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COMPARATIVE STUDY OF PERFORMANCE BETWEEN MILLER CYCLE ENGINE AND DIESEL CYCLE ENGINE

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Abstract. *Otto type internal combustion engines are the most common type in the automotive market today, however, when it comes to efficiency these engines leave a lot to be desired. But it is possible to increase its efficiency and match or even surpass the performance of the diesel cycle engines, whose technology has improved greatly in recent years, using techniques of variation of valve opening and change in compression ratio. This process would reduce fuel consumption and contributing to the economy of this sector that is one of the most growing.*
This paper reports the modification of a diesel cycle engine in an Otto cycle over-expanded engine (called the Miller cycle). These engines have efficiency higher than that of diesel engines. For this we will use two approaches: one is based on thermodynamic theory, the other is based on the experimental comparative analysis of the operation of diesel cycle engine transformed to Otto cycle engine and Diesel engines. To obtain a Miller cycle, a Diesel engine is converted to an Otto cycle by changing the opening of the intake valve and varying the compression ratio. At the end of this study we concluded in the two methods that the Miller cycle with varying compression ratio provides higher output.

Keywords: *Miller; Engines; Combustion; Performance; Diesel*

1. INTRODUCTION

Due the higher prices of fuels, the energy efficiency became the goal of many researches, trying to avoid the cost impact caused by the raise of the fuels. Among the solutions presented are the development of the cheap and friendly fuels and the search of energy production methods more efficient providing more fuel save, being the last one the target of this paper. With the same proposal, at the beginning of the twenty century, engineers developed an engine with a more complex mechanics making the expansion rate higher than the compression rate offering more efficiency than the Otto engines, but that complexity made that type of engine less usable (Dorić, 2011).

In (Miller R. H., 1947) a different type of Otto cycle was proposed by Miller. That cycle takes more time to expand the gases inside de chamber than to compress them, and that cycle was named Miller cycle. That engine is a regular Otto cycle engine with different valve timings and compression ratios. Martins and Ribeiro made in (Martins, J J.G, Ribeiro, 2006) and (Martins, J J.G, Ribeiro, 2007) a direct comparison between a Otto engine, a Diesel engine, a Miller engine and then they conclude that the Miller cycle engine was the most effective engine. At (Mikalsen, R. et al, 2009) Mikalsen proposed the usage of a Miller engine for heat and power generation in United Kingdom homes, which is a common practice in that country. In that experiment Mikalsen obtained 5 to 10% more fuel efficiency, but suggested future studies of applicability of the method. In (Chen, Lingen et. al., 2010) Chen analyzed the performance of a Miller cycle engine taking care about the friction and heat loss and specific heat of the working fluid using thermodynamic fundamentals. The Miller cycle isn´t only a reference in the efficiency field but also in the gas emissions fields as demonstrated in (Wang et. al., 2006). Wang used a Miller engine where the intake valve was closed long after the bottom dead center (BDC) reducing the combustion temperature. It was concluded that the reduction of the combustion temperature is directly related to the reduction of Nitrogen oxide (Nox) emissions.

This assay will analyze experimentally the efficiency of a regular Miller engine compared to a Diesel engine and a over expanded Miller engine (Martins, J J.G, Ribeiro, 2006) and (Martins, J J.G, Ribeiro, 2007).

2. ENGINE CYCLES

The internal combustion engines are the main way to provide power for vehicles that run on spark ignition (called Otto cycle) or compression ignition (Diesel cycle). Diesel engines consume less fuel, so they are more efficient and have a low emission of polluting gases while gasoline engines can provide more power, but has a disadvantage: it reduce the performance when working at partial loads, which is the most common condition of work of these types of engines.

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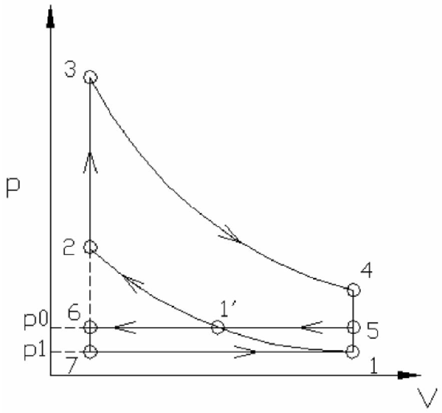

Figure 1 - P-V diagram of Otto Cycle

The reason for the decrease of its performance when the gasoline engines works at partial load, is that the admission is usually restricted, producing a partial vacuum in the intake manifold, so that the engine can operate with lower stoichiometric flow of fuel . This process can be seen in Fig.1 (Martins, 2007). The area defined by 1'-6-7-1-1' represents pumping work (negative) while the area 1'-2-3-4-5-1' shows the positive work on the developed piston. But it is possible to break this tendency of gasoline or ethanols have to increase consumption, by lowering the load.

In order to solve the issue of low efficiency of gasoline engines Ralph Miller showed up in his work "Supercharging and internally cooling for high output" 1947 the concept of a gasoline engine based on the Otto cycle, which offers a output higher even of that of Diesel engines, called Miller cycle. The basic principle of this engine is that the expansion phase must be longer than the compression phase.

2.1 Miller Cycle

The Miller cycle was proposed by Miller (1947), with the main objective of improving engine efficiency. It is an over-expanded cycle, i.e. a cycle with an expansion ratio higher than its compression ratio. Recently, the Miller cycle has also been proposed as a means of reducing harmful NOx emissions while maintaining a high engine efficiency, by reducing the engine compression ratio and thereby also peak in-cylinder gas temperatures and pressures. A number of reports have described the Miller cycle engine concept and investigated various aspects of its design and operation. R. Mikalsen et al (2009) compared the use os Miller and Otto cycle natural gas engines for small scale combined heat and power (CHP) applications. Al-Sarkhi et al. (2006) and Zhao and Chen (2007) presented theoretical investigations into Miller cycle engine performance, studying the influence of the main engine design variables and system irreversibilities. Endo et al. (2001) described the design of a commercially available large scale (280–1100 kW) gas engine using the Miller cycle principle, claiming a fuel efficiency advantage of more than 5% over comparable conventional technology.

In the design of a combustion engine there is a clear trade-off between power density and efficiency, and, since the power density requirement is relaxed in stationary applications such as thermoelectric stations, the engine can be optimised for high efficiency and long life. Engine optimization may include factors such as: the use of a high stroke to bore ratio to reduce in-cylinder heat transfer losses; reducing engine mean significant NOx reductions could be achieved, albeit with a penalty in engine fuel consumption. Fig. 2 shows the air-standard Otto and Miller cycles, and illustrates the additional work that can be extracted from the Miller cycle, look at shaded area (Mikalsen et al, 2009). Heywood (1988), showed that significant increases in engine efficiency can be achieved in over-expanded cycles, especially at low compression ratios.

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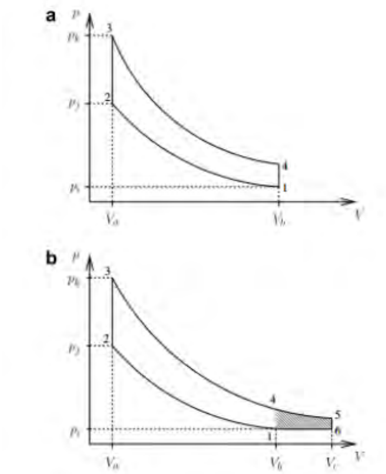

Figure 2 Comparison of Otto and Miller air-standard cycles. (a) Air-standard Otto
Adapted from (Mikalsen et al, 2009)

The Miller cycle is an offshoot of the mainstay Otto cycle and its four strokes. But in a Miller-cycle engine, the intake valve doesn't close at the beginning of the compression stroke. It doesn't close until the piston is nearly a fifth of the way through its travel up the cylinder wall. The Miller cycle is more efficient than its peers due result of several factors coming into play at once.

The Miller cycle is defined as earlier or later closing of the intake valves which increases the effective expansion ratio in relation to the compression. This means that the indicated engine efficiency should be increased. The earlier or later closing of the inlet valves means that the effective swept volume is reduced and must therefore be compensated by increasing the intake pressure with a turbo to make sure the same air mass is found in the cylinder when compression takes place. This is in effect an outsourcing of work from the cylinder to the turbo. This outsourcing will place greater demands on the turbo system, a result which means that a new turbo must be matched to the engine.

With the intake valve open during the first fifth of its compression stroke, a Miller engine's piston will push some of air/fuel mix it sucked back out through the intake port. Once the intake valve closes, there's less of an air/fuel mixture to compress in the cylinder, so less energy is spent compressing it, reducing the pumping loss. But even with more efficient compression, the smaller charge of air and fuel in the cylinder means less energy density, which means that the engine makes less power.

To counteract this, Millers are equipped with a supercharger. Pressurized air from the supercharger is run through an intercooler, which lowers the air charge's temperature — and also makes it denser, so more air fits in the same volume. During the intake stroke, that charge of cold, compressed air rushes into the cylinder when the intake valve opens, filling it with a greater volume than could be sucked in by the simple downward movement of the piston. As the compression stroke starts, the supercharger's output keeps the cylinder pressurized until the intake valve closes, preventing the air/fuel mixture from being pushed back into the intake manifold. The piston pushing back against the air charge from the supercharger requires less energy than the piston compressing air in a closed cylinder, so pumping losses still remain lower than a traditional engine (Dembinski & Lewis, 2009).

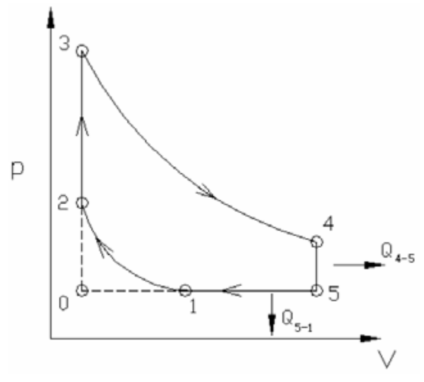

Figure 3 - P-V diagram of Miller cycle
Adapted from (Martins, 2007)

Figure 2 shows a volume x pressure diagram of the Miller cycle, where the compression phase, which is represented by points 1-2, is smaller than the expansion phase, represented by points 3-4. In the figure can also be seen

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that the inlet valve is closed before the piston reaching the BTC, so that this will imply a lower expansion (thereby lowering the pressure) followed by compression to the point 2. At one point, the pressure is atmospheric, it is considered that the compression starts at this point. Thus, the intake volume in both cases is $\Delta V = V_1 - V_2$ which is the compression volume also.

Regarding the compression ratio, this cycle is different from the Otto or Diesel, since the compression is called "retained" ϵ_{ret} and is given by:

$$\epsilon_{ret} = \frac{V_1}{V_2} \quad (1)$$

Considering the displacement as $V_c = V_5 - V_2$ one can also define the geometric compression ratio ϵ_g:

$$\epsilon_g = \frac{V_5}{V_2} \quad (2)$$

An important feature is the Miller cyclic expansion ratio (σ), which again is a relation between volumes, or heat between the BDC and the value at the time it begins the compression:

$$\sigma = \frac{V_5}{V_1} \text{ e } \sigma = \frac{\epsilon_g}{\epsilon_{ret}} \rightarrow \epsilon_{ret} = \frac{\epsilon_g}{\sigma} \quad (3)$$

In fact this value gives us the relation between expansion and compression, which is unitary in Otto and Diesel cycles. The larger σ means higher the utilization of the expansion work.

2.2 Miller Cycle Performance

In this part are presented the equations developed in (Martins, 2004) for calculating the efficiency of Miller cycle taking into account the geometric parameters of the engine. For the Miller cycle with geometric compression ratio ϵ_g fixed considering the capacity (swept volume V_c) fixed and stationary combustion volume (V_2) can carry out the Miller cycle by changing the opening of the intake valve while maintaining geometric compression ratio and volume of the combustion chamber fixed. In this case the efficiency of the Miller cycle is:

$$\eta = 1 - \frac{1}{\epsilon_g^{\gamma-1}} - \frac{1+\sigma^{\gamma}(\gamma-1)-\gamma\sigma^{\gamma-1}}{(\gamma-1)\sigma^{\gamma-1}B} \quad (4)$$

$$B = \frac{LHV}{RT_1(1+\lambda)} \quad (5)$$

Where B is a constant assuming that the mixture is always stoichiometric and

$\lambda = \frac{Air}{Fuel}$ air fuel ratio;

LHV is the Low Heating Value;
R is the gas constant;
T_1 is the point 1 temperature;
γ is the ratio of heat capacities

The efficiency variation with σ and ϵ_g can be seen in the figure below. The various lines (corresponding to the various geometric compression ratios) decreases with the increase of σ as the compression ratio decreases with σ retained. Again passing discontinuous line continues to mark the value of atmospheric pressure at the end of the expansion, Fig (3).

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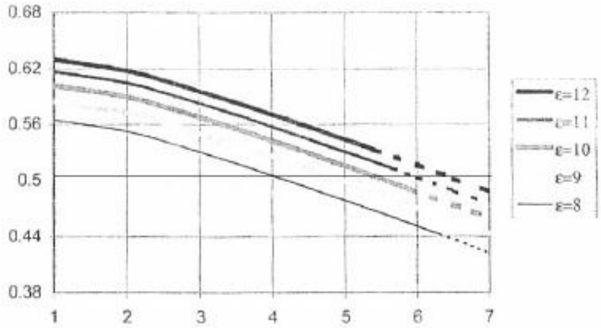

Figure 4 Efficiency variation as a function of σ

Adapted from (Martins, 2006)

The compression ratio in modern conventional spark ignited engines is typically limited to around 10 due to fuel auto ignition limitations. Ethanol has higher knock resistance (octane number), and may utilize higher compression ratios. For applications such as the one investigated here, a lower compression ratio is, however, desirable in order to reduce cylinder sealing requirements, frictional losses and engine noise.

The maximum cycle pressure will depend on the amount of heat added to the charge in the constant-volume process 2–3. For spark ignition engines, the ratio of end-of-compression pressure to peak pressure is typically around 3 at full load.

Figure 5 shows the air-standard efficiency advantage that can be achieved in a Miller cycle engine compared to the equivalent Otto cycle. It is seen that the Miller cycle advantages are larger for engines with low compression ratios (Mikalsen et al (2009) apud Heywood (1988). Furthermore, a significant influence of the heat input is seen, indicating that the Miller cycle is suitable for engines operating at high loads, and possibly more beneficial for fuels with high heat content.

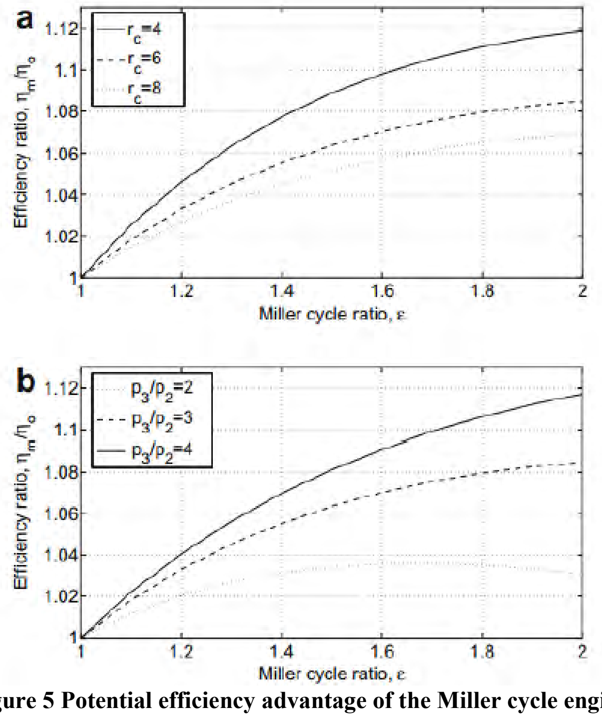

Figure 5 Potential efficiency advantage of the Miller cycle engine.
Adapted from (Mikalsen et al, 2009)

Due the installed park of power generation in north of Brazil burn mainly fuel diesel, the next section discusses which diesel cycle engine works.

2.3 Diesel Cycle

The essential features of the compression-ignition or diesel engine combustion process can be summarized as follows. Fuel is injected by the fuel-injection system into the engine cylinder toward the end of the compression stroke, just before the desired start of combustion. The liquid fuel, usually injected at high velocity as one or more jets through small orifices or nozzles in the injector tip, atomizes into small drops and penetrates into the combustion chamber. The fuel vaporizes and mixes with the high-temperature high-pressure cylinder air. Since the air temperature and pressure are above the fuel's ignition point, spontaneous ignition of portions of the already-mixed fuel and air occurs after a delay period of a few crank angle degrees. The cylinder pressure increases as combustion of the fuel-air mixture occurs. The consequent compression of the unburned portion of the charge shortens the delay before ignition for the fuel and air which has mixed to within combustible limits, which then burns rapidly. It also reduces the evaporation time of the remaining liquid fuel. Injection continues until the desired amount of fuel has entered the cylinder. Atomization, vaporization, fuel-air mixing, and combustion continue until essentially all the fuel has passed through each process. In addition, mixing of the air remaining in the cylinder with burning and already burned gases continues throughout the combustion and expansion processes (Heywood, 1988).

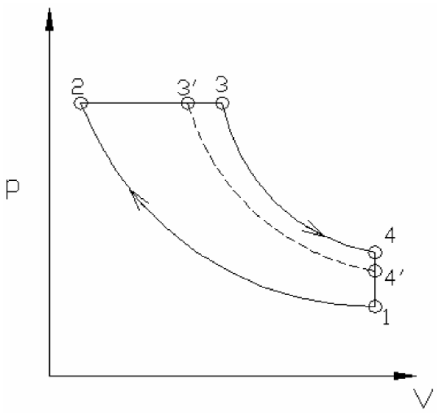

Figure 6 - P-V diagram of Diesel cycle

2.4 Dual Cycle

The Dual cycle shows with more reality, the theoretical operation of the rapid diesel engines, since it presents one initial supply of heat, which in modern engines can be considered the result of pre-injections. Then the rest supply of heat is given at constant pressure (Martins, 2007). The P-V diagram of the real internal combustion engines are not well described by the Otto and Diesel cycles. A standard air cycle could be elaborated for a better approach of the pressure variations is the Dual cycle. The Dual cycle is shown in figure 7. Like on the Otto and Diesel cycles, the process 1-2 is a isentropic compression. However, the heat supply occurs in two stages: the process 2-3 is an addition of heat at constant volume; the process 3-4 is an addition of heat at constant pressure. The process 3-4 also makes part of the first stage of power. The isentropic expansion from state 4 to state 5 is the remaining stage of power. Like on the Otto and Diesel cycles, the cycle is completed by a process of heat rejection at constant volume, the process 5-1

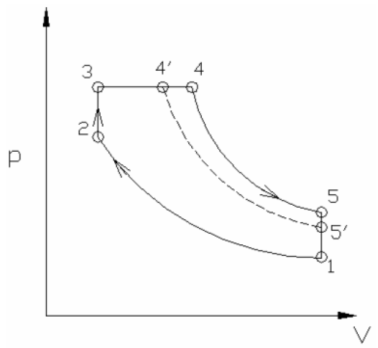

Figure 7 P-V diagram of Dual cycle
Adapted from (Martins, 2004)

3. USEFUL POWER

In order to obtain the information necessary for the analysis of the engine to be experienced follows some necessary formulations.

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3.1. Useful Power Output

The useful power is taken as being the product of torque and angular velocity of the shaft, as seen in Eq. (6) known as brake horsepower (Filho, 2008), which is received by the dynamometer.

$$P = (T * 2\pi * \omega)/60000 \tag{6}$$

Where the power P is given in (kW), torque T in (Nm) and angular velocity ω in (rpm).

3.2. Specific Fuel consumption

This is important information for the analysis of the engine in question, as it gives us a vision of how much fuel the engine is leveraging to generate useful work. Fuel consumption can be given by Eq. (7).

$$sfc = m/P \tag{7}$$

Where *sfc* is the specific fuel consumption given in (kJ / kg), *m* is the mass of fuel consumed in (kg), and *P* is the useful power.

3.3. Thermal Efficiency

Thermal efficiency η is a dimensionless value used to measure the efficiency of a power of an engine from the power of the shaft and the heat generated in the combustion. Thermal efficiency can be given by Eq. (8).

$$\eta = P/(m * LHV) \tag{8}$$

Substituting the Eq. (7) on the Eq. (8) we have:

$$\eta = 1/(sfc * LHV) \tag{9}$$

4. EXPERIMENTAL TEST

The engine to be experienced is a turbocharged engine 12A SGI modified to close the intake valve after Botton Death Center, connected to an electric generator in order to produce electricity, as shown in Fig. (5), whose specifications are shown in Tab. (1). To perform the experiment, the engine in question is operated with ethanol fuel from 9 pm until 16:14 pm (local time) and data were collected by computer.

Figure 8 - Engine SGI 12A

Table 1 - Engine specifications

Engine	Scania SG1 12A
Bore	127 (mm)
Stroke	154 (mm)

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Cylinders	6
Arrangement of cylinders	Line
Ignition order	1-5-3-6-2-4
Displacement	0,0117 (m^3)
Compression Ratio	11:1
Maximum Power	250 (kW)

After the experiment, calculations were performed considering the Lower Heating Value of the Ethanol equal to 28225 (kJ / kg) (Etanol como combustível Veicular, 2008), and engine working at 1789 (rpm). The results are shown in Tab. (2) to four operating ranges.

Table 2 Experimental results to Miller engine using ethanol as fuel

Power Charge (%)	Energy (kWh)	Sfc* (kg/kWh)	Efficiency (%)
78	195	0.39	33
52	131	0,41	32
40	100	0,44	29
20	49	0,91	14

In order to compae the ethanol Miller engine, a diesel engine was tested with the following specifications: diesel engine MWM D229-4 according to the manufacturer nominal potential of 54 kW total of 3,922L cylinder compression ratio and 17:1 at 2500 rpm, connected to a three-phase synchronous generator KOHALBACH brand, model LA 180 40 kVA , 220/127 V, 1800 rpm, with a frequency of 60 Hz. In the table 3 is showed the tests results:

Table 3 Experimental results to compression ignition engine using diesel as fuel

Power Charge (%)	Energy (kWh)	Sfc* (kg/kWh)	Efficiency (%)
75	30	0,285	28,48
50	20	0,290	27,99
40	16	0,322	25,21
20	8	0,415	19,56

Is possible to see over the tables that Miller engines have most thermodynamic efficiency those diesel engines, and despite the higher specific consumption shows it as an excellent option in view of energy use.

5. CONCLUSION

In this paper were presented an experiment to compare the thermal efficiency of an engine under Miller cycle with the thermal efficiency of an engine under Diesel cycle. The experiment consisted of collecting data from an engine Scania SGI 12A which operated with fuel ethanol. The motor is connected to an electric generator to convert the mechanical energy from the motor shaft into electrical energy. Was observed that the efficiency of ethanol Miller engine is very close to the diesel engine. Considering that ethanol can be distilled from fermented cassava, sugar cane or any plant capable of producing starch or sugar, the production of power electricity using the ethanol Miller engines is very feasible. The use of these devices may represent the supply of large isolated areas and allow the human development index for that region is increased.

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ACRONYMS
BDC – Botton Death Center
NOx – Nitrogen Oxides
P – Power
T – Torque
ω – Angular velocity
sfc – Specific fuel consumption
LHV – Lower heating value
η – Efficiency
γ - Especific Heat Ratio

4. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.