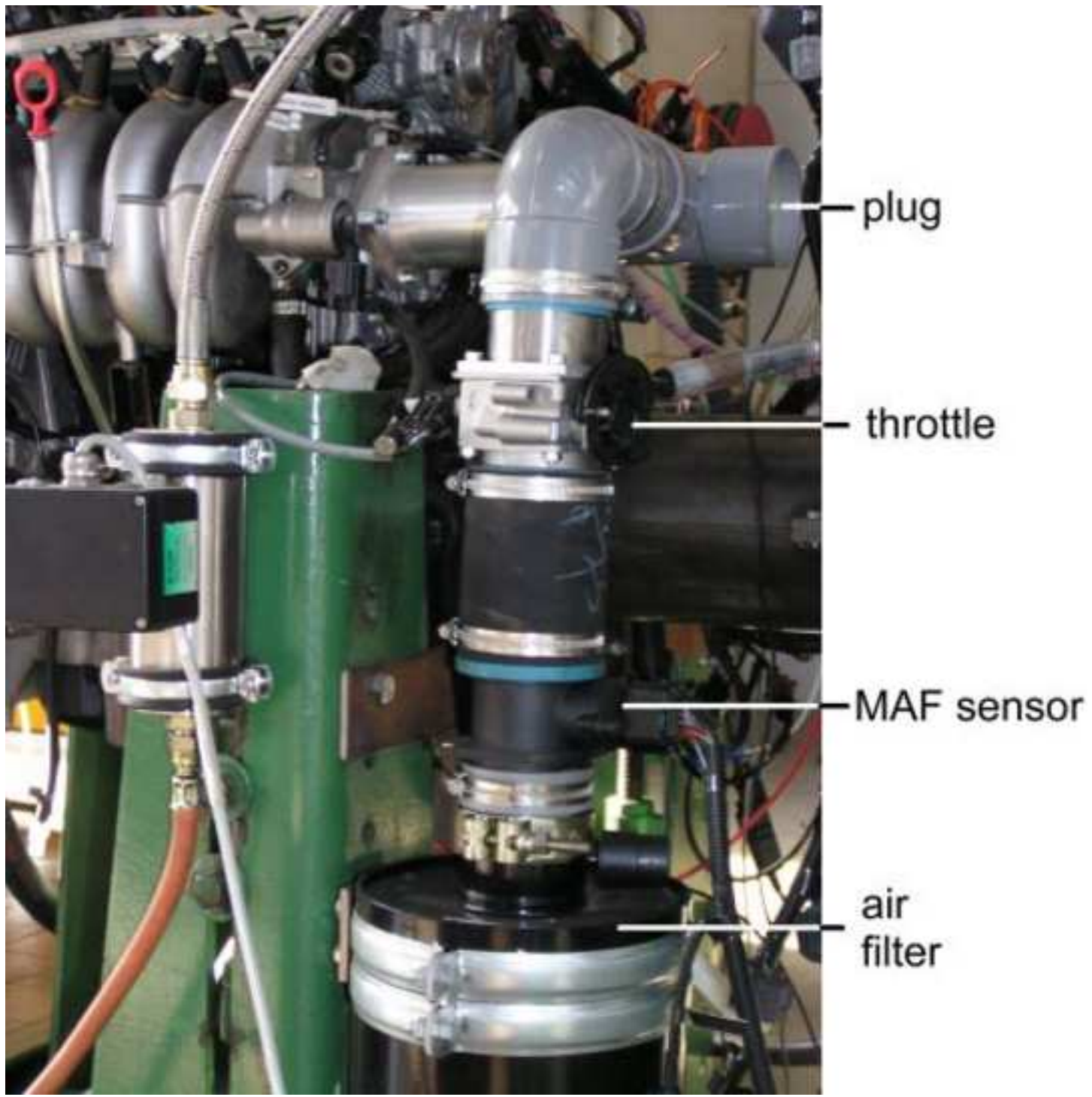
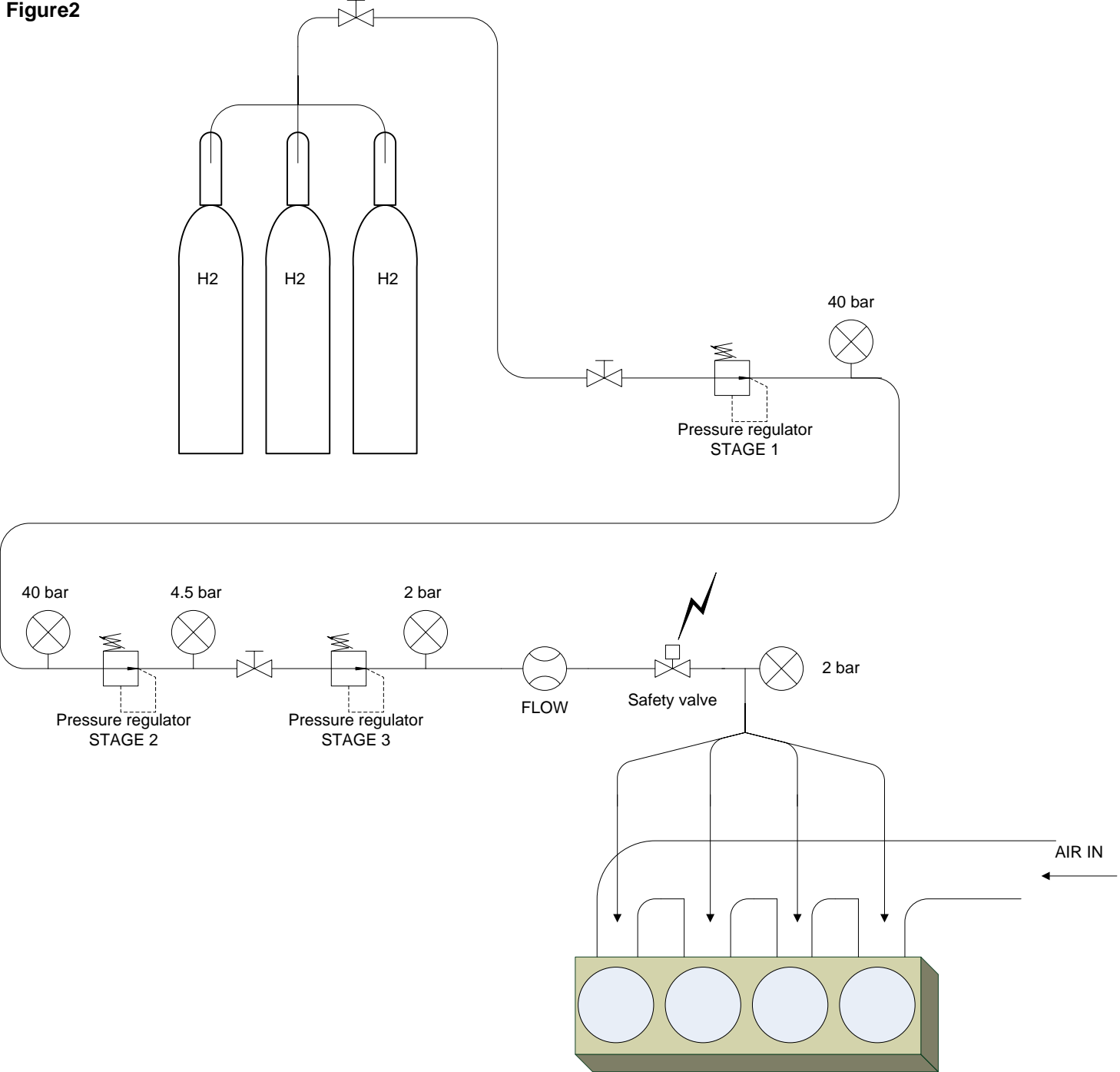
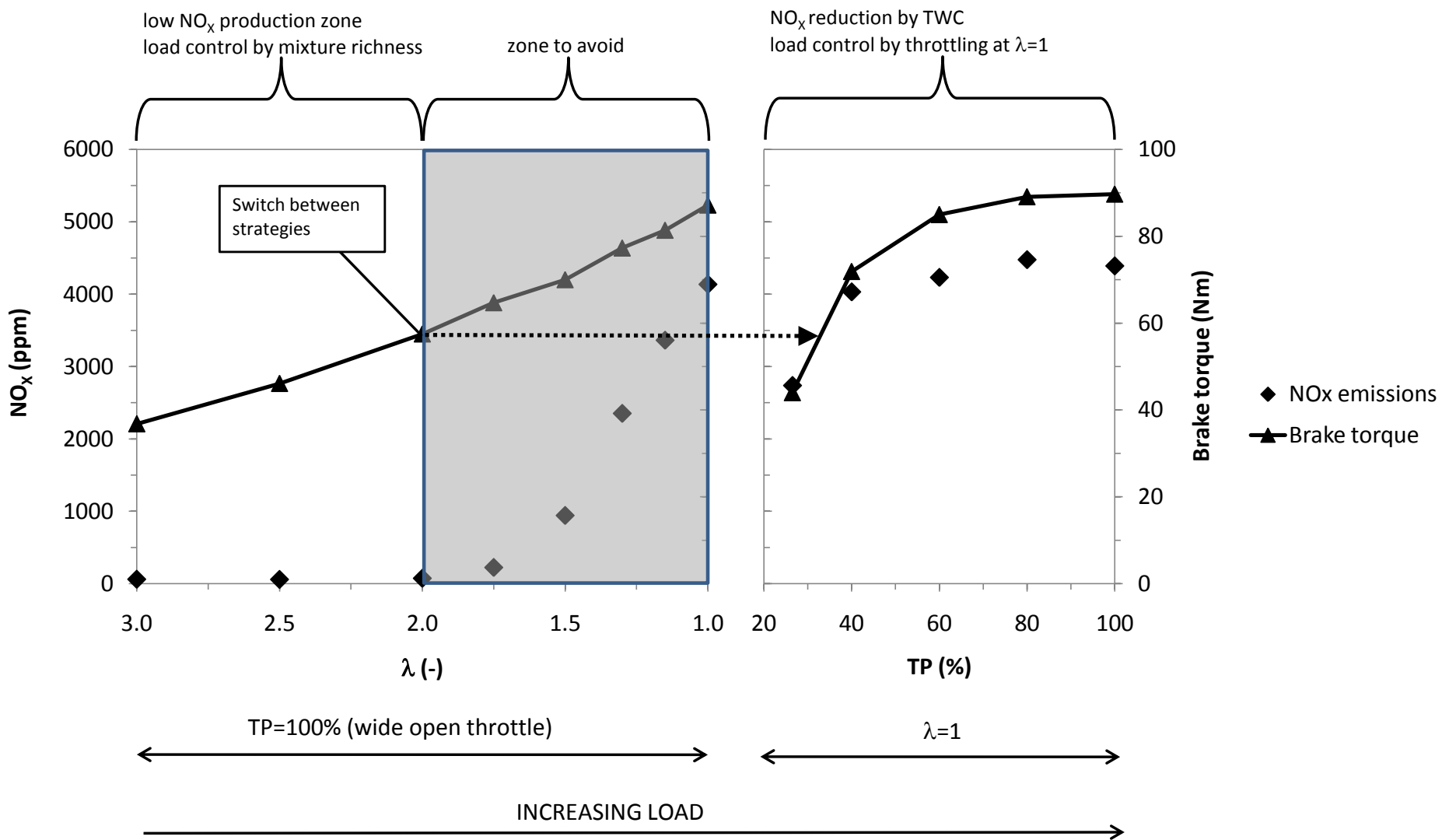


Figure2





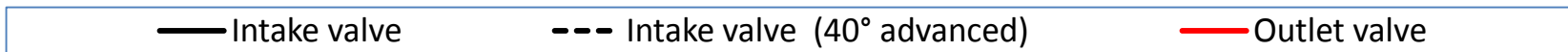
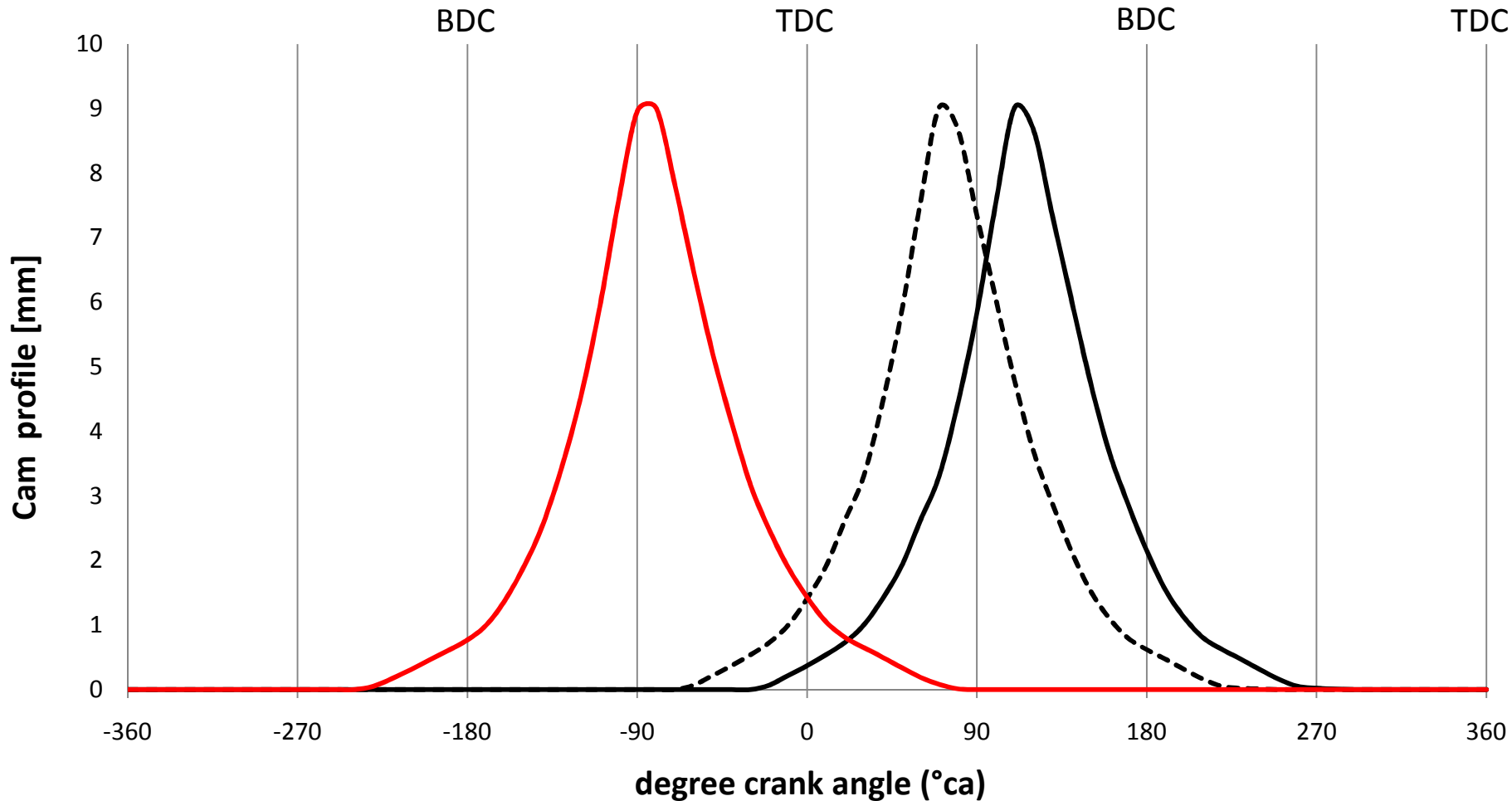


Figure5

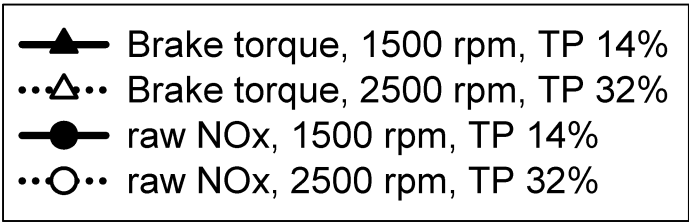
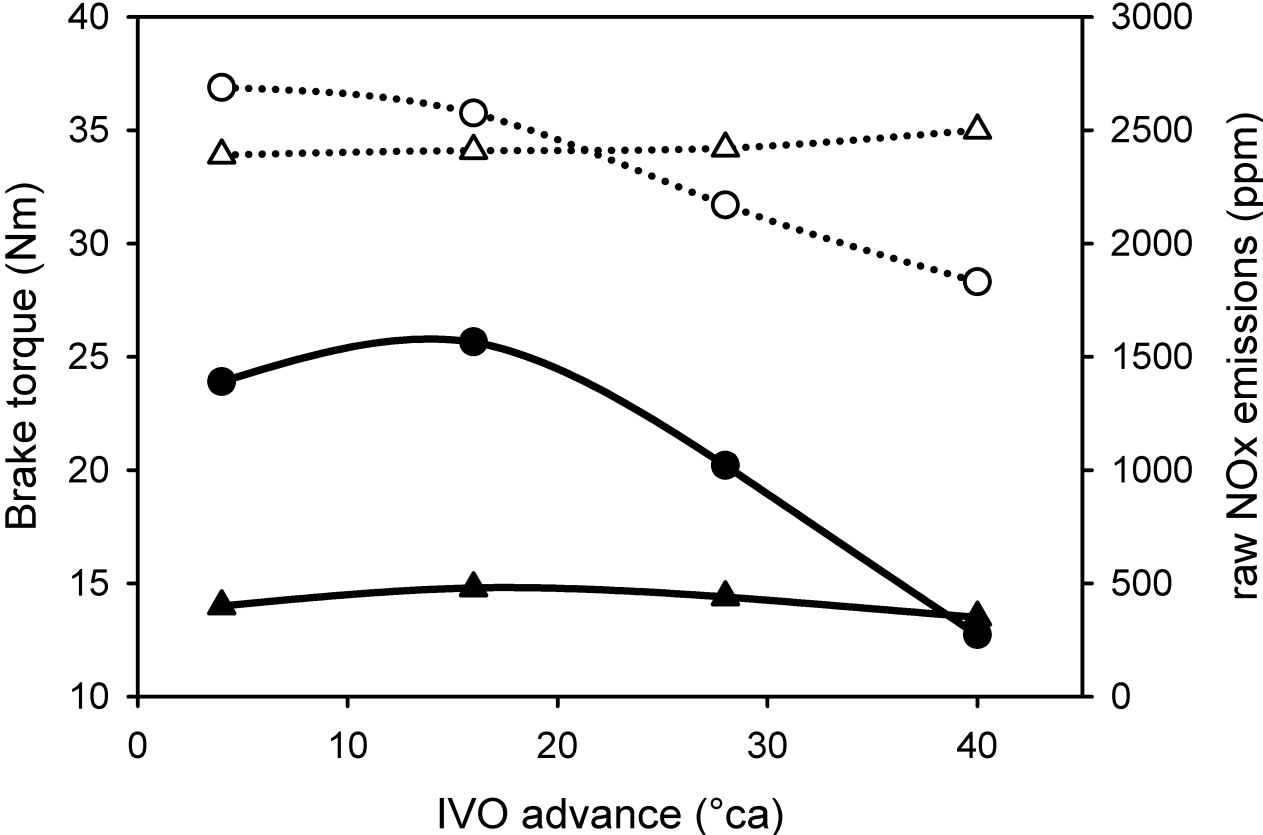
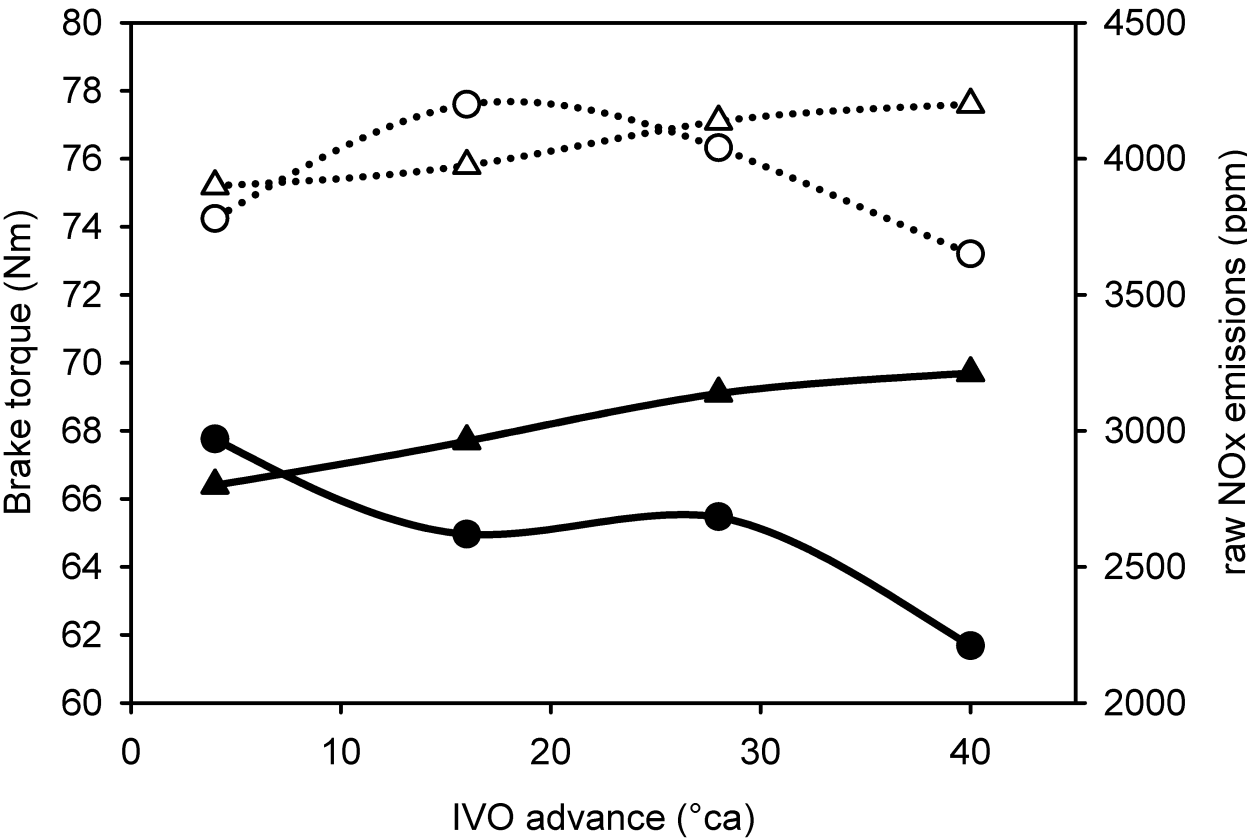
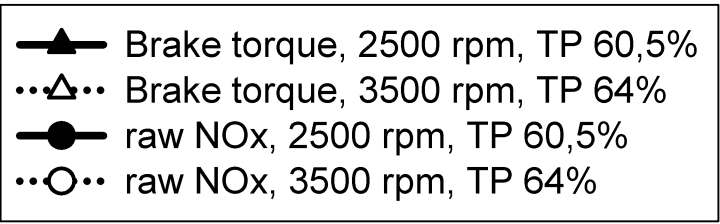
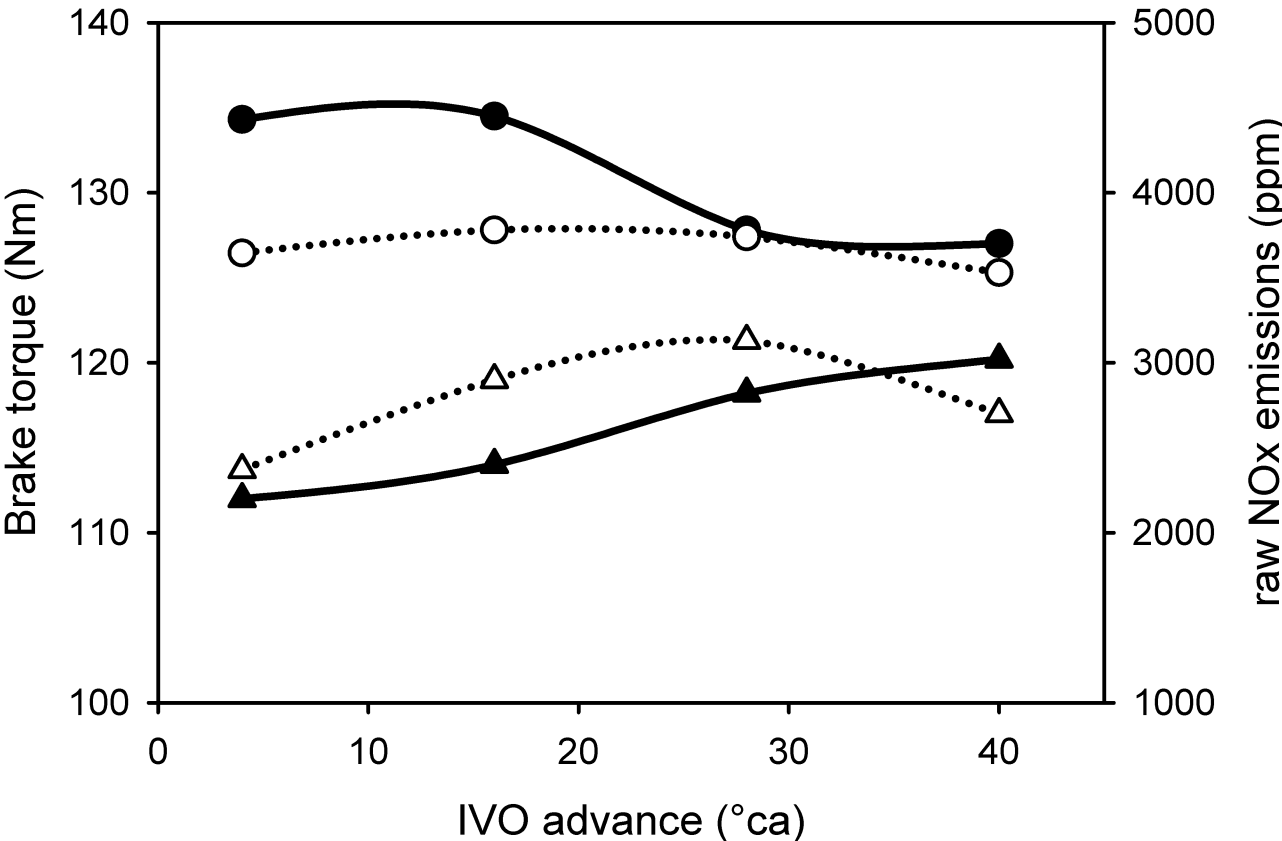


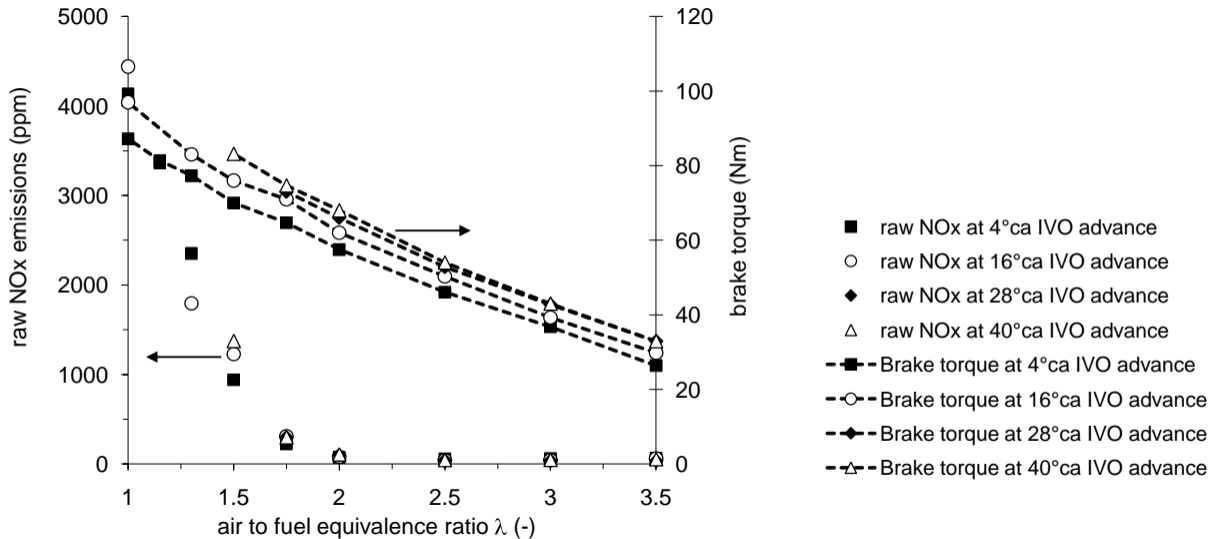
Figure6



- ▲— Brake torque, 1500 rpm, TP 32%
- △·· Brake torque, 2500 rpm, TP 45%
- raw NOx, 1500 rpm, TP 32%
- raw NOx, 2500 rpm, TP 45%

Figure7





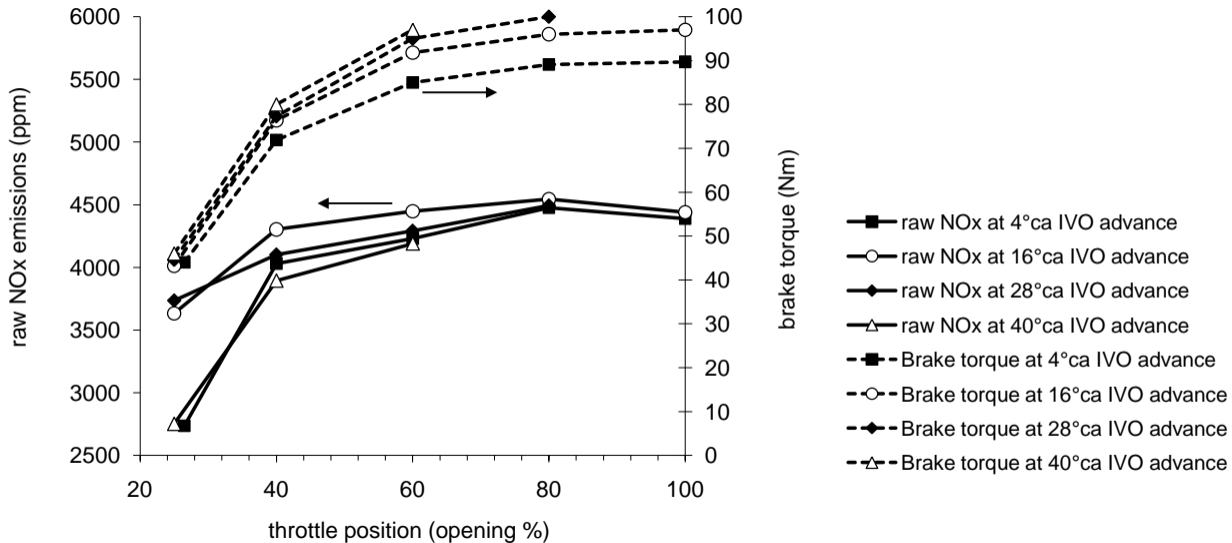


Figure10

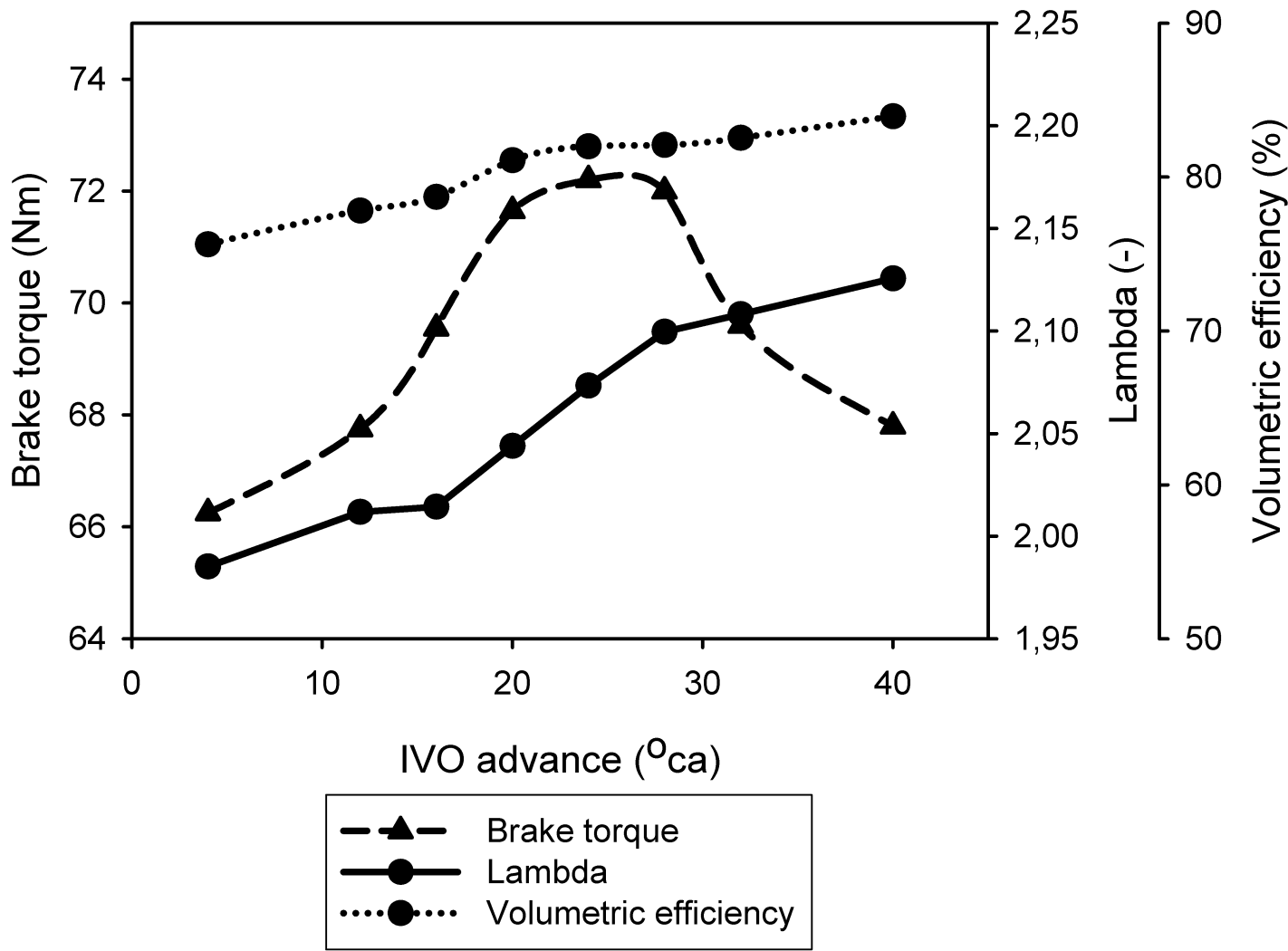
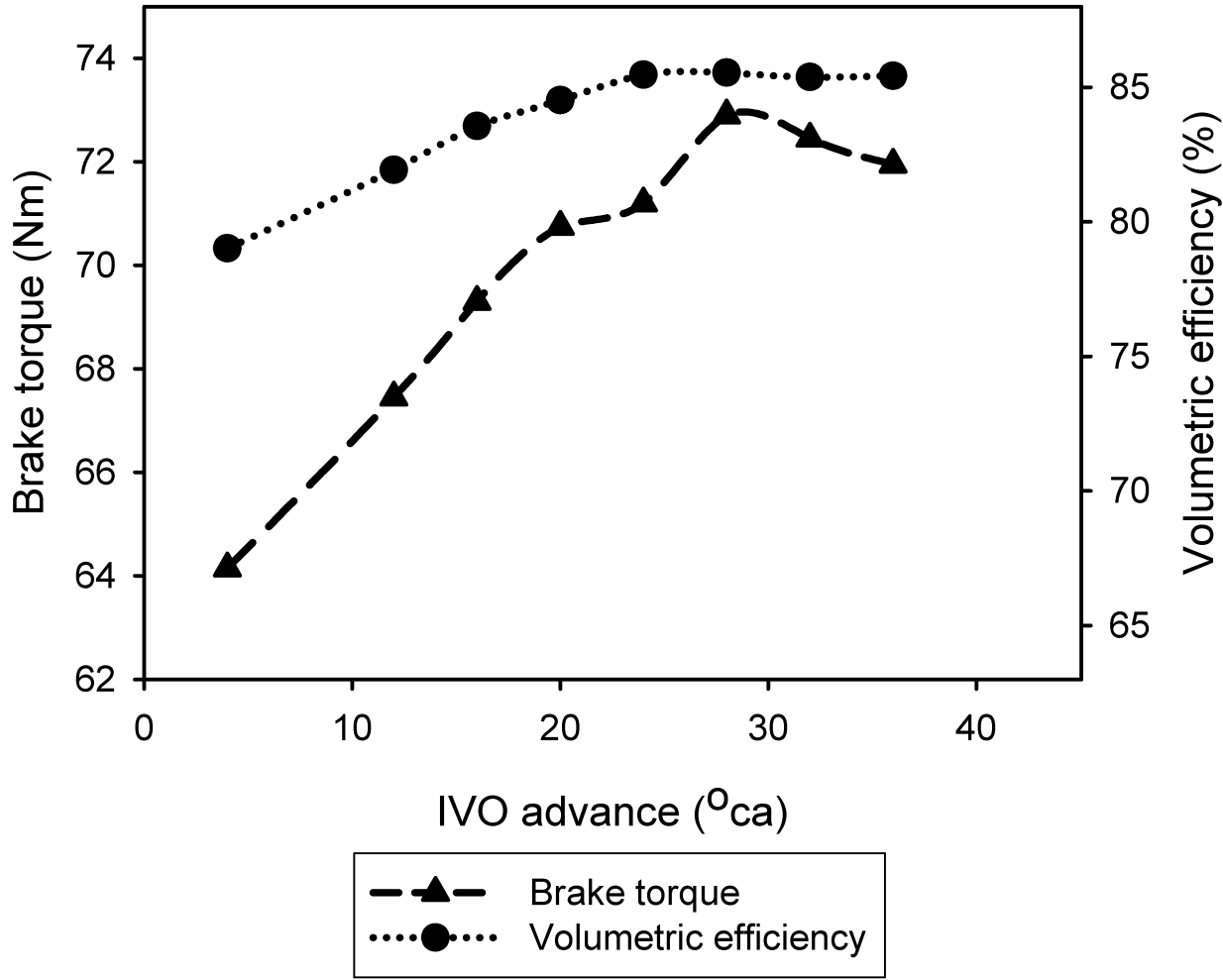
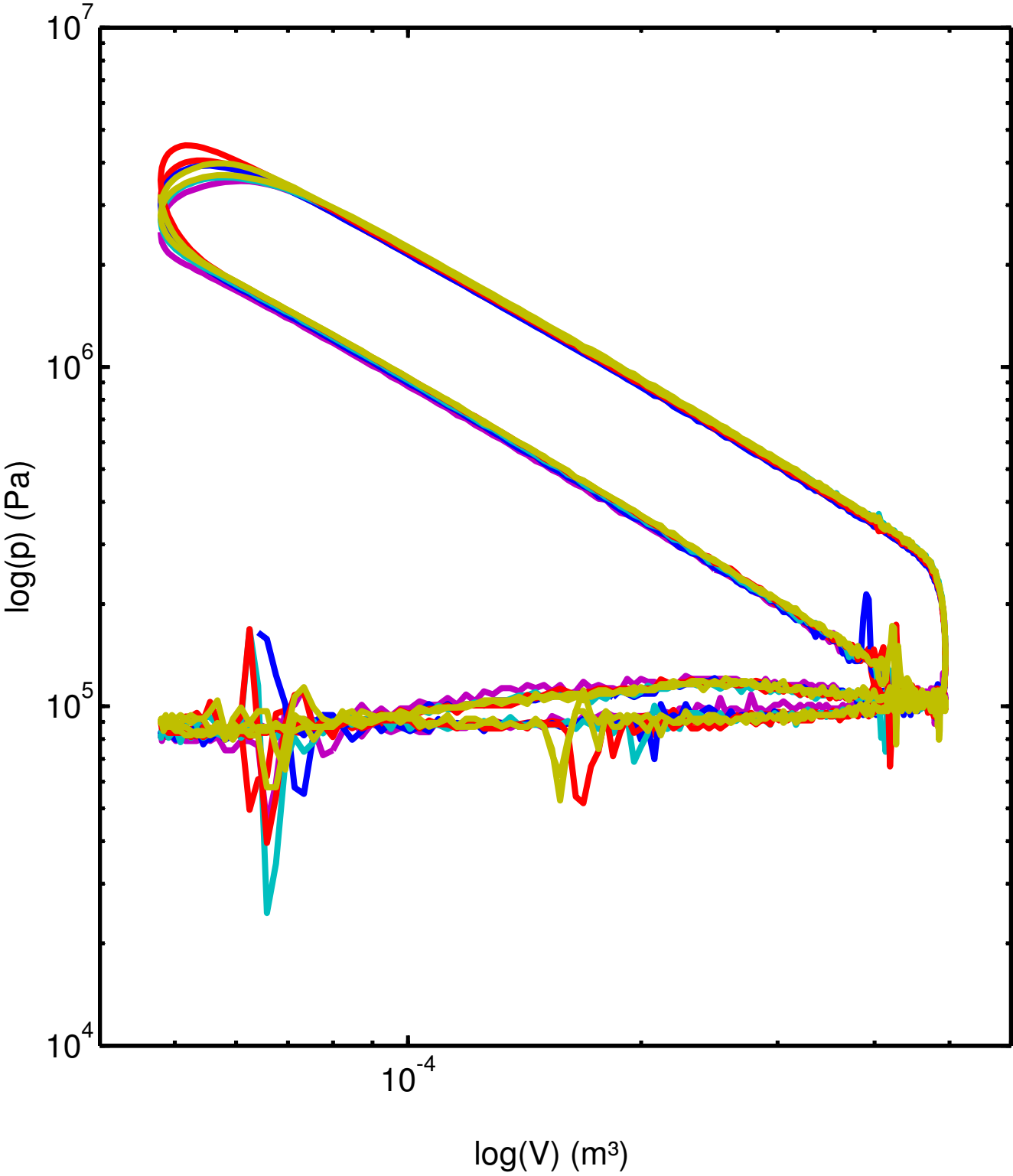


Figure11





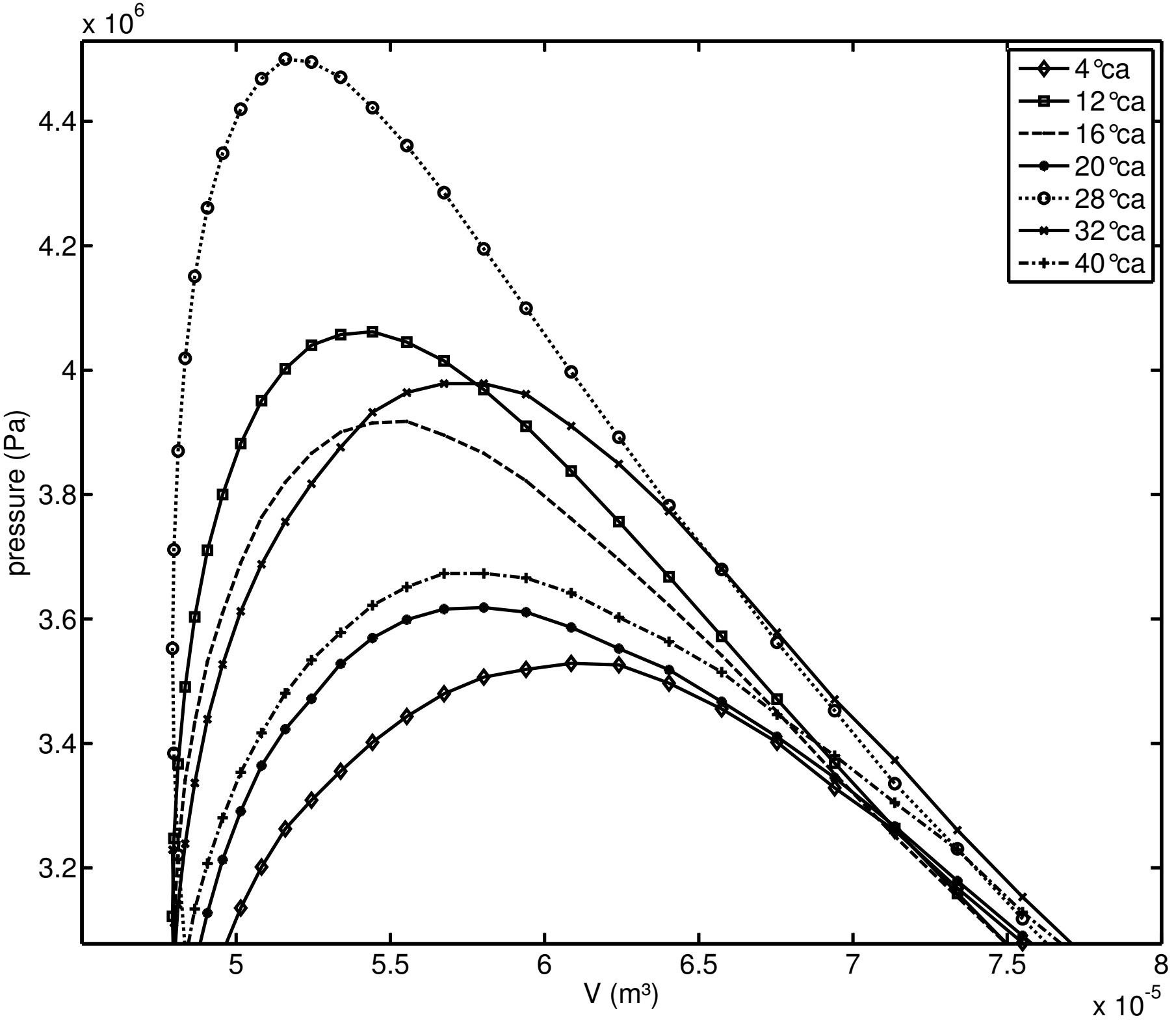




Figure15

