**HEFAT2007**
**5th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics**
**Sun City, South Africa**
**Paper number: GV1**

# ENHANCEMENT POTENTIAL OF THE THERMAL CONVERSION EFFICIENCY OF ICE CYCLES ESPECIALLY FOR USE IN HYBRID VEHICLES

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## ABSTRACT

Most automobile manufacturers have developed hybrid vehicles that combine an internal combustion engine and an electric motor, fusing the advantages of these two power sources. For example Toyota in its Prius II uses a gasoline engine which achieves high efficiency by using a modified Atkinson cycle based on variable valve timing management. This implementation of the Atkinson cycle is not the optimal solution because e.g. some of the air is first sucked from the intake manifold into the cylinder and subsequently returned. This oscillating air stream considerably reduces the thermal conversion efficiency of this cycle.

In this paper the losses in the thermal conversion efficiency of internal combustion engines, and especially of Atkinson cycles, are analyzed in detail and a proposal for an improvement cycle for such applications is made.

## INTRODUCTION

Most automobile manufacturers have been actively developing various new technologies aimed at reducing fuel consumption and diversifying energy sources, which is necessitated by the dwindling supply of petroleum resources. For example, in motive power sources for automobiles alone, they have been continuously improving conventional engines and have developed and commercialized lean-burn gasoline engines, direct injection gasoline engines and common rail direct-injection diesel engines, etc. They have also been modifying internal combustion engines (**ICE**) so that these can use alternative fuels, such as compressed natural gas, instead of gasoline or light oil, and have been installing these engines in commercially available vehicles. Toyota, Honda etc. have also developed and marketed hybrid vehicles that combine an engine and an electric motor, merging the advantages of these two power sources.

In conventional engines, because the volumetric compression and expansion strokes are nearly identical and the cylinder filling is completely, the effective compression ratio and the effective expansion ratio are basically identical as shown in Figure 1, where $p_0$ is the ambient pressure.

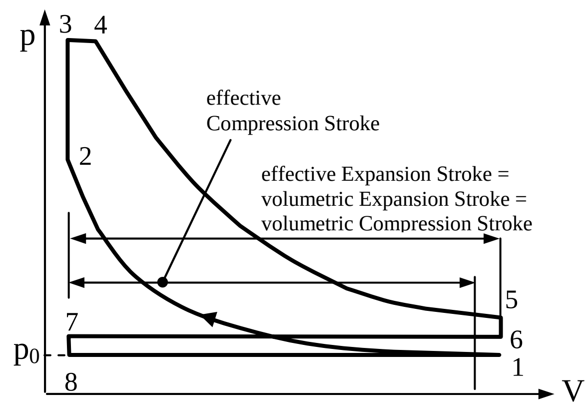


**Figure 1** Schematic Pressure-Volume diagram of a classical four stroke cycle of conventional ICE

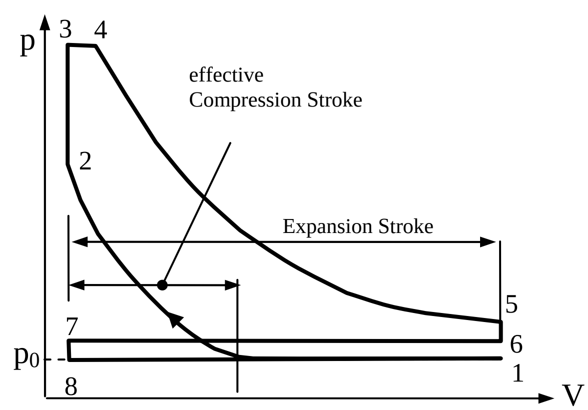


**Figure 2** Schematic Pressure-Volume diagram of a **modified** (i.e. four stroke) Atkinson cycle

Consequently, any attempt to increase the expansion ratio also increases the volumetric compression ratio (computed by means of the volumetric strokes), inevitably resulting in knocking and placing a limit on increases in the expansion ratio.

During the last few years the market share of hybrid vehicles with spark ignition (**SI**) engines has steadily increased. For example Toyota [5] uses in its Prius II a SI engine which tries to achieve high efficiency by using a **modified** (i.e. four stroke) Atkinson cycle as shown in Figure 2 (classical Atkinson cycle has only two strokes). As well known [6], [7] the thermal conversion efficiency increases when the effective compression ratio grows up and/or the effective expansion becomes completed. Because the expansion ratio is increased in comparison to the effective compression ratio and the latter is at a high level, this engine and cycle should theoretically yield higher thermal conversion efficiency.

Usual for realizing a modified Atkinson cycle, the volumetric compression ratio is increased and at the same time the timing of the intake valve closing is delayed. Consequently in the initial stage of the compression stroke (when the piston begins to ascend), some of the air that has entered the cylinder is returned to the intake manifold, in effect delaying the start of compression [5]. In this way, the expansion ratio is increased without increasing the effective compression ratio. Since this method can increase the throttle valve opening time, it can reduce the intake pipe negative pressure during partial load, thus reducing intake loss. Sophisticated variable valve timing is used to carefully adjust the intake valve timing to operating conditions in order to obtain maximum efficiency.

This implementation of the modified Atkinson cycle is not the optimal solution because e.g. some of the air is first sucked from the intake manifold into the cylinder and subsequently returned. Consequently the oscillating air stream considerably reduces the thermal conversion efficiency of the cycle. In the following, this implementation of the Atkinson cycle is analyzed in detail and improvement proposals are made.

## ANALYSIS INSTRUMENTAL

In order to carry out this analysis the entire cycle, i.e. including the gas exchange processes, must be simulated. The simulation program and the engine described in [1], [2], [3] and [4] were used in this investigation. To simulate the combustion process a single-zone model is used with a simple Vibe function for the heat release. The specific heat depends on the temperature and the composition of the gas mixture. With the exception of the gas flow through the valves, all the parts of the cycle are treated as reversible processes.

First the cycle of the engine is shown in Figure 3 at an operating point (here OP 51, full load at 2500 rpm, and see for more information [1] to [4]) with and without heat exchange.

In the part **a** of the Figure 3 the pressure-volume (p,V) diagram is shown during the gas exchange processes, and the valve management times are entered.

In the part **b** of the Figure 3 the temperature - specific entropy (T,s) diagram is shown in which the differences between the real and adiabatic system behavior are clearly visible. A special feature is that this diagram also shows what happens

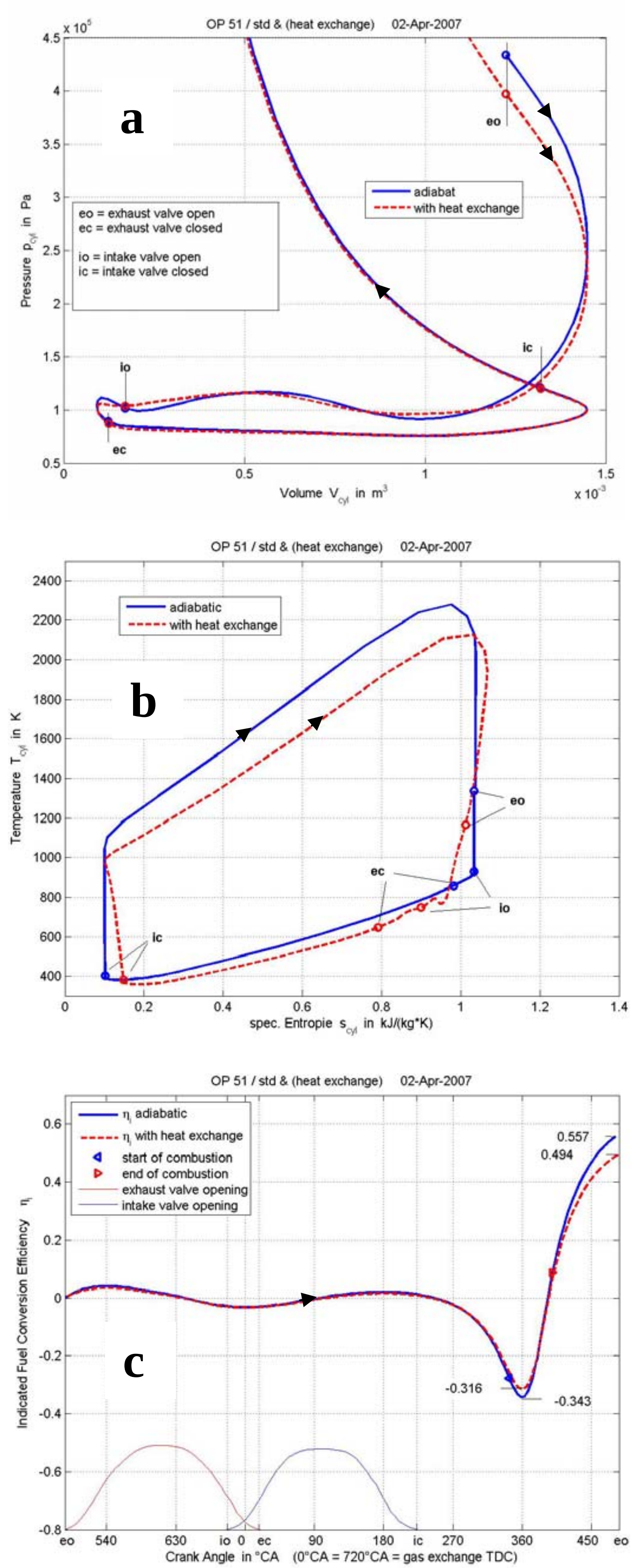


**Figure 3** Comparison between adiabatic (**SV**) and non adiabatic cycles.

during the gas exchange processes (i.e. between the time when the exhaust valve opens "eo" and intake valve closes "ic") which is normally simply omitted in most publications (e.g. [6], [7]) on account of widespread lack of knowledge about its nature. The areas beneath the curves and those within the cycle (as is known) provide information on the thermal conversion efficiency of this cycle. In the case of reversible processes and an adiabatic system is the thermal conversion efficiency equivalent to the indicated fuel conversion efficiency of the cycle.

In the part **c** of the Figure 3, the indicated fuel conversion efficiency is shown above the crank angle. In this case the curve shows that the gas exchange processes in the **SV** do not cause any losses or gains in the efficiency and that only compression, combustion and expansion (i.e. the high pressure processes) have any major influence.

**In order to eliminate the influence of the heat exchange between the following different variants, which is difficult to control, the thermodynamic system is henceforth basically treated as adiabatic. This variant is called the Standard Variant (SV) and used as a comparison for all other variants.**

## APPLICATION OF THE ANALYSIS INSTRUMENTAL ON THE ATKINSON CYCLE

**In the first modified variant** (**1V**) only the intake valve 100° crank angle (°CA) is closed later. This means that not only the filling of the cylinder but also its emptying is strongly influenced (see Figures 4 and 5). Consequently the mass fraction of the waste gases remaining in the cylinder (residual gas fraction) rises to 11.2% in comparison to 9.2% in the standard version. The mass first drawn into the cylinder reaches its maximum at roughly 1.15 grams before partly flowing back into the intake pipe during the compression stroke. Accordingly only just half the mixture mass from the standard version remains in the cylinder.

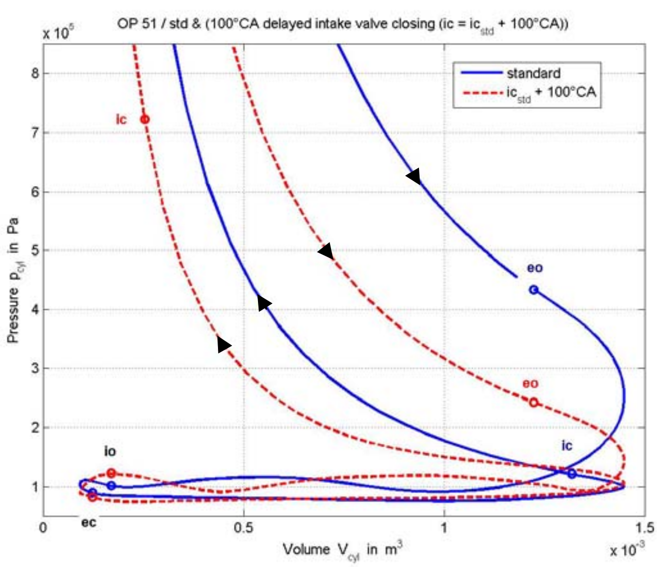


**Figure 4** Pressure-Volume diagram (**SV & 1V**)

Because less fresh charge is sucked in while the residual gas proportion increases, the temperature in the cycle rises, as can be seen in T,s diagram (see Figure 6). The curve of the indicated fuel conversion efficiency in Figure 7 shows that more work is exerted in **1V** for emptying (due to the sluggish expulsion of the exhaust gases) and filling (due to the in and outflow through the intake valve) of the cylinder in comparison to the **SV**. The theoretical advantage of such a cycle (caused by less compression and greater expansion effort) does not therefore apply in this case.

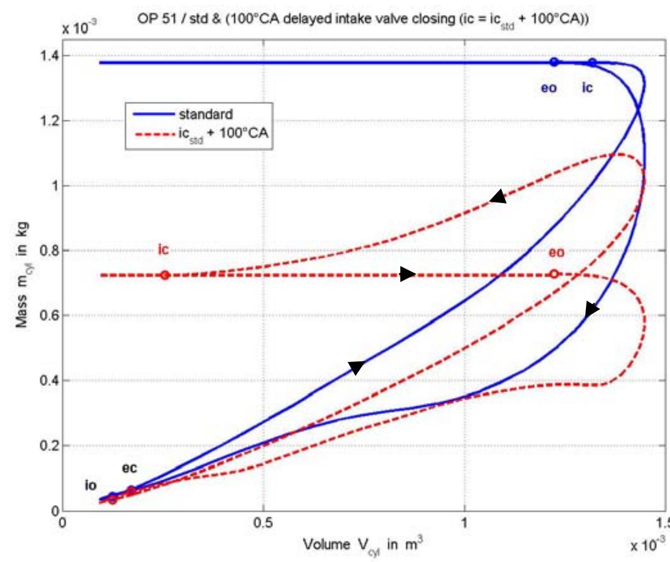


**Figure 5** (gas mixture) Mass-Volume diagram (**SV & 1V**)

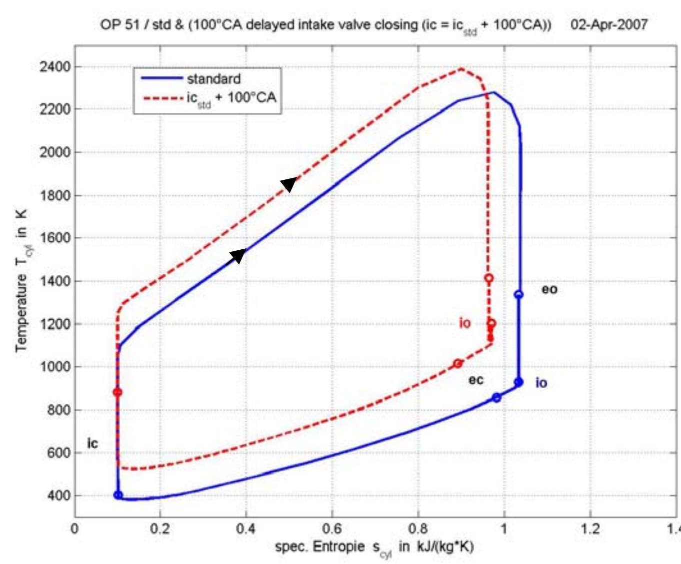


**Figure 6** Temperature–(specific) Entropy diagram (**SV & 1V**)

For this reason - as demonstrated by Toyota in its Prius II - the compression ratio **in the second modified variant (2V)** is increased to improve the thermal conversion efficiency of the cycle. Although the compression ratio in this variant is increased by 90% this does not change much in the curves of the p,V and m,V diagrams (for this reason they are no longer presented here). However, if we compare the T,s diagrams from Figures 6 and 8 we can clearly see the positive effect of increasing the compression ratio (compare the area below the curves and the area within the cycle).

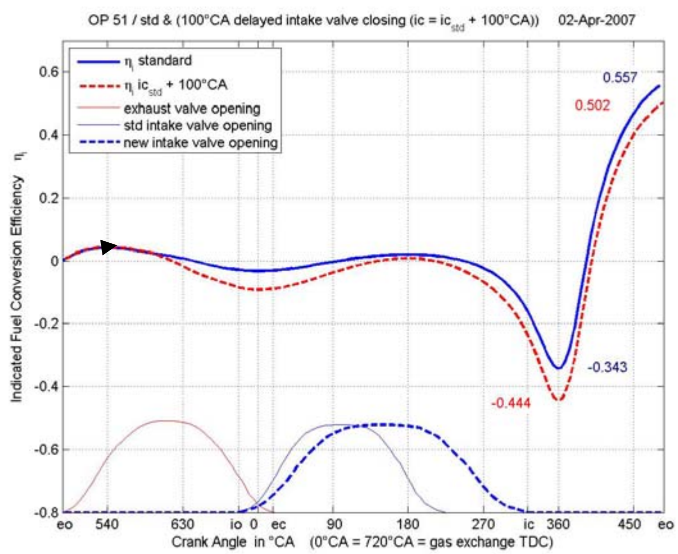


**Figure 7** (indicated fuel conversion) Efficiency–Crank Angle diagram (**SV & 2V**)

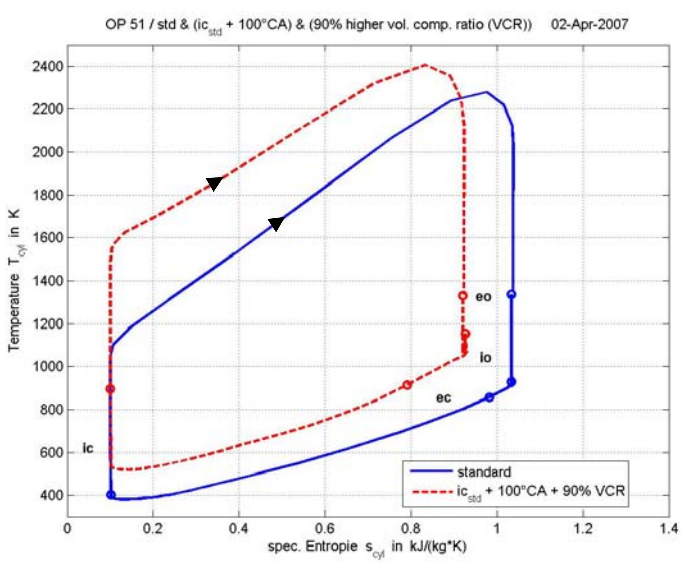


**Figure 8** Temperature–(specific) Entropy diagram (**SV & 2V**)

The curve of the indicated fuel conversion efficiency above the crank angle in Figure 9 shows that although the compression work in the **2V** is greater than in the **1V** (see Figure 7) and therefore it falls to a lower minimum (-0.601), its increase as a consequence of the greater compression ratio has a very positive effect, whereby its level of the **SV** (0.557) is once again reached (see Figure 9).

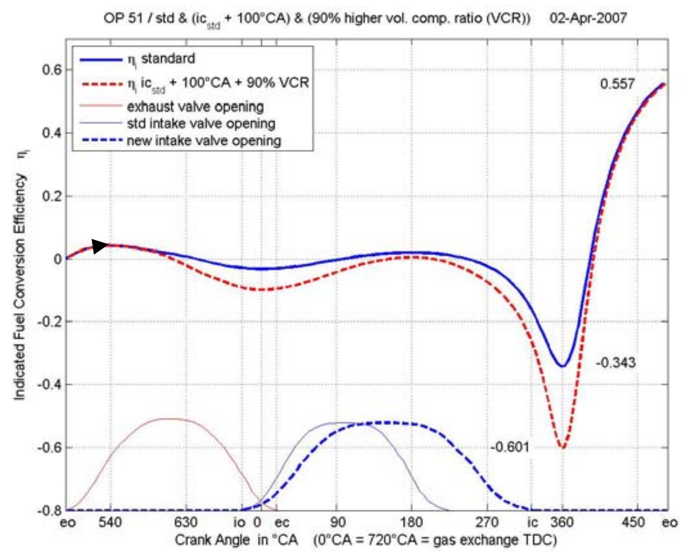


**Figure 9** (indicated fuel conversion) Efficiency–Crank Angle diagram (**SV & 2V**)

This shows, however, that in the case of an aspirated engine in which the intake valve is kept open for a large proportion of the compression, the advantage of performing the cycle using the Atkinson principle is of little benefit for the following reasons:

- The indicated fuel conversion efficiency gain is modest and is largely dependent on the fine tuning of all parameters (valve timing etc.). On the contrary, such variable valve management is highly complex and very expensive.
- The specific power of the engine is too low because of the decreased retained mass of fresh charge in cylinder before compression. This means that a relatively large (due to the large displacement) and therefore heavy engine is needed to power the vehicle.

In order to compensate the power loss caused by the reduction of the fresh charge mass, such an engine should normally be supercharged. In this case, however, the valve timing and also the charge pressure (and therefore the turbocharger) need to be regulated, which would lead to increased complexity in the overall system.

In its Prius II, Toyota has forgone at turbocharging, probably for the following reasons:

- The engine of the hybrid Prius II works almost only at practically full load [5] to achieve maximum efficiency (without throttling and with stoichiometric gas mixture).

- Because the engine is repeatedly turned off and restarted, regulation of the charge pressure and the turbocharger in these conditions - without electrical or mechanical support of the turbocharger - is very difficult to achieve.
- Such a turbocharging method is very complex and would therefore considerably raise the price.

**In the third modified variant (3V)** the valve timing of **SV** and the increased compression ratio of **2V** are used. In order to realize the Atkinson cycle - i.e. shortened compression and extended expansion - a special crankshaft drive is used which permits geometrically different strokes for compression and expansion (see Figure 10). The design of this crankshaft drive is not the subject of this investigation and is therefore not described here.

The aim here is merely to estimate what potential for increasing the indicated fuel conversion efficiency would arise if such a crankshaft drive were used. In other words, **3V** investigates the extent to which the losses caused by the suction and partial expulsion of the fresh charge reduce the indicated fuel conversion efficiency.

As before, a comparison is once again made with the **SV**. Because in the modified crankshaft drive the angle positions of the top dead center (**TDC**) in comparison to the **SV** are slightly shifted, the closing angle of the outlet valve in **3V** 20°CA is also delayed to avoid causing any large counter pressure during the valve overlap time.

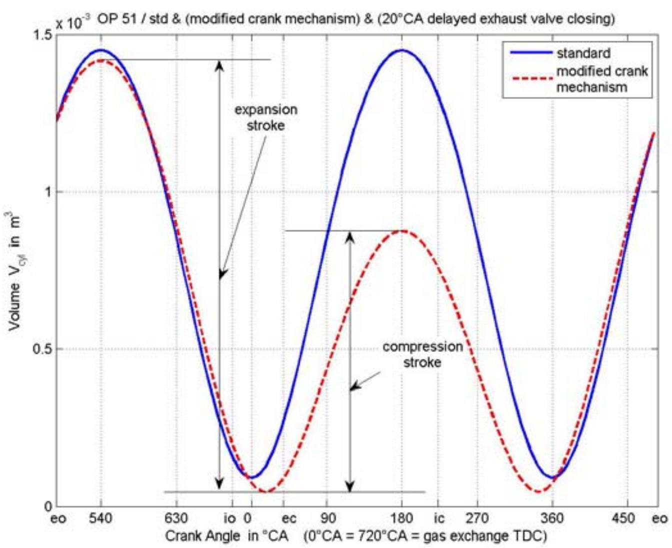


**Figure 10** Volume-Crank Angle diagram (**SV & 3V**)

The p,V diagram is shown in Figure 11, in which both the different compression and expansion strokes and also the increased compression ratio can clearly be seen in the modified crankshaft drive.

An analysis of the T,s diagram in Figure 12 and comparison with that in Figure 8 reveals that the thermal conversion efficiency is better in this case than it would be in **SV** and **2V**. The only factor which could have contributed to this is the elimination of the streaming back and forth through the intake valve, as no other changes or parameter optimizations in comparison to **2V** were made.

The curve of the cylinder gas mass in Figure 13 reinforces this assumption as the entire gas mass sucked in remains in the cylinder for the combustion. Although the compression stroke is much shorter than in **2V** and the intake valve is opened for a shorter time, the mass sucked in **3V** is roughly 6 % greater than in **2V**. All these factors hint that the indicated fuel conversion efficiency in **3V** would be greater than in all the variants presented previously (see Figure 14).

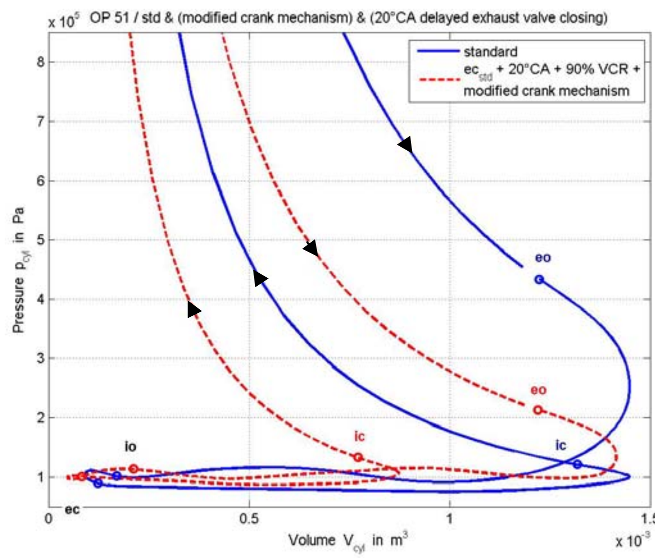


**Figure 11** Pressure–Volume diagram (**SV & 3V**)

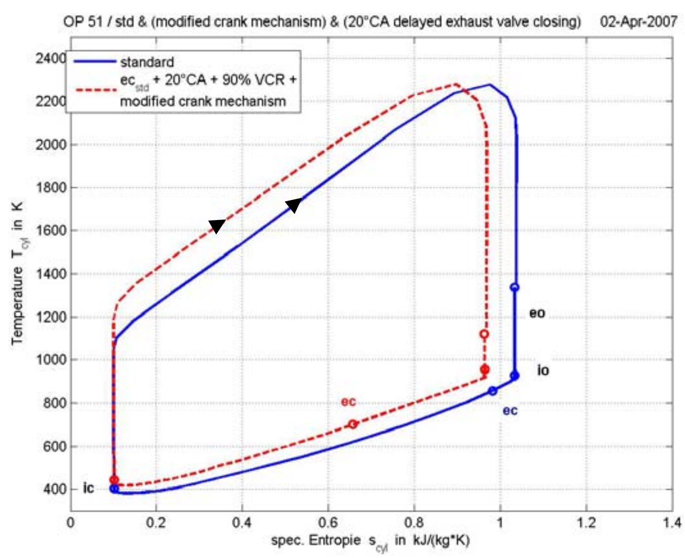


**Figure 12** Temperature–(specific) Entropy diagram (**SV & 3V**)

Analysis of the curves in Figure 14 shows that more effort needs to be exerted in **3V** to void the cylinder in comparison to **SV**. This can be explained by the low pressure level in the cylinder before the outlet valve is opened (see Figure 11), with the exhaust gases only flowing out very sluggishly as a result. After the intake valve has closed and the compression has started, the indicated fuel conversion efficiency drops again sharply as a result of the increased compression ratio, but less sharply than in the case of **2V**. A higher value of this efficiency is therefore achieved after compression and expansion than in the **SV**.

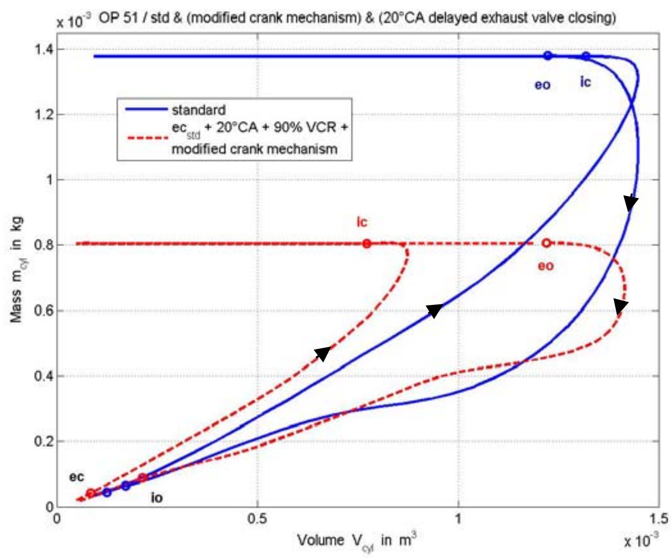


**Figure 13** (gas mixture) Mass-Crank Angle diagram (**SV & 3V**)

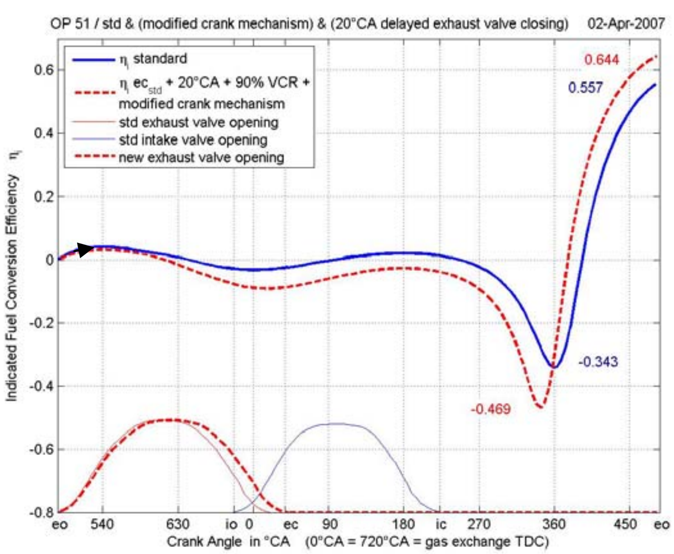


**Figure 14** (indicated fuel conversion) Efficiency–Crank Angle diagram (**SV & 3V**)

## CONCLUSION

The potential of the Atkinson cycle realized through extended opening of the intake valve - as implemented by Toyota in its Prius II - was investigated here in a number of different variants. Analysis of the simulation results shows that the benefits in the form of increased efficiency would be minimal in this kind of cycle.

As an additional variant, the use of a new kind of crankshaft drive was presented which permits different size strokes for compression and expansion. In this case the improvements in the efficiency are clearly visible, even without supercharging of the engine.

## REFERENCES


[1] Gheorghiu V., *Higher Accuracy through Combining of quasi-3D (instead of 1D) with true-3D Manifold Flow Models during the Simulation of ICE Gas Exchange Processes*, SAE-Paper 2001-01-1913, SAE-Congress, 2001, Orlando, Florida, USA, (pdf, 2.4 KB)

[2] Gheorghiu V., *Quasi-3D Simulation of Compressible Unsteady Flows*, HEFAT 2003, 2nd International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics, Paper number: GV1, 2003, Victoria Falls, Zambia, (pdf, 816 KB)

[3] Gheorghiu V., *Simulation Results of Compressible Unsteady Flows Using the quasi 3D-Method*, ESFA Congress (FISITA affiliation), 2003, Bucharest, Romania, (pdf, 2.72 MB)

[4] Gheorghiu V., *Quasi-3D Simulation of Compressible Unsteady Flows Through ICE-Manifolds*, Paper Number F2004F427, FISITA Congress, 2004, Barcelona, Spanien (pdf, 3.03 MB)

[5] Toyota Motor Corporation, *Toyota Hybrid System THS II*

[6] Heywood J. B., *Internal Combustion Engine Fundamentals*, MacGraw-Hill Book Company, 1988

[7] Pischinger A., Kraßnig G., Taucar G. Sams Th., *Thermodynamic of Internal Combustion Engines* (German), Springer-Verlag, Wien New York, 1989