23rdABCM International Congress of Mechanical Engineering
December 6-11, 2015, Rio de Janeiro, RJ, Brazil

APPLICATION OF ATKINSON / MILLER CYCLE ON A ROTARY INTERNAL COMBUSTION ENGINE THE XXII COBEM

Alexandre Zuquete Guarato
Epifanio M. Ticona
Sergio Leal Braga
Pontifical Catholic University of Rio de Janeiro (PUC-Rio), Rua Marquês de São Vicente, 225, Gávea, Rio de Janeiro, Brazil
guarato@esp.puc-rio.br, emt@puc-rio.br, slbraga@puc-rio.br

Abstract.*Atkinson cycle is already recognized for having a better thermal efficiency then Otto cycle. That cycle is used so far mostly on reciprocating internal combustion engines (ICE) by changing the configuration of crank mechanisms. Miller cycle simulates Atkinson cycle by late closing of intake valves on ICE with classical (symmetrical) crank mechanisms. Rotary engines presents several advantages over conventional reciprocating ICE while having, in general, no valves or crankshafts. In this paper, the use of Atkinson / Miller cycle on a valveless rotary engine is proposed. The results highlight the rise on thermal efficiency while keeping inherent advantages of rotary engines.*

Keywords:*Atkinson / Miller cycle, rotary engine, thermal efficiency*

1. INTRODUCTION

Reciprocating internal combustion engines (ICE) are largely used in the industry for stationary electric generation. They are also widely used on automobiles and other vehicles and they can be flexible, depending on the power requirements of the applications. They have one combustion process per two revolutions of the output shaft in a 4-stroke engine and two in a 2-stroke engine.Therefore, their power density are very low. Furthermore, in order to get the fuel mixture into the combustion chambers and to get the exhaust gases out of the combustion chambers at proper time, there must be a complex valve system.

Wankel rotary engines avoid the problem of piston stoppage and reversal, and can provide on power stroke for each revolution of the shaft. However, these engines had limited use due to poor fuel economy, short operating life and high emissions (Zou, *et al.*, 2014).

The differential rotary engine was originally proposed by Bullington (Bullington, 1925). This engine is also known as cat-and-mouse engine due to its differential velocity drive mechanism (DVDM) which allows an oscillating movement of the pistons in a fixed arc (Zou, *et al.*, 2014).

Recently, differential rotary engines have been quickly developed, such as: the Wankel Engine (Norbye, 1971), the Drury engine (Drury, 1971), the Toroidal Taurozzi (1974), the Rotoblock (1993), Roundengine (Karim, 2000), the Massive Yet Tiny (MYT) (Morgado, 2004), the Rotatorque (2005) and the Yo-mobile engine (Sterling, 2011). The first one has the highest level of development among all the rotary engines.

This paper refers to a differential rotary engine called Kopelrot (Kopelowicz, 1988), (Kopelowicz, 2003), (Jiménez, 2008). It was successfully tested as a compressor (Barreto, *et al.*, 2003), (Barreto, *et al.*, 2004), (Kopelowicz, *et al.*, 2005).The aim of this research is to highlight the advantages of this differential rotary ICE and compare his thermal efficiency while running on Otto cycle or Atkinson / Miller cycle. The next section presents a review of Atkinson / Miller cycle.

2. ATKINSON / MILLER CYCLE

The Atkinson cycle, or the complete expansion cycle, is named after its inventor James Atkinson in 1882. This cycle consists on an isentropic compression followed by isochoric heat addition. The expansion occurs isentropically and the heat rejection is isobaric (Fig. 1).

In the Atkinson cycle, the expansion phase (3 to 4) is longer than the compression phase (1 to 2). This behavior was first created by an asymmetric crank mechanism due to an articulated rod. On conventional internal combustion engines (ICE) with symmetrical mechanisms, the Atkinson cycle is simulated by early inlet valve closure (EIVC) or late inlet valve closure (LIVC) in relation to the bottom dead center (BDC). Thus, part of the air-fuel mixture is forced back into the admission system and the effective compression ratio is reduced, making expansion ratio higher than compression ratio. This solution for simulating Atkinson cycle without asymmetric crank mechanism is called Miller cycle and it will be discussed in this paper. Previous works propose the use of Miller cycle in order to improve the thermal efficiency or reducing NOx emissions.

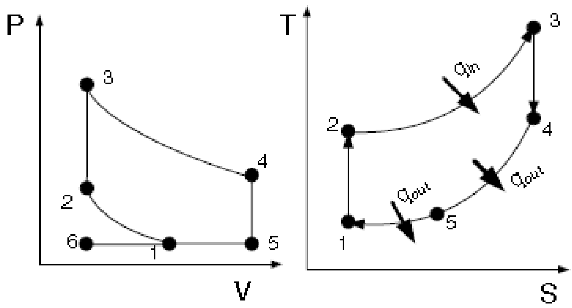

Figure 1. P – V and T – s diagrams of Atkinson cycle.

Fukuzawa, *et al.*, (2001) propose the use of Miller cycle on a lean-burn gas engines in order to achieve 40% on generating efficiency. They have studied the relation between expansion ratio and effective compression ratio on the engine thermal efficiency. Using the Miller cycle these authors demonstrated that low unburnt chamber type, i.e a chamber with low swirl ratio, have better results on thermal efficiency than high turbulence chamber type. They also developed a special high efficiency supercharged system with low flow rate and high-pressure ratio in order to achieve 42.2% thermal efficiency and 40% generating efficiency.

Mikalsen, *et al.*,(2009) present the advantages of using Miller cycle on natural gas engines for domestic applications such as combined heat and power (CHP) systems. They compare Miller and Otto cycles using a multidimensional simulation and discuss basic design implications. It was found that the combustion in the Miller cycle benefits from the lower engine speed and is slightly closer to a constant-volume process. An improvement from 5 to 10% on fuel consumption efficiency is possible in comparison to standard Otto cycle, but at the cost of a reduced power to weight ratio. Furthermore, Miller cycle advantages are more suitable for engines with low compression ratios, such as petrol or ethanol engines.

Gheorghiu (2011) propose the use of Atkinson cycle with asymmetrical crank mechanisms and very high-pressure turbocharging in order to increase internal combustion engine efficiency and power and reduce NOx emissions. With this configuration, called ultra-downsized, the author could achieve up to 40% better fuel conversion efficiency compared with the Otto cycle.

The majority of previous researches concern the use of the Atkinson or the Miller cycle on reciprocating internal combustion engines (ICE). In the special case of Miller cycle, late intake valve closure (LIVC) or early intake valve closure (EIVC) are used in order to have an equivalent behavior to an Atkinson cycle. Miller (1943) first proposed the use of EIVC to provide internal cooling before compression in order to reduce the compression work. However, EIVC has certain limitations in real applications such as reducing turbulence and mixing levels, with the subsequent reduction on speed ofcombustion processes. For these reasons, Anderson (1998) proposes the use of LIVC, in which the compression ratio decreases while retaining the nominal expansion ratio by discharging a fraction of the mass during the compression stroke that was inducted during the intake process. Moreover, the only Miller cycle ICE currently being manufactured for passenger cars employ LIVC. Consequently, we propose in this paper the use of Miller cycle by discharging some of the mass back to the intake systems with an equivalent system to LIVC.

2.1 Advantages of Atkinson / Miller cycle

As previously presented by several authors, the use of Atkinson / Miller cycle lead to the following advantages over the use of conventional Otto cycle:

1. a more complete expansion
2. better use of the combustion energy
3. lower noise in the exhaust port
4. lower temperature in the exhaust port
5. better overall combustion efficiency

These advantages are widely proved on conventional reciprocating ICE but there are few researches on applying this working cycle on rotary engines. The next section presents the configuration of the Kopelrot engine and how to the Miller cyclewas implemented on it.

3. KOPELROT ENGINE CONFIGURATION

Figure 2 shows two principal component parts of Kopelrot engine: the energy conversion system (ECS) and the differential velocity drive mechanism (DVDM).

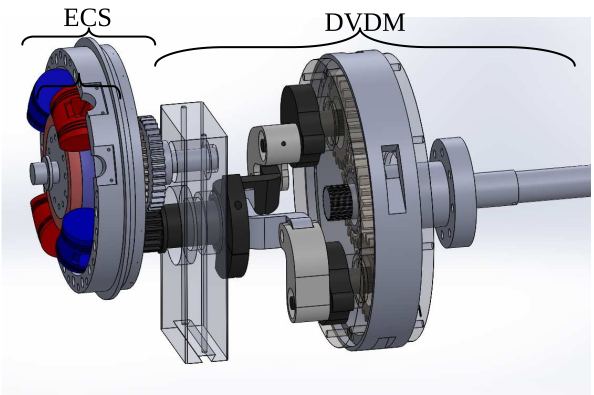

Figure 2. Configuration of the Kopelrot engine.

3.1 Configuration of ECS

The ECS has a two parts stationary housing and a pair of concentric rotors as shown in Fig.3. Each rotor having *N* diametrically opposed pistons, where *N* is an even number (in the case of Kopelrot engine *N*=2).

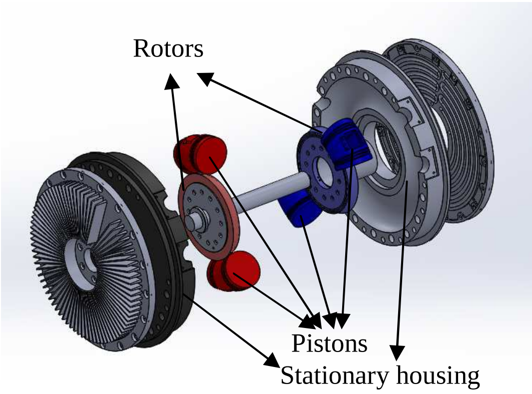

Figure 3. Configuration of the ECS.

The working chambers are defined by an upper wall from one piston, an lower wall from the subsequently piston, a part of the toroidal walls from the engine block and the radial surface from the rotors. The engine has four pistons, so it also has four working chambers at the same time.

3.2 Configuration of DVDM

DVDM is composed of a stationary sun gear, two planetary gears and two crank-rocker mechanisms. The two planetary gears mesh with the stationary gear for orbital motion (Fig.4).

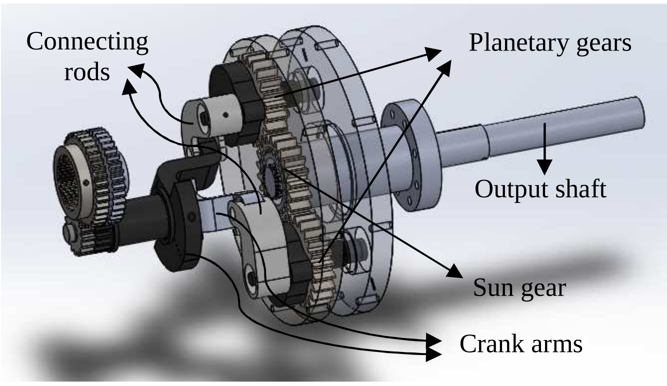

Figure 4. Configuration of DVDM system.

The two crank-rocker mechanisms are composed of a planet carrier, two cranks, two connecting rods and two rockers. The cranks are fixed on each respective planet gear, and positioned usually out of phase i.e. at 180º. The two rockers are respectively fixed to the two rotors by two separate hollow rotor shafts. The output shaft, which passes coaxially through the two hollow rotor shafts, is rigidly connected to the planet carrier. As the output shaft rotates, the two cranks, along with the planet gears are rotating in the same speed. With a non-zero phase, the two cranks also restrain the two rockers (and the two rotors) to move in the same variable oscillatory rotation at a different pace. Therefore, relative motion between rotors generates the volumetric change within each working chambers and creates the four-strokes cycle.

3.3 Working principle

DVDM interconnects the two rotors and enables them to speed up and slows down in a cyclical oscillatory rotation. The rotors are driven about their common axes and their relative angular velocity vary so that the volume of each working chamber is alternately expanded and then contracted. An inlet port, an exhaust port and an ignition device are provided at appropriate points on the toroidal housing, so that the expansion and contraction of the working chambers will provide admission, compression, combustion and exhaustion strokes (Zou, *et al.*, 2014).

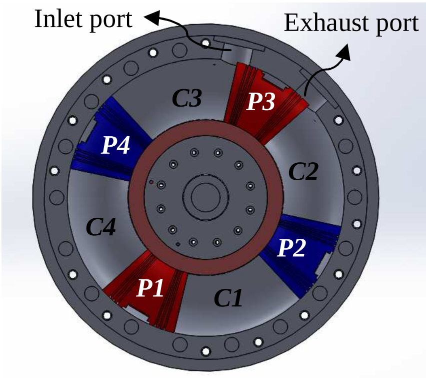

Figure 5. Pistons and chambers of ECS.

As this engine has four pistons *P1*, *P2*, *P3*, *P4*, there are also four working chambers *C1*, *C2*, *C3*, *C4* (Fig. 5). The working chamber *C1* lies in the position between the pistons *P1* and *P2*, the working chamber *C2* lies in the position between the pistons *P2* and *P3,* and so on. When *N* is 2, there are 1 inlet port and 1 exhaust port. In a general case, if the piston number is *N*, then *2N* working chambers are formed, and the number of inlet ports, exhaust ports and ignition devices are *N/2* (Deng, 2012).

One oscillation means one contraction and one expansion stroke for each working chamber. Consequently, for one revolution of the output shaft, each of 2*N* working chambers completes *N/*2 power strokes. As a result, the number of power strokes per revolution of the output shaft *PS* can be expressed as:

$$PS = N^2 \tag{1}$$

Figure 6 shows that for one chamber, formed between pistons *P1* and *P2*, there are two contractions and two expansions for every two complete rotations of the output shaft. Kopelrot engine has four chambers, so there are 8 contractions and 8 expansions for every two complete rotations of the output shaft.

The same stroke is performed simultaneously in *N/*2 different working chambers, i.e., with multiple use of the working space of the housing, the same stroke provides a balanced force to the rotors and increases the power density of the engine. The compression ratio r_c can be expressed as:

$$r_c = \frac{\delta_{max}}{\delta_{min}} \tag{2}$$

where δ_{max} and δ_{min} are the maximum and minimum of δ, which is the relative angle of the adjacent pistons. In the contraction phase the distance between pistons δ_{min} is 8.60° and in the expansion phase, the distance δ_{max} is 100.25°. As a consequence, the compression ratio ε is 11.66. δ is proportional to the volume of working chambers *V*, which is expressed as:

$$V = \frac{\pi d_1^2 \delta_{max} d_2}{8} \tag{3}$$

where d_1 and d_2 are respectively the diameters of the pistons and of the torus. In the case of Kopelrot engine $d_1 = 56.50\ mm$ and $d_2 = 183.46\ mm$. Thus, the volume of each combustion chamber is 0.128 liter.

Similar to the displacement definition of the conventional ICE (Stone, 1999), the displacement of the Kopelrot engine, Q_{ex}, is defined as the gas volume exhausted from the $2N$ working chambers per revolution of the output shaft, which can be expressed as:

$$Q_{ex} = \frac{N\pi d_1^2(\pi - N\beta)d_2}{4} \tag{4}$$

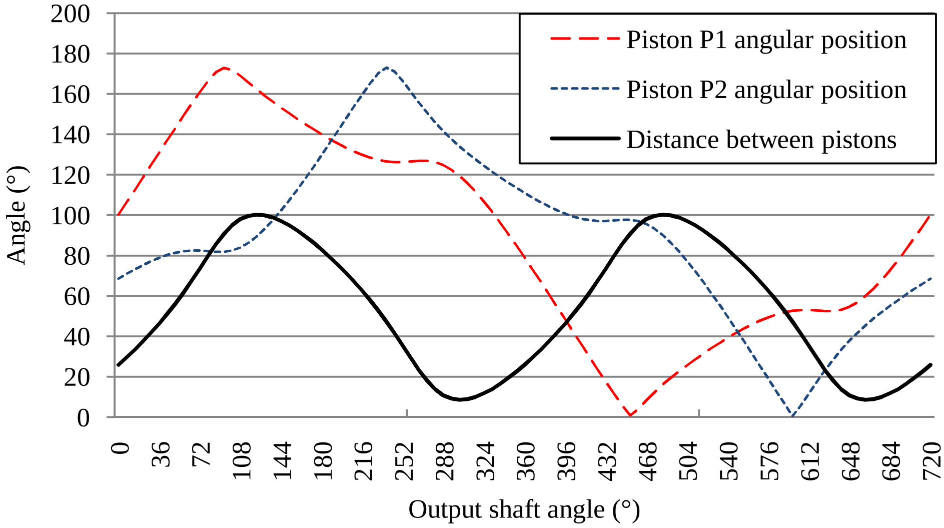

Figure 6. Angular position between pistons *versus* output shaft angle.

3.4 Advantages of Kopelrot rotary engine

As the others rotary internal combustion engines, the Kopelrot engine has the following advantages over traditional reciprocating ICE (Kopelowicz, 1988), (Kopelowicz, 2003), (Jiménez, 2008):

1. Higher power density
2. Multiple power strokes per revolution
3. Complex valve systems are not required
4. More uniform torque
5. More space for ignition spark plugs, inlet ports and exhaust ports

Although these advantages are inherent of every rotary engines, Kopelrot engine have also the advantage of a more efficient combustion due to the use of Miller cycle. Kopelrot engine does not have any complex valve timing system and, as stated before, the opening and closing of the intake and exhaust ports are made only by the movement of pistons, which are mounted in the toroidal chamber.

3.5 Aplication of Miller cycle

For applying Miller cycle with an equivalent effect of late intake valve closure (LIVC), the Kopelrot engine employs another port after the intake port. That port, called bypass, is connected to the inlet port. In consequence, it is possible to let some of the air-fuel mixture go out of the chambers and back to the intake port during the compression stroke, reducing the compression ratio. The expansion ratio is unchangeable.

The bypass port contains an adjustable pressure-sensitive valve that allows the compression ratio to vary in relation to the expansion ratio (Wittry, 1995). When the compression ratio is the same as the expansion ratio, the engine runs on Otto cycle. When the compression ratio is lower than the expansion ratio, the engine runs on Miller cycle, which generates a higher thermal efficiency.

In the angular position of the toroidal chamber where the power stroke takes place, there is the highest level of pressure. The bypass valve is always blocked by the lateral walls of one pistonduring the power stroke in order to prevent it from opening due to high pressure levels from combustion. For so, this port is located in a part of the toroidal chamber in which the minimum compression ratio is 30% in relation to the expansion ratio. Thus, it is possible to vary the compression ratio from 30% up to 100% of the expansion ratio.

Hence, the use of a pressure-sensitive valve on the bypass port is a good solution for applying Miller cycle on Kopelrot rotary engine for keeping its simplicity and avoiding the use of complex valve timing systems.

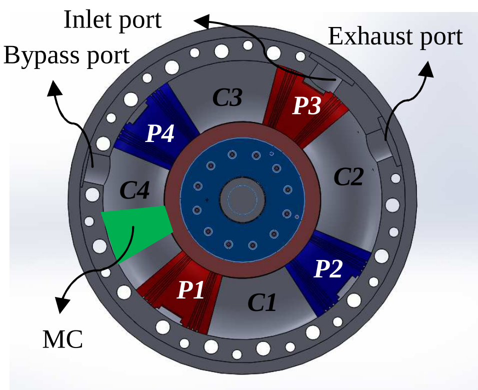

Figure 7. Application of Miller cycle with a bypass port before the position of maximum compression (MC).

The next section deals with a thermodynamic simulation in order to express the gain on thermal efficiency due to Miller cycle application.

4. THERMODYNAMIC SIMULATION

The thermodynamic simulation using ordinary differential equations (ODE) was solved by a Fortran VODE solver (Brown, *et al.*, 1989) that uses a multistep methods with fully variable step sizes and coefficients (Ticona, *et al.*, 2015). The total volume displacement and the expansion ratio are unchanged for every simulation. Miller cycle are simulated with different compression ratios, varying from near 30% up to 100% (Otto cycle) of the invariable expansion ratio.

The relation between compression ratio r_c and expansion ratio r_e is expressed as:

$$\varepsilon = r_c / r_e \tag{5}$$

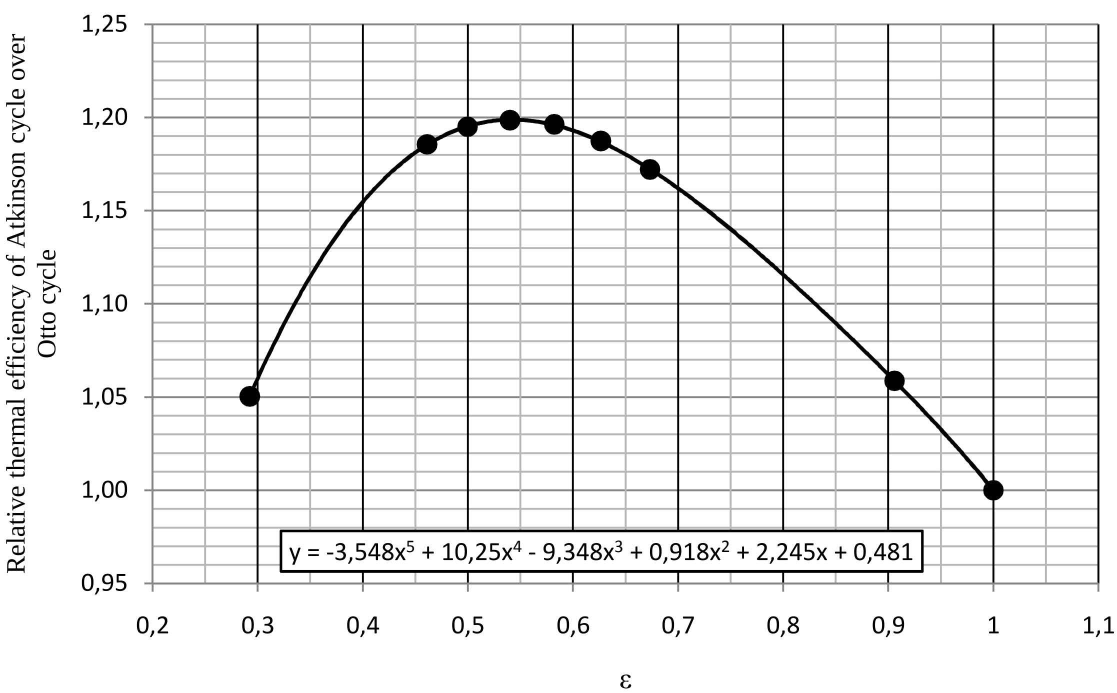

Figure 8. Relative efficiency in function of ε.

Figure 8 presents the relative thermal efficiency of Kopelrot engine running on Miller cycle in relation to Kopelrot engine running on Otto cycle. A 5 degree polynomial function was fitted to the simulation results to indicate the tendency. It is possible to note that efficiency rise until about $\varepsilon = 0.55$ and then decreases. Better efficiencies occur in the range from ε = 0.5 to ε = 0.6. The gain due to the Miller cycle is up to 19.85% when the compression ratio is 53.99% of the expansion ratio.

5. CONCLUSIONS

This research work highlights the advantages of rotary engines, particularly the Kopelrot engine, as well as the gain in thermal efficiency due to the use of Miller cycle. Rotary engines have several advantages over conventional reciprocating engines, such as higher power density, multiple power strokes per revolution, more uniform torque, more space for ignition spark plugs, inlet ports and exhaust ports, as well as no need for complex valve timing systems, which reduces production and maintenance costs. This paper demonstrates that it is possible to combine these advantages with higher engine efficiency using Miller cycle. The thermodynamic simulation, using ordinary differential equations, exhibits that whit Miller cycle almost 20% higher engine efficiency can be achieve than with Otto cycle on the Kopelrot engine.

Future work consists on the construction of the Kopelrot rotary engine and comparing the results from experimental tests with simulation results from this paper.

6. ACKNOWLEDGEMENTS

This work is supportedby CAPES grants (Science without Borders Project - CSF-PAJT - 88887.068777/2014-00).

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