

Novel Crankshaft Mechanism and Regenerative Braking System to Improve the Fuel Economy of Light Duty Vehicles and Passenger Cars

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ABSTRACT

Improvements of vehicle fuel economy may be achieved by the introduction of advanced internal combustion engines (ICE) improving the fuel conversion efficiency of the engine and of advanced power trains (PWT) reducing the amount of fuel energy needed to power the vehicle. The paper presents a novel design of a variable compression ratio advanced spark ignition engine that also permits an expansion ratio that may differ from the compression ratio hence generating an Atkinson cycle effect. The stroke ratio and the ratio of maximum to minimum in-cylinder volumes may change with load and speed to provide the best fuel conversion efficiency. The variable ratio of maximum to minimum in-cylinder volumes also improves the full load torque output of the engine. The paper also presents an evolved mechanical kinetic energy recovery system delivering better round trip efficiencies with a design tailored to store a smaller quantity of energy over a reduced time frame with a non-driveline configuration. Simulations show an improvement of full load torque output and fuel conversion efficiency. Brake specific fuel consumption maps are computed for a gasoline engine 2 liters, in-line four, turbocharged and directly fuel injected showings significant fuel savings during light and medium loads operation. Results of vehicle driving cycle simulations are presented for a full size car equipped with the 2 liters turbo GDI engine and a compact car with a downsized 1 liter turbo GDI engine. These results show dramatic improvements of fuel economies for similar to Diesel fuel energy usage and CO₂ production. The turbo GDI engines with the alternative crank trains offer better than hybrids fuel economies if the vehicles are also equipped with the mechanical kinetic energy recovery system (KERS) recovering the braking energy to reduce the thermal energy supply in the following acceleration of a driving cycle.

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INTRODUCTION

Direct fuel injection (DI), turbo charging (TC) and downsizing are the major features in stoichiometric gasoline engines currently being adopted to dramatically improve the fuel economy during driving cycles and therefore largely reduce the emissions of carbon dioxide. Even with these features, stoichiometric gasoline engines still have lower top and part load fuel efficiencies in comparison with higher compression ratio, lean burning higher power density diesel engines able to be controlled throttle-less by quantity of fuel injected. Furthermore, stoichiometric gasoline engines have very low air pollutant emissions thanks to the well-established three-way catalyst technology to meet future

emission regulations much easier than Diesel [1 and 2]. Major areas of development are variable valve actuation (VVA), exhaust gas recirculation (EGR), variable compression ratio (VCR), and alternative power cycles as well as crank train mechanisms all to boost the fuel conversion efficiency. The aim of this paper is to introduce a new crank train mechanism enabling variable compression ratio and variable Atkinson effect applied to one of the latest turbo GDI engine.

The Atkinson cycle engine is basically an engine permitting the strokes to be of different lengths. In the Atkinson cycle, the power and exhaust strokes are longer than the intake and compression strokes and should result in better fuel efficiency. Atkinson designed more than one engine to

capitalize on this important property. The original Atkinson cycle engine allowed the intake, compression, power, and exhaust strokes of the four-stroke cycle to occur in a single turn of the crankshaft and was designed to avoid infringing patents covering Otto cycle engines. Because of the unique crankshaft design of the Atkinson engine, its expansion stroke can differ from its compression stroke and, with a power stroke longer than its compression stroke the engine can achieve larger thermal efficiency than a traditional engine. The disadvantage of the four-stroke Atkinson cycle engine versus the more common Otto cycle engine is reduced power density. Because a smaller portion of the compression stroke is devoted to compressing the intake charge, an Atkinson cycle engine does not take in as much charge as a similar Otto cycle engine would do. Although the Atkinson's original design is merely a historical curiosity, many modern engines use unconventional valve timing to produce the effect of a shorter compression stroke/longer power stroke, but a few solutions have also been proposed with a different means of linking the piston to the crankshaft [3,4,5,6,7,8,9,10].

Variable compression ratio is the technology to adjust internal combustion engine cylinder compression ratios to increase fuel efficiency under varying loads [11,12,13,14,15,16,17,18,19,20,21,22,23,24,25]. Higher loads require lower ratios to be more efficient and vice versa. Variable compression ratio engines allow for the volume above the piston at top dead centre to be changed. For automotive use this needs to be done dynamically in response to the load and driving demands. Output of spark ignition engines is controlled by the quantity of mixture introduced within the cylinder. Occurrence of knock limits the values of compression ratio and boost pressures that can be adopted in turbocharged spark ignition engines or the compression ratio that may be used in naturally aspirated spark ignition engines. If the compression ratio is set to avoid knock in the most unsafe wide open throttle operating points, then especially under light loading, the engine can lack torque. The solution is to be able to vary the compression ratio to suit the reduced amount of charge introduced within the cylinder. In all the variable compression ratio systems, the basic advantage is the one that may follow the increased indicated efficiency of the cycle. Small changes in absolute values of efficiency may produce large percentage increases in fuel economy especially at very low loads. Coupling of variable compression ratio to turbo charging may increase the options to achieve better fuel economies through synergies with downsizing and pressure boost.

In the present approach, a new mechanism is defined to change not only the compression ratio but also the length of intake, compression, expansion and exhaust strokes as in Atkinson cycles. This system is employed to improve the light load efficiency as well as to boost the full load output and efficiency seeking operation in every point with same margins to knock and switching to expansion strokes longer than compression strokes when more efficient at light loads.

The system is fitted on an in-line four cylinder turbocharged gasoline engine with direct fuel injection.

The idea of using a link of the piston to the crankshaft enabling both variable compression ratio and variable ratio of the expansion to the compression stroke is a novelty. It also offers the opportunity to use the technique to improve the fuel economy over the full range of loads and speed and boost the power output. In the exercise we have selected a maximum CR of 15.5:1 and a minimum CR of 9.5:1. The combustion chamber shape is optimized for the 15.5:1 CR, and the lower CR of 9.5:1 is obtained by increasing the piston to head clearances. The lowest CR operating conditions therefore may suffer from less than optimum combustion chamber shape but minimised due to the introduction of turbo boost. The minimum CR can be further reduced to 7 or 8:1 if higher boost pressures are desired to downsize the engine further. With regards to the Atkinson cycle effect, the lengths of the expansion, exhaust, intake and compression strokes differ. The TDC piston positions at the start of the expansion and the intake strokes, as well as the BDC positions at the start of the exhaust and compression strokes also differ. The in-cylinder volumes in the two TDC and in the two BDC are therefore also different. The CR ratio when the Atkinson cycle is enabled refers to the ratio in between the maximum in-cylinder volume at the end of the expansion stroke and the minimum in-cylinder volume at the beginning of the expansion stroke. We consider here an exhaust to intake stroke ratio 1 (no Atkinson) to 1.05. Because of the variability of the stroke, the two TDC and the two BDC volumes differ. Therefore, CR with Atkinson is conventionally defined as the ratio of maximum to minimum in-cylinder volume over the two strokes.

Regenerative braking is finally considered to reduce the thermal engine energy supply when covering driving cycles characterized by accelerations following decelerations, where part of the propulsive power needed to accelerate the car may be the braking energy recovered with a regenerative device. Mechanical regenerative systems are much more efficient than electric regenerative systems permitting almost double the round trip efficiency wheels to wheels [38,39,40,41,42,43,44]. The first generation of mechanical kinetic energy recovery systems (KERS) was delivering round trip efficiencies of 70%. Today's second generation of mechanical KERS is designed to deliver round trip efficiencies of almost 80% thanks to the better attention to the power and energy storage capacity and timing of passenger car applications and the focus on improving the efficiency of the constant variable transmission that was the weak part of the first generation KERS design as well as on the off driveline configuration [45].

MECHANISM FOR VARIABLE COMPRESSION AND STROKE RATIOS

The mechanism is comprised of a four bar mechanism that connects the piston to the crankshaft via an oscillating member, positioned on the opposite side of the cylinder relative to the crankshaft. The oscillating member is supported on an eccentric shaft that rotates at half shaft speed preferably in the same direction as the crankshaft, and driven by the crankshaft. A phasing mechanism between the crankshaft and the eccentric shaft allows relative angular motion between the two shafts. This relative motion allows the compression ratio to be altered continuously between two set limits. The size of the eccentric and angular movement of the control shaft determines the extent of the CR adjustment. The maximum angular phasing is approximately 50 degrees and is determined by the control shaft eccentric offset generally not more than 5 mm.

Another feature of the mechanism is that it has an inherent Atkinson cycle phase as well as an exhaust gas retention phase, at various settings of the CR adjustment. [Figure 1](#) shows the piston in the low and high CR positions at TDC and BDC positions. For a typical small engine in this position, when the engine is at a wider throttle open condition and under heavier loads, the induction stroke may be a little shorter than the compression stroke (by about 2%) and this may produce a small supercharging effect. In addition, the induction stroke is shorter than the power stroke and this may produce an Atkinson effect (by about 11%).

Also there is no exhaust gas retention. In the high CR position, when the engine is at lower throttle settings and under lower loads, the induction stroke may be shorter than the compression stroke (by about 6%) and this may produce a reasonable supercharging effect as well as the Atkinson effect (by about 11%). Also there is a small gas retention feature. The mechanism can be configured in all inline engines and all V type engines by having a control shaft for each bank of cylinders in a back to back arrangement with a common crankshaft. By varying the extent of the eccentric and the geometric position of the eccentric shaft, a number of different characteristics can be achieved. For example, an Atkinson effect can be achieved in both low and high CR of up to 20% and 10% respectively.

It is to be noted that all connections are pin jointed and use known conventional technologies and production techniques. The use of a forked piston con-rod allows the main con-rod to be received in the cavity. Initial dynamic simulation studies indicate that the inertia and combustion loads on the various connections are within the bearing pressure limits of conventional bearing materials. It has also been shown that the engine can be balanced to acceptable levels without the need of balance shafts.

The baseline VCR and Atkinson engine has a compression ratio CR of 9.5:1. This is the selected lower

limit in the Variable Compression Ratio mechanism. For the upper limit, a CR of 15.5:1 is selected to provide an effective higher compression ratio at low engine speed and partial loads when the turbo is less effective. The mechanism of the baseline engine can be designed to produce a much wider CR limits if desired and will depend on the maximum boost pressures that can be accommodated in the engine design and the extent of downsizing. The VCR allows for the "effective" compression ratio to remain relatively constant throughout the engine load and speed and to prevent knock. At the higher speed and loads the turbocharger may require by-passing via the waste gate in the normal manner. This arrangement would deliver the optimum engine brake thermal efficiency and maximize the torque and power output. The VCR engine is expected to be operated with a high "mechanical" CR at low to medium speed and load therefore the squish effect is at its highest. At the higher load and speeds the CR is reduced to allow the turbocharger to take effect and thus the squish is reduced, however, the effective CR is maintained at the optimum level to prevent knock, and we do not expect any emission penalties. The response to changes in compression ratio is similar to the response to the variable valve timing mechanism in modern engines; however this is a subject that needs to be further investigated. The virtual elimination of piston side thrust due to the small piston connecting rod angular movement will reduce piston friction and may well compensate for any friction caused by the additional pin connections. This will also need to be further investigated.

With reference to the Atkinson cycle effect, the lengths of the expansion, exhaust, intake and compression strokes differ. The TDC piston positions at the start of the expansion and the intake strokes, as well as the BDC positions at the start of the exhaust and compression strokes also differ. The in-cylinder volumes in the two TDC and in the two BDC are therefore also different. The CR ratio when the Atkinson cycle is enabled refers to the ratio in between the maximum in-cylinder volume at the end of the expansion stroke and the minimum in-cylinder volume at the beginning of the expansion stroke. We consider here an exhaust to intake stroke ratio 1 (no Atkinson) to 1.05. Because of the variability of the stroke, the two TDC and the two BDC volumes differ. Therefore, CR with Atkinson is conventionally defined as the ratio of maximum to minimum in-cylinder volume over the two strokes.

The mechanism will incur additional costs due to its complexity, weight and packaging. The reliability and transient response will need to be investigated. It is obvious that the mechanism is more complex than the conventional piston-connecting rod-crank arrangement, however all of the connections use pressure lubricated conventional pin joints and the eccentric shaft is hydrodynamically lubricated. The reliability should not be any different than a conventional engine given that the bearing pressures can be designed to be within the limits of conventional bearing materials, taking into consideration, the higher inertia loads produced by the mechanism. This will also have a limitation on the maximum

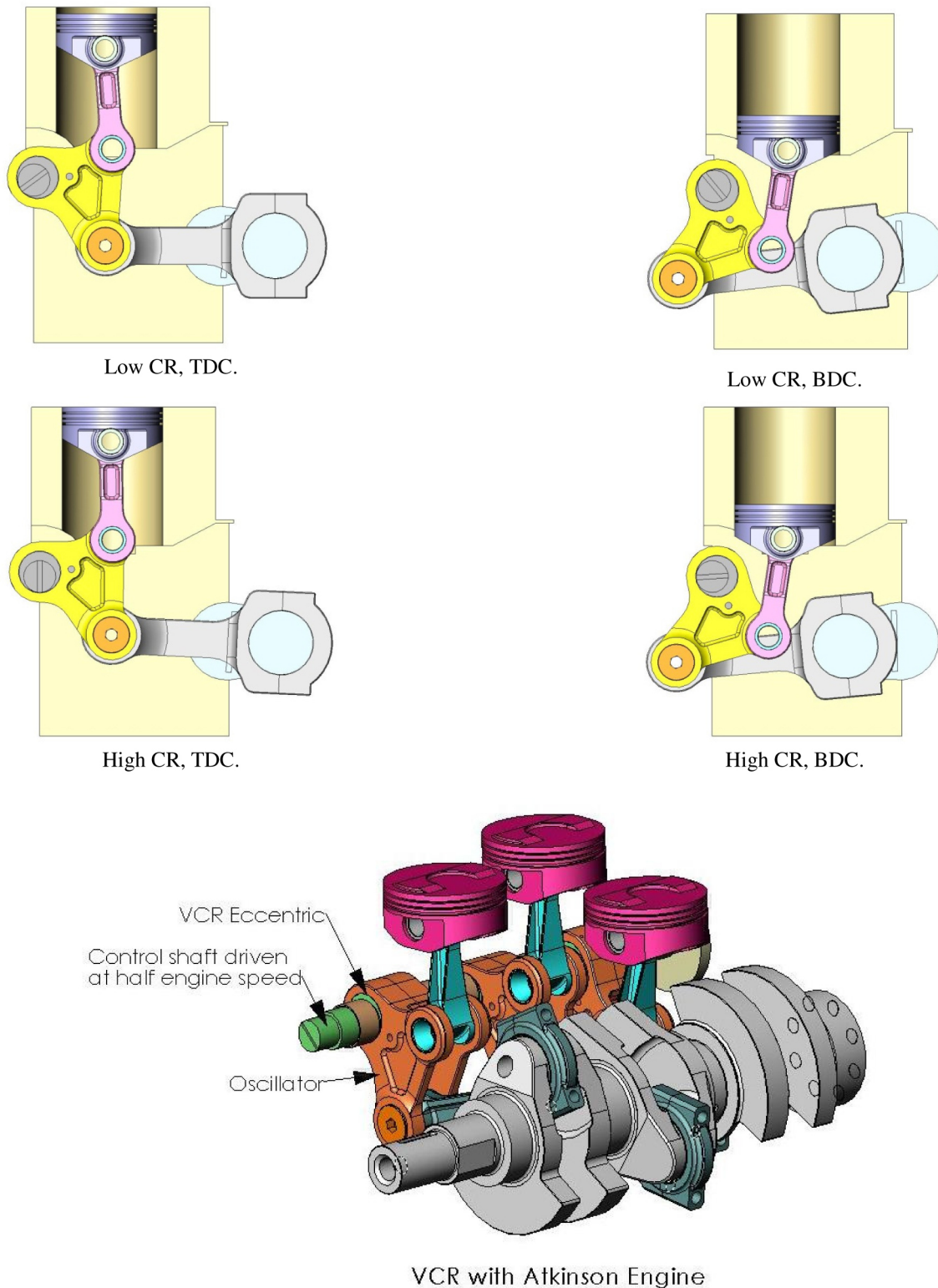


Figure 1. Low and high CR mechanism at TDC and BDC and 3D view of the mechanism in a three cylinder engine.

engine speed, however engines adopting this approach, are designed for economy rather than very high speed. For the larger engines the speed factor is generally not an issue as the mechanism can be designed to operate effectively up to 5000

rpm. In comparison with other known mechanisms used for VCR, this mechanism offers simplicity and compactness. The components are simple to manufacture in mass production and should not add more than 5% to the cost of the overall

engine, and similarly to the overall weight. This additional cost is small compared to the fuel savings and can be recouped in a very short time by the customer. For in-line engines, the packaging is similar to a conventional engine except for a small crankshaft offset that can be a benefit to front drive vehicles. For V type engines the packaging is very similar to a conventional engine and should not create any difficulties. It is to be noted that patent protection has been applied for the nominated mechanism in all the major vehicle producing countries.

While the variable compression ratio is absolutely not a novelty, the way to achieve this variability and the control of this feature linked to a knock sensor to operate all over the range of engine speeds and loads as close as possible to about same margin to knock is certainly an interesting new option offered by this paper.

ENGINE RESULTS FOR A 2 LITRES TURBO GDI

Engine performance simulations have been performed with the GT-SUITE code [26] for a 2 Liter in-line four cylinder turbo charged gasoline engine. The engine geometry (baseline) is presented in Table 1. For the evaluation of benefits of the alternative crank train mechanisms proposed here, there is no reason to modify the design of the intake to follow the variable compression ratio or the variable compression ratio and stroke ratio. Further refinements of the design of the engine with alternative crank train mechanisms are certainly possible in both the intake and the exhaust systems including pipe lengths and diameters as well as valve lift profiles. These refinements may certainly further improve the brake fuel conversion efficiency as well as the torque output.

A modified form of the Chen-Flynn correlation is used to model friction [26 and 27]. The correlation has a constant term (for accessory friction), a term which varies with peak cylinder pressure, a third term linearly dependent on mean piston velocity (for hydrodynamic friction) and a fourth term quadratic with mean piston velocity (for windage losses). Values used for the operation with all the cylinders are presented in Table 2. A_{cf} is the constant portion of the Chen-Flynn friction correlation, B_{cf} the term which varies linearly

with peak cylinder pressure, C_{cf} the term accounting for hydrodynamic friction in the power cylinder which varies linearly with the piston speed and Q_{cf} the term which varies quadratically with the piston speed and accounts for windage losses in the power cylinder. The equation used to calculate friction is given below:

$$FMEP = A_{cf} + \frac{1}{ncyl} \sum_{i=1}^{ncyl} [B_{cf} \cdot (P_{cyl})_i + C_{cf} \cdot (S_{fact})_i + Q_{cf} \cdot (S_{fact})_i^2] \quad (1)$$

with: $S_{fact} = 1/2 \cdot RPM \cdot S$, A_{cf} , B_{cf} , C_{cf} and Q_{cf} user inputs, P_{max} the maximum cylinder pressure, RPM the cycle-average engine speed and S the cylinder stroke. Each cylinder has its own contribution to the total engine friction based upon its own maximum cylinder pressure and stroke (folded into the speed factor, S_{fact}).

The proposed parameters for the FMEP correlations are those following fit of experimental data from measured BMEP and IMEP (output of torque meter and pressure cycles recorded through an in-cylinder pressure sensor) of the GDI turbo engine with a traditional crank train. Despite the new crank train it is not expected to have FMEP values differing too much from those defined above, the subject obviously need further investigation, with assembly of an engine now planned for future tests. Only these experiments may provide a better estimation of the FMEP vs. speed and load of the new engine with a novel crank train mechanism never tested before.

Combustion is modeled using a non-predictive Wiebe combustion model [26] with the look up table from the baseline configuration modified by using the predictive Spark Ignition turbulent combustion model [26] simulations. Simulations were performed with either (1) a non-predictive Wiebe function combustion model [26] with angles for 50% mass fuel burned and angles for 10% to 90% combustion tabled vs. engine speed and BMEP from data measured for a GDI turbo engine and a traditional crank train, and (2) a predictive Spark Ignition turbulent combustion model [26] calibrated to reproduce these same experimental data. These two models are very well known models adopted in engine performance simulations over several decades with details

Table 1. 2 Litre in-line four gasoline engine geometry

displacement per cylinder [litres]	0.4995	no. of intake valve per cylinder	2
number of cylinders	4	Intake valve dia. [mm]	34.5
engine layout	L-4	Intake valve max. lift [mm]	10.05
compression ratio	10	IVO [deg]	358(-2)
bore [mm]	86	IVC [deg]	619 (+79)
stroke [mm]	86	No. of exhaust valve per cylinder	2
connecting rod length [mm]	143	Exhaust valve dia. [mm]	31
wrist pin offset [mm]	0	Exhaust valve max. lift [mm]	10
clearance vol. [litres]	0.0550	EVO [deg]	131 (-49)
engine type	S.I.	EVC [deg]	384 (+24)

Table 2. 2 Litres in-line four gasoline engine friction correlation parameters

Engine operation	L-4
A_{cf} [bar]	0.3
B_{cf}	0.006
C_{cf} [bar·s/m]	0.09
Q_{cf} [bar·s ² /m ²]	0

eventually available in [26]. This latter predictive model properly calibrated was then used to evaluate the combustion performances with variable spark timing for all the engine solutions considered with the novel crank trains. Knock limited maximum brake torque spark timings were obtained performing simulations for different values of the spark advance and monitoring torque and knock index to select the best spark advance within the constraints. It is worth mentioning that in consideration of the fact the predictive spark ignition turbulent combustion model may fail in some points of a full engine map, we decided to run the final simulations with a non-predictive Wiebe function combustion model [26] having angles for 50% mass fuel burned and angles for 10% to 90% combustion tabled vs. engine speed and BMEP from data measured for the GDI turbo engine and a traditional crank train corrected for the results of the predictive spark ignition turbulent combustion model for the actual engine crank trains simulated.

The simple knock model is based on the Douaud and Eyzat induction time correlation [26 and 28]. The induction time (ignition delay) in seconds is calculated at every time step using the equation:

$$\tau = \frac{0.01869}{A_p} \cdot \left(\frac{ON}{100} \right)^{3.4107} \cdot P^{-1.7} \cdot \exp \left(\frac{3800}{A_T \cdot T} \right) \quad (2)$$

Where A_p is a user-entered pre-exponential multiplier, ON the user-entered fuel octane number (RON 95 in the particular application to gasoline), P the cylinder pressure [kgf/cm²], A_T the user-entered activation temperature multiplier and T the unburned gas temperature [K]. In general, this induction time continually decreases as combustion progresses and the unburned zone temperature rises. The end-gas auto-ignites (knocks) if the induction time is less than the flame arrival time. The model assumes that auto-ignition occurs when:

$$\int_0^i \frac{d\tau}{\tau} = 1 \quad (3)$$

t_0 is the start of end-gas compression, t_i the time of auto-ignition and t the induction time. This model is used just for a first computational assessment of the advantages of the knock controlled variable compression ratio operation. The model represents major factor affecting knock. At high compression ratios, even before spark ignition, the fuel-air mixture may be

compressed to a high pressure and temperature which promotes auto ignition. At low engine speeds the flame velocity is slow and thus the burn time is long, and this results in more time for auto ignition. At high engine speeds, there is less heat loss so the unburned gas temperature is higher which again promotes auto ignition. All these competing effects are included in the Douaud and Eyzat induction time correlation [28] despite their description is quite simplified.

Modeling of turbo charged engines is not a novelty [26] and the baseline turbo charged GDI engine was modeled as usual introducing the measured maps of turbine and compressor, the one of turbine given as reduced speed, reduced mass flow rate, pressure ratio and efficiency, and the one of the compressor given as actual speed and mass flow rate, pressure ratio and efficiency. The waste gate opening was adjusted to achieve stable operation within the mapped area and the measured pressure out of the compressor of the baseline turbo charged GDI engine. With alternative crank trains, the same turbocharger was used and the waste gate opening was adjusted to get the maximum boost pressure compatible with stable operation within the mapped area. The optimization of boost pressure and compression ratio for maximizing the brake fuel conversion efficiency and the brake torque output within the knock limits has not been performed at this stage because the model of the FMEP with the alternative crank train mechanisms need first to be improved to make this latter optimization meaningful. The scope of the paper is to perform a first assessment of the performance increases of the novel crank train mechanisms and not to compute the optimum compression ratio, direct injector injection profile, spark advance and waste gate opening of the turbocharger coupled to the modified crank trains.

The compression ratio was selected to be the largest possible within the limits of about the same knock resistance of the baseline GDI turbocharged engine in the worst (for knock) operating point. The CR=15.5 value of the maximum compression ratio has been selected conservatively, and the largest values may possibly follow further refinements of the model. However, the relative benefits in terms of efficiency and torque output reduce further increasing the compression ratio. With regards to the Atkinson effect, the selection of the stroke ratio is even more arbitrary than the selection of the maximum compression ratio described above, and a further optimized stroke ratio may allow a better consideration of the compelling efficiency benefits. Every user of GT-POWER