OTIMIZATION STUDY OF DUAL-FUEL, GASOLINE-ETHANOL ENGINE PERFORMANCE, USING ONE TURBOCHARGER

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Abstract. This work is an optimization study of dual fuel engines performance burning gasoline or ethanol (under a flex-fuel mode). It has been done for an Otto Cycle, four strokes, system electronically commanded by a turbocharger as function of the gasoline-ethanol ratio. The turbocharger action increases the compression ratio in order to adequate values to of any gasoline/alcohol mixture combination, improving the volumetric relations and leading to better thermal efficiencies. As the compression ratio is fixed and limited by the gasoline octane number, the turbocharger action can increase density and pressure of the admitted mixture, adjusting the compression ratio for some ethanol percentage in the gasoline. In the engines possessing oxygen sensors, a signal will trigger an actuator at the valve that determines the correct load of the turbocharger in order to optimize the compression ratio. Hence this study demonstrates the benefits which a controlled turbocharger system brings to "flex" engines.

Keywords: dual fuel engines, Otto cycle, turbine, compression ratio, relief valve.

1. INTRODUCTION

Spark ignition engines compression ratio is established according to the fuel thermal stability. Compression ratio large values are highly desirable, for the engine thermodynamic efficiency depend on them. In the ethanol/gasoline dual fuel engines industry, the tendency in general leans towards the adequate compression ratio of the fuel possessing the smallest thermal stability, i.e., the gasoline. The maximum value of the compression ratio is limited by the fuel self ignition point (detonation), i.e., it is established as the lowest limiting value for which there is risk that burning might occur before the point of ignition (Heywood 1988).. The fuel thermal stability is a highly relevant parameter and it can be evaluated as a function of its octane level. When the engine is designed to operate with ethanol, the compression ratio to which the mixture is submitted reaches values above those adopted by flex engines. Actually the engines running on gasoline only possess a compression ratio around 10:1, while those running on ethanol can operate on ratios around 13:1. (Lichty, 1951).

The engine overall efficiency depends on the compression ratio, the fuel heating power(LHV), the combustion chamber geometry, the mixture quality, among other factors which, after burning, lead to high gases pressure conditions and the useful work performed by the engine. In the dual fuel engine operating with ethanol, relevant parameters such as the compression ratio can be neglected, compromising the engine overall efficiency. For the engine with an electronic injection system, a fuel to oxidizer ratio close to its stoichiometric is kept for any fuel entering the combustion chamber due to the many sensors readings and the action of the actuators. It is worth noticing that under common use the tendency is whenever possible, if available, to use the ethanol.

As the geometry and its volumes are fixed, the suggested solution to vary the compression ratio is to adequate the admission mixture density up to values compatible with fuel or the fuels mixture under use. Each time there occur a significant change in the ethanol/gasoline mixture, sensors will collect data of these conditions and the electronic center will act on the turbine blades on the way of adjust the compression ratio for this situation, leading this way to an increase of the engine efficiency.

This is based on an 1.6 flex engine, whose characteristics are presented in Figs. 1 and 2. The engine runs on a compression ratio around 10,16/1.

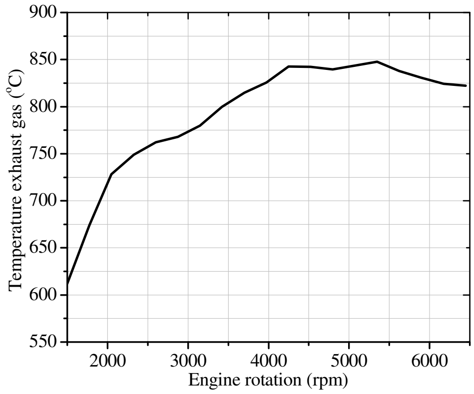

Figure 1- Flex Engine experimental results I: Manifold gas temperature vs rpm.

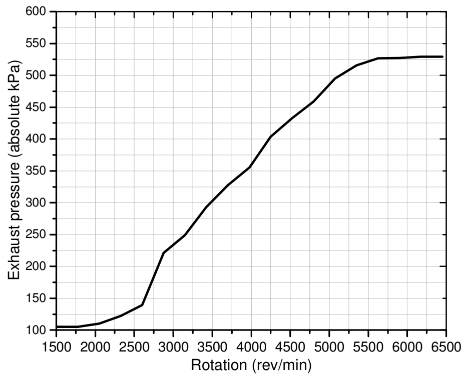

Figure 2- Flex Engine experimental results II: Manifold gas pressure vs rpm.

2. THERMODYNAMIC STUDY

Results of tests carried out on an 1.6 flex-fuel engine.
Compressive work, adiabatic process, $\gamma = c_p/c_v$ constant, air standard cycle. Fig. (3):

$$W_c = \frac{p_1 v_1 - p_2 v_2}{\gamma - 1} \quad (1)$$

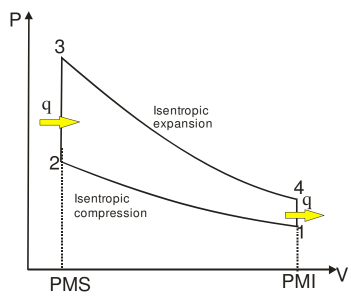

Fig. 3. Standard air Otto cycle Scheme.

It is important to notice that the actual compressive work is a function of the volumetric efficiency, and it depends on the engine rotational speed. For the ideal case one has (Van Wylen, 2004):

$$W_c = \frac{p_1 v_1 \left(1 - r_c^{\gamma - 1}\right)}{\gamma - 1} \quad (2)$$

W_c → Compressive work
r_c → Compression ratio.

v → Specific volume

The amount of heat released to the system per cycle is:

$$q = m_c \cdot \mathrm{LHV} \tag{3}$$

q → Heat received per cycle
m_c → Fuel mass burned per cycle

$$T_3 = T_2 + \frac{Q}{m_a c_v} \tag{3}$$

m_a → Air mass admitted per cycle

The RAC, the Air to Fuel mass ratio can be written as function of the fuel mass:

$$T_3 = T_2 + \frac{Q}{(RAC+1)\cdot m_c c_v} \tag{4}$$

RAC → air/fuel ratio
The ideal compression work is:

$$W_E = \frac{p_3 v_3 - p_4 v_4}{\gamma - 1} \tag{5}$$

W_E → Expansion work

$$W_E = \frac{p_3 v_3 - p_3 \cdot r_E^{-\gamma} r_E v_3}{\gamma - 1} \tag{6}$$

$$W_E = \frac{RT_3\left(1 - r_E^{1-\gamma}\right)}{\gamma - 1} \tag{7}$$

$$W_\ell = \frac{RT_3\left(1 - r_E^{1-\gamma}\right)}{\gamma - 1} + \frac{RT_1\left(1 - r_c^{\gamma-1}\right)}{\gamma - 1} \tag{8}$$

W_ℓ → Cycle net work

$$W_\ell = c_v T_3\left(1 - r_E^{1-\gamma}\right) - c_v T_1\left(1 - r_c^{\gamma-1}\right) \tag{9}$$

$$W_\ell = c_v\left(T_1 r_c^{\gamma-1} + \frac{Q}{(RAC+1)\cdot m_c c_v}\right)\left(1 - r_E^{1-\gamma}\right) - c_v T_1\left(1 - r_c^{\gamma-1}\right) \tag{10}$$

An analysis of the discharge gases thermodynamic conditions allow to observe that these gases yield work to run a turbine only at speeds above 2050 rev/min (Fig.4)

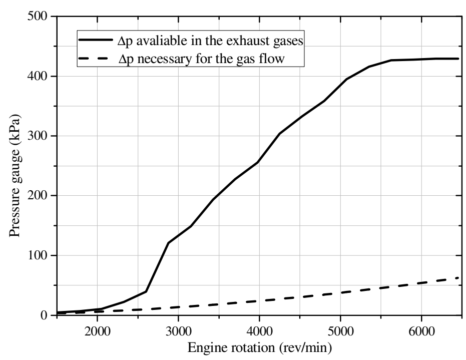

Fig.4 – Flex Engine experimental results III: Exhaust gas pressure available vs rpm.

The calculation of this work allows estimating the possible compression rates to be reached and their respective thermodynamic efficiencies Fig. 5.

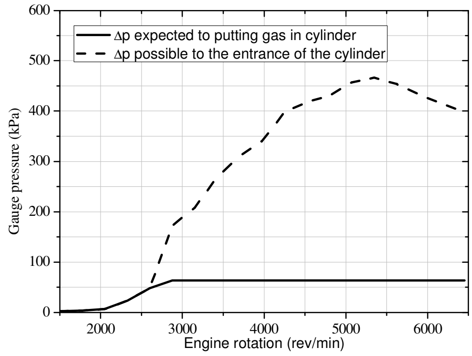

Fig. 5 – Admission pressure distribution vs engine rpm.

The air mass flow rate through the discharge pipes require a minimum pressure difference for proper flow to occur Bernoulli equation yields:

$$\frac{\Delta p}{\rho} = f\frac{L}{D}\frac{v^2}{2} \tag{11}$$

$$\Delta p = \frac{\rho L f}{2D}\left(\frac{CT}{60xA_N}\right)^2 n^2 \tag{12}$$

A_N → The area normal to the flow through the exhaust pipe.
f → Represents the sum of all flow friction factors.

According to Martins et al (2006), the pressure difference to take into account all pressure losses which take place in the piping system, catalyst, muffler, etc., can be estimated as

$$\Delta p = 3000\left(\frac{CT}{60xA_N}\right)^2 n^2 \tag{13}$$

When the engine has a turbocharger (ideal cycle, Fig. 6), the energy received by the turbine along an isentropic process is given by (Stone, 1999):

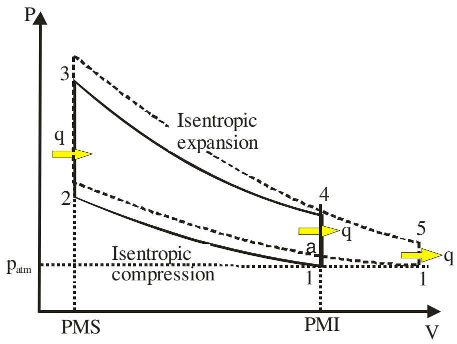

Fig. 6 – Ideal cycle with turbocharger.

$$W_T = c_p(T_4 - T_5) \tag{14}$$

T_4 → Turbine entrance temperature
T_4 → Turbine exit temperature
W_T → Energy taken from the fluid by the turbine

$$W_T = c_p T_4 \left(1 - \frac{T_5}{T_4}\right) \tag{15}$$

$$W_T = \frac{\gamma R T_4}{(\gamma - 1)}\left[1 - \left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{16}$$

The work available in the turbine axis is given by:

$$W_{TR} = \eta_T \frac{\gamma R T_4}{(\gamma - 1)}\left[1 - \left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{17}$$

The work available on the turbine axis is given by (Cohen et al, 1996):

$$W_{TR} = \eta_T \frac{\gamma R T_4}{(\gamma - 1)}\left[1 - \left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{18}$$

where
η_T → Is the turbine efficiency
W_{TR} → Work available at the turbines axis

The actual work to deploy the compressor, supplied by the turbine is:

$W_{CR} = W_{TR}$
W_{CR} → actual work received by the compressor

The compressor work effectively transferred to the fluid for its compression is:

$$W_C = c_p (T_a - T_1) \tag{19}$$

W_C → Energy yielded by the compressor to the fluid

$$W_C = c_p T_1 \left(\frac{T_a}{T_1} - 1\right) \tag{20}$$

$$W_C = \frac{\gamma R T_1}{(\gamma - 1)}\left[\left(\frac{p_a}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \tag{21}$$

Actual compressor work , W_{CR}.

$$W_{CR} = \frac{1}{\eta_C}\frac{\gamma R T_1}{(\gamma - 1)}\left[\left(\frac{p_a}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \tag{22}$$

The work transferred to the fluid as a function of the exhaust temperature is:

$$W_C = \eta_T \eta_C \frac{\gamma R T_4}{(\gamma - 1)}\left[1 - \left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{23}$$

The suction pressure is:

$$\frac{\gamma R T_1}{(\gamma - 1)}\left[\left(\frac{p_a}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = \eta_T \eta_C \frac{\gamma R T_4}{(\gamma - 1)}\left[1 - \left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{24}$$

$$p_a = p_1 \left\{\eta_T \eta_C \frac{T_4}{T_1}\left[1 - \left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}}\right] + 1\right\}^{\frac{\gamma}{\gamma-1}} \tag{25}$$

The pressure obtained after the compression at the compressor constant volume is:

$$p_a = p_1 \left(\frac{W_c(\gamma-1)}{\gamma R T_1} + 1\right)^{\frac{\gamma}{\gamma-1}} \tag{26}$$

The actual compressor work WCR as a function of T_4 is.

$$W_{cR} = \eta_T \eta_C \frac{\gamma R T_4}{(\gamma-1)}\left[1-\left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{27}$$

Thermodynamic efficiency of an engine, air standard Otto cycle. For the engine under consideration, with a compression ratio of 10,16/1, the ideal thermodynamic efficiency is:

$$\eta_T = 1 - r_c^{1-\gamma} \tag{28}$$

With a variable geometry turbocharger Bauduin (2002), the thermodynamic efficiency will increase with compression ratio applied to the system up to acceptable values according Fig. 7.

For a maximum acceptable compression ratio of 14.4/1 one has:

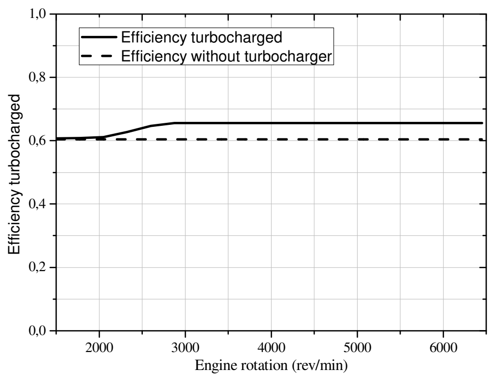

Fig. 7 – Change of the thermodynamic efficiency of an Air Standard Otto Cycle, overloaded.

Starting at 2850 rpm, it is observed the efficiency gain is of the order of 8.5% with the addition of a turbocharger

3.CONCLUSION

The use of a turbocharger will allow new curves for the specific consumption, as well as for the torque and the power. The engine must reach higher speed of rotation in less time due to the better cylinder filling. The use of a variable geometry turbine controlled by actuators according to measurements made by mixture identification sensors will allow the correct engine operation with proper compression ratio for all ethanol/gasoline mixture relations. As one does not need maximum turbine pressure to increase the admission density and pressure, for the pressure will be limited by the compression relation of an engine with a compression ratio of 14:1, one can obtain these densities and pressures under lower rotation rates, achieving proper torque and power already in lower rotation rates.

4. REFERENCES

Bauduin, Philippe. The Turbocompressor. *Aerostories*, [S.I.], 2002. Acesso em: nov. 2006, disponível em: http://aerostories.free.fr/technique/turbo/page2.html.

Cohen, H; Rogers, C.F.C; Saravanamutoo, H.I.H. *Gas Turbine Theory,* 4th edition. Longman Group1996.

Heywood, J. B. *Internal Combustion Engine Fundamentals*. Copyright McGraw-Hill Inc, New York, 1988.

Lichty, L.C. *Internal Combustion Engines.* Tókio: Mc Graw-Hill, 1951.

Martins, K.C.R. *Analise Experimental, Teórica e Computacional dos Gases de Exaustão no Conversor Catalítico Platina-Paládio.* Tese de Doutoramento- EESC-2006.

Stone, R. *Introduction to Internal Combustion Engines*. 3. ed. New York: Palgrave, 1999.

Van Wylen, G.V. ;Sonntag, R. E.; Borgnakke, C.,*Fundamentos da Termodinâmica*. 6. ed. São Paulo: Editora Edgard Blücher, 2004.

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