18th Brazilian Congress of Thermal Sciences and Engineering
November 16-20, 2020 (Online)

ENC-2020-0030
ANALYSIS OF VEHICLE POWERED BY SUPERCHARGED VERSUS ASPIRATED OTTO CYCLE INTERNAL COMBUSTION ENGINE

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Abstract. *This article has experimentally analyzed the influence of the turbocharger on the performance parameters of a spark ignition engine. Initially, a 2-liters naturally aspirated engine operating with podium gasoline was tested. Torque and power measurements were taken directly on the engine flywheel with a bench-top electric dynamometer obtaining the greatest torque of 148 N·m and power of 79.9 kW. After the bench test, measurements were taken directly on the vehicle's wheels using an inertial roller dynamometer being possible to verify a greater transmission loss of 16.9%. In order to supercharge the engine, a turbocharger set and its components were dimensioned to achieve a greater fuel flow (pump and injectors). As fuels, podium gasoline, ethanol and blends of gasoline with ethanol were used. The emission tests for the natural aspirated and supercharged vehicle were compared with the CONAMA standard 418/2009, with CO, HC and* CO_2 *levels being measured. The supercharged vehicle operating with ethanol showed the best results: torque of 218.6 N·m and power of 114.5 kW, both at 5000 RPM.*

Keywords: *Internal combustion engine, turbocharger, electronic fuel injection, electronic ignition, emissions.*

1. INTRODUCTION

Due to demand for internal combustion engines with lower levels of emissions there was a search for small models, however, with greater power output. One way to achieve such product is by using a turbocharger which improves the engine volumetric efficiency. According to Gheorghiu (2013), the reduction in engine displacement and the use of a supercharging system to maintain torque and power is called downsizing.

According to Brunetti (2012), volumetric efficiency is the ratio between the mass of air that effectively enters the cylinder and the mass of air that would occupy the volume displaced by the piston. Martins (2006) comments that because volumetric efficiency is directly associated to air density, high-performance and supercharged engines are able to have an efficiency greater than 100%.

The turbocharger consists of a compressor and a turbine installed on the same axis. The turbine uses the energy from the exhaust gas to operate the compressor, which supplies air in greater volume to the cylinders. Turbochargers are traditionally used on diesel cycle engines. However, with the constant search for reducing the number of cylinders, gaining power and reducing fuel consumption, the use of supercharging in Otto cycle engines has become more frequent (BOSCH, 2005).

In Brazil, the emission standards from the internal combustion engine (ICE) are stablished by CONAMA (2009), which specifies vehicle inspection procedures, emission limits and vehicle pollution control plans. Vehicle inspection must be carried out with the vehicle in neutral operating at 1000 RPM and 2500 RPM, with tolerance of ±200 RPM.

An important factor for the correct functioning of internal combustion engines is the lambda factor (λ), which relates the actual mixture to the stoichiometric mixture of air-fuel, as shown in Eq (1).

$$\lambda = \frac{\left(\mathrm{A}/_{\mathrm{F}}\right)_{\mathrm{actual}}}{\left(\mathrm{A}/_{\mathrm{F}}\right)_{\mathrm{stoich}}} \tag{1}$$

where $\left(\mathrm{A}/_{\mathrm{F}}\right)_{\mathrm{actual}}$ is the actual air-fuel ratio and $\left(\mathrm{A}/_{\mathrm{F}}\right)_{\mathrm{stoich}}$ is the stoichiometric air-fuel ratio. When $\lambda > 1$ the mixture is considered poor and there is excess of air. When $\lambda > 1$, less fuel consumption occurs and is applied in decelerations. When $\lambda < 1$ the mixture is considered rich, being used for accelerations. The λ for maximum power occurs between 0.85 e 0.9. Stoichiometric air-fuel mixture ($\lambda = 1$) is normally used at cruising speed (CRUZ, 2015).

2. EXPERIMENTAL PROCEDURES

A VW Golf 1994 passenger vehicle was used, equipped with a spark ignition (SI) engine, electronic injection and a single-point injection in each cylinder. Main engine characteristics are presented in Tab. 1.

Table 1. VW Golf 2.0 engine characteristics.

Parameter	Specification
Number of cylinders	4
Power at 5400 RPM	85 kW
Torque at 3200 RPM	166 N·m
Operation mode	Four-stroke
Engine displacement (cm^3)	1984 cm^3
Management system	BOSCH 2.9
Cylinder bore (mm)	82.5 mm
Cylinder stroke (mm)	92.8 mm
Compression ratio	10.4:1
Mass of vehicle (kg)	1060

A bench dynamometer model 2030 from Servitec with the capacity to measure up to 450 kW was used. For carrying out the tests on this dynamometer, the engine was removed from the vehicle and the fuel tank and external fuel pump were installed, as well as an electric power source for the electronic management. Thus, tests were performed to measure torque and power directly on the flywheel. Figure 1 illustrates the dynamometer and the engine under test.

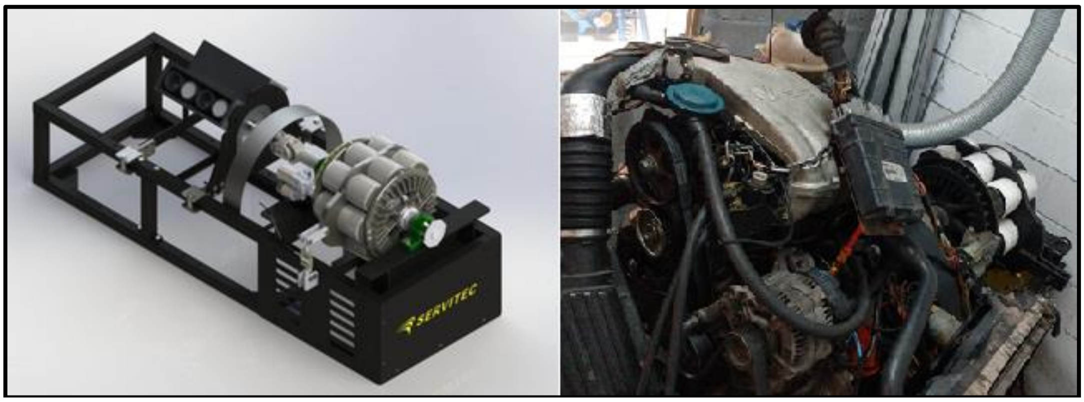

Figure 1. Tests on bench dynamometer.

Afterwards, torque and power measurements were performed on a Servitec model 2025 inertial roller dynamometer, as shown in Fig. 2. The tests were carried out in accordance with NBR ISO 1585: 1996 and comparisons were made in relation to transmission losses.

Figure 2. Tests on inertial roller dynamometer.

During the tests on inertial roller dynamometer with natural aspirated engine, the emission levels of HC, CO and CO_2 were also collected. These measurements were done following the CONAMA standard (2009) using a portable gas analyzer from Tecnomotor brand, model TM131.

Afterwards, the supercharge system was installed in the engine. From this point, the programmable electronic management module from Fueltech brand, model FT450, controlled the injection and ignition of fuel.

The tests performed with the supercharged engine in vehicle occurred with the maximum pressure gauge of the compressor discharge set to 80 kPa (0.8 bar). Four types of fuel were used: podium gasoline (100%); mixture of 50% in volume of podium gasoline and 50% in volume of hydrous ethanol, blend denoted by E50; mixture of 25% of podium gasoline and 75% of hydrous ethanol, whose blend was denoted by E75; and only hydrous ethanol, i.e., E100.

For the supercharged engine, torque, power and emissions tests were again performed and its results were then compared to the results of the naturally aspirated engine. All steps of the experiment are shown in Fig. 3.

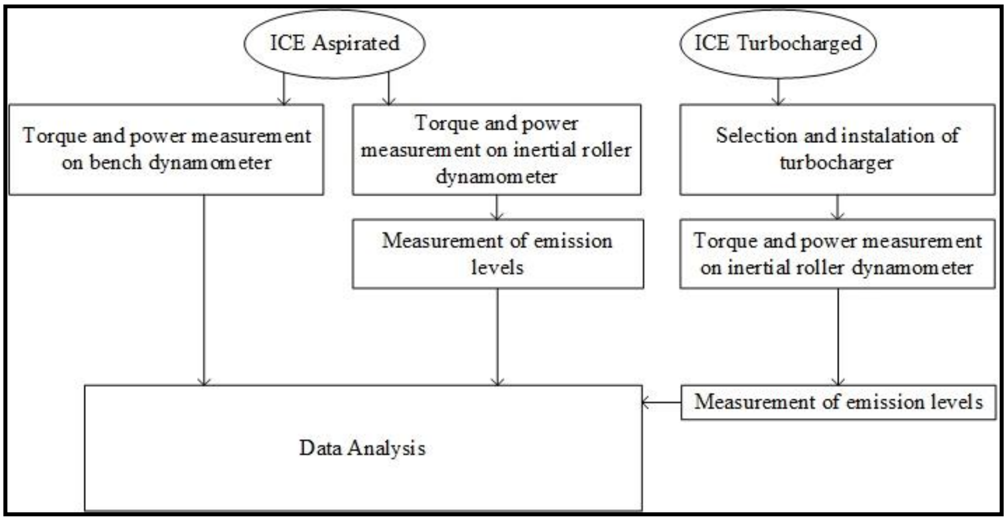

Figure 3. Performed activities flowchart.

2.1 Turbocharger selection

For the selection of a turbocharger to be installed in an ICE, it is necessary to know its constructive characteristics, establishing ways of using the vehicle. The manufacturers of the turbocharger system provide compressor performance maps (BELL, 1997).

According to Garrett® (2018), in order to find the operating points, the pressure ratio must be calculated using Eq. (2):

$$\Pi_c = \frac{p_{ec}}{p_{1c}} \tag{2}$$

where, Π_c is the pressure ratio, referring to the ordinate of the compressor efficiency map. In the abscissa axis there is the mass air flow, given by Eq. (3):

$$\dot{m}_{\text{air}} = \frac{V_{motor}}{1000} \cdot \frac{n}{60 \cdot 2} \cdot \rho_{ar} \cdot \eta_v \tag{3}$$

where $\dot{m}_{air}$ is the mass flow rate of air admitted to the engine, in kg/s; V_{motor} is the engine volumetric displacement, in liters; n is the engine speed in revolutions per minute (RPM); ρ_{ar} is the density of the air, in [kg/m^3]; η_v, is the volumetric efficiency of the engine in question (NAKANO, 2007). As described by Davis and Perkins-Davis (2002), the volumetric efficiency of naturally aspirated internal combustion engines is approximately 85%. Air density can be obtained from Eq. (4):

$$P = \rho_{ar} \cdot R \cdot T \tag{4}$$

where P is the pressure that the gas is subjected [Pa]; R is the gas constant [J/kg·K] e T is the absolute air temperature [K] (FOX, PRITCHARD, McDONALD, 2010).

According to Nakano (2007), the air compression process in this type of system can be considered as a reversible adiabatic process, for estimation of temperature increase; in other words, it is considered that the process is isentropic and demonstrated by Eq. (5).

$$\frac{T_1}{T_2} = \left(\frac{p_{1c}}{p_{2c}}\right)^{\left(\frac{k-1}{k}\right)} \tag{5}$$

where T_1 is the intake air temperature of the compressor [K]; T_2 is the discharge air temperature of the compressor [K]; p_{1c} is the absolute intake air pressure of the compressor; p_{2c} is the absolute discharge air pressure of the compressor; and k is the isentropic exponent, which for air is considered equal to 1.4.

The efficiency of the supercharger at different engine speeds was calculated. Defining the maximum discharge pressure of 80 kPa, the atmospheric pressure of 100 kPa, and the ambient air temperature of 25 ºC (298.15 K), the intake air temperature manifold, i.e., in compressor discharge ($T_{manifold\text{-}int}$), calculated by Eq. (5).

After defining the initial parameters: pressures p_{1c} = 100 kPa; p_{2c} = 180 kPa, volumetric efficiency of 85%; and engine volumetric displacement of 2 L; the compressor pressure ratio (Π_c), $T_{manifold\text{-}int}$ and air mass flow rate were determined. The calculated values are reported in Tab. 2.

Table 2. Compressor efficiency results.

Speed [RPM]	$P_{2c,\,man}$ **[kPa]**	$P_{2c,\,abs}$ **[kPa]**	Πc **[-]**	$T_{manifold-int}$ **[K]**	ρ_{ar} **[kg/m³]**	$\dot{m}_{air}$ **[kg/s]**
3500	80	180	1.80	352.67	1.779	0.0882
4000	80	180	1.80	352.67	1.779	0.1008
4500	80	180	1.80	352.67	1.779	0.1134
5000	80	180	1.80	352.67	1.779	0.1260
5500	80	180	1.80	352.67	1.779	0.1386
6000	80	180	1.80	352.67	1.779	0.1512
6500	80	180	1.80	352.67	1.779	0.1638

In sequence, the MasterPower R494-2 turbocharger was selected, with a compressor rotor diameter of 49 mm and a turbine of 49.5 mm. Its efficiency is shown in Fig. 4.

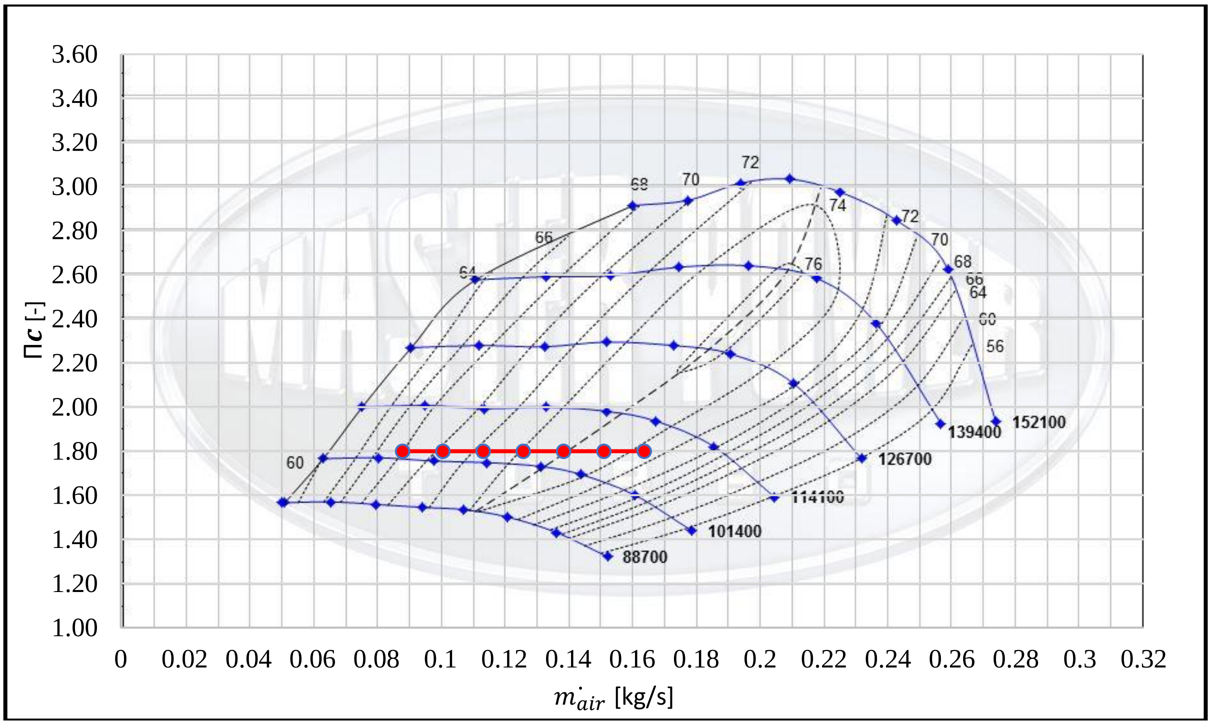

Figure 4. Operating point of the MasterPower R494-2 turbocharger.

2.2 Dimensioning of the fuel supply system

Using the air mass flow rate, it was possible to obtain the fuel mass flow rate from Eq. (1). Ethanol was considered in the design, with a stoichiometric, nearly 9 kg of air by kg of ethanol, as it is the highest fuel flow required in the operation. The result was used to purchase the fuel pump and injectors suitable for the engine to operate with greater fuel demand. Therefore, a fuel mass flow rate of 0.0214 kg/s was required.

Based on the results, a pump of the GTI 12 bar model from Dinâmica brand was chosen. This pump has a fuel flow rate of 0.0479 kg/s, operating at gauge pressure of 3.5 bar and 0.0455 kg/s at 4.2 bar. BOSCH injectors model 0 280 156 453, with maximum fuel mass flow rate of 0.0082 kg/s, were selected. All components have been dimensioned in such a way that still have some opportunities for improvements in vehicle performance.

2.3 Fuel injection map development

The development of the map to manage ignition and electronic fuel injection was carried out with the help of Fueltech's own control software. The λ values were corrected at full load on the inertial roller dynamometer, using the internal injection datalogger to record and analyze the data. This can be seen in Fig. 5 for values of λ, speed and test time.

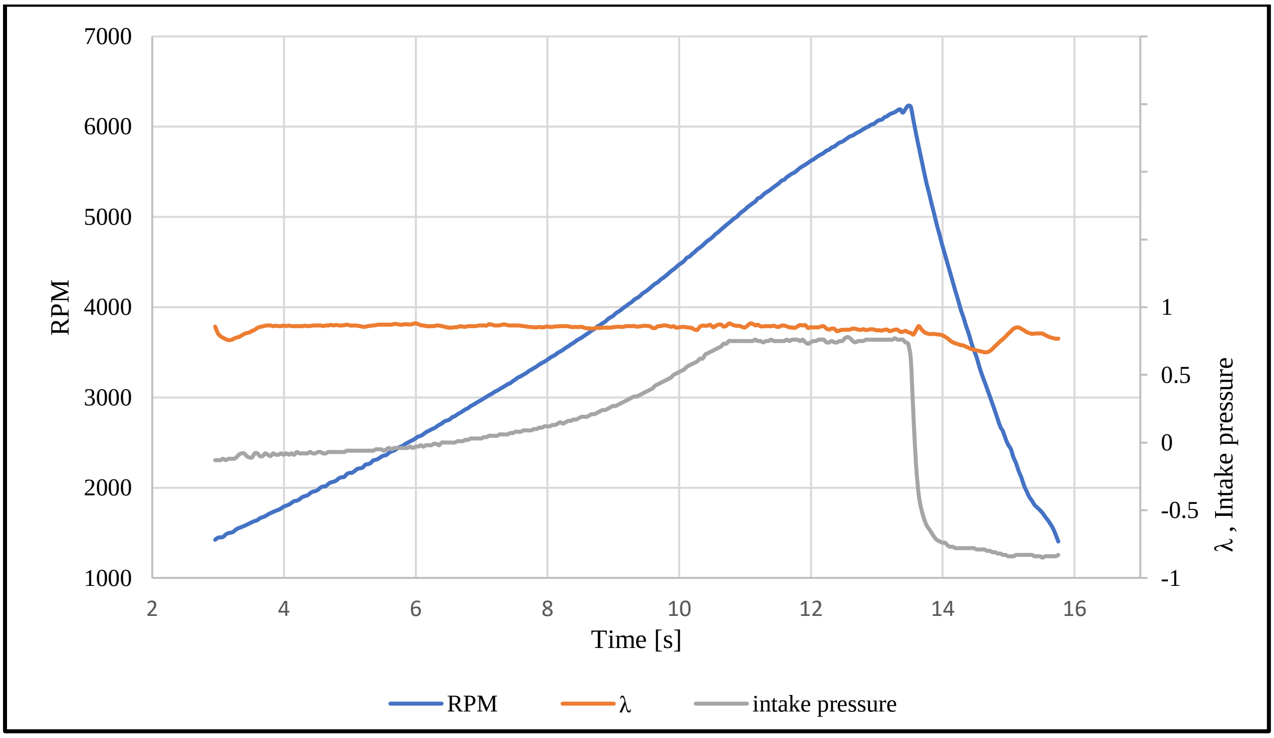

Figure 5. Values obtained from the programmable module's datalogger.

After the adjustment made with lambda values of approximately 0.85, as described by Cruz (2015) with the engine at full load and maximum power, the main fuel injection map was obtained as shown in Fig. 6. This map was generated with aid of the management module software. The necessary compensations are all made on the main map, such as percentage of fuel injected by percentage of the accelerator pedal pressed (TPS), by the amount of ethanol in the fuel mixture, and by temperature of the coolant.

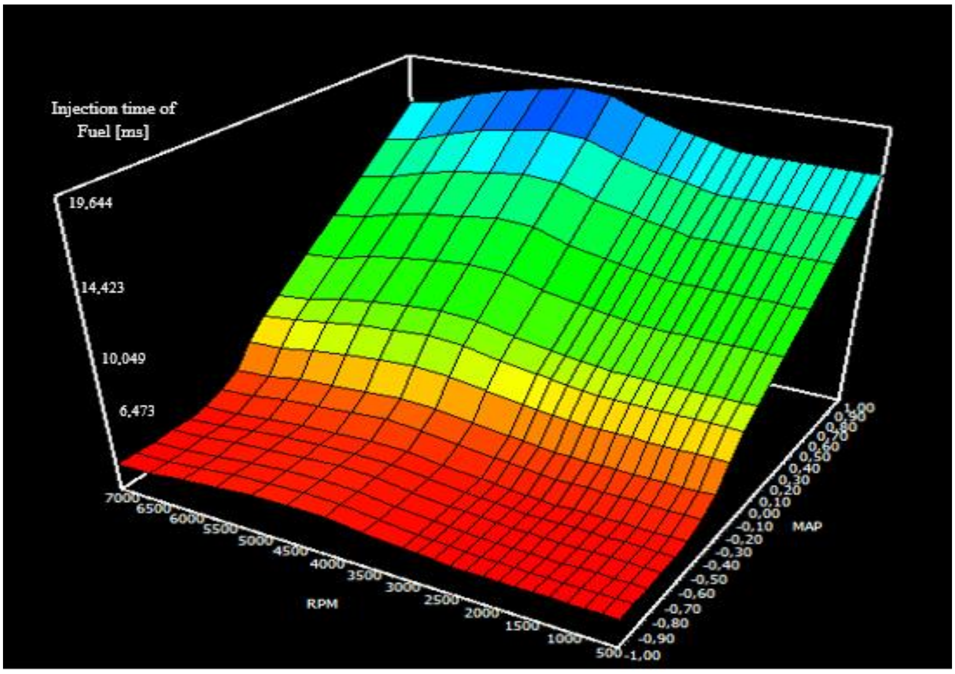

Figure 6. Main fuel injection map.

For the ignition map, the recommended by the software itself was used, which generates a conservative pattern due to the mechanical characteristics of the engine. In this case, no detonations occur in the combustion chamber.

3. RESULTS AND DISCUSSION

3.1 Torque and power results

Initially, the graph of torque and power of the engine operating with podium gasoline in the bench dynamometer and without supercharging, is shown in Fig. 7. The measurements were taken directly on the flywheel, being the peak power of 79.9 kW (108.6 cv) at 5500 RPM and torque of 148 N·m at 4000 RPM. The data on the vehicle wheels are also presented in Figure 7 obtained in the inertial roller dynamometer. In these tests, the main results were: peak power of 66.4 kW (90.3 cv) at 5400 RPM and the highest torque of 139 N·m at 3000 RPM.

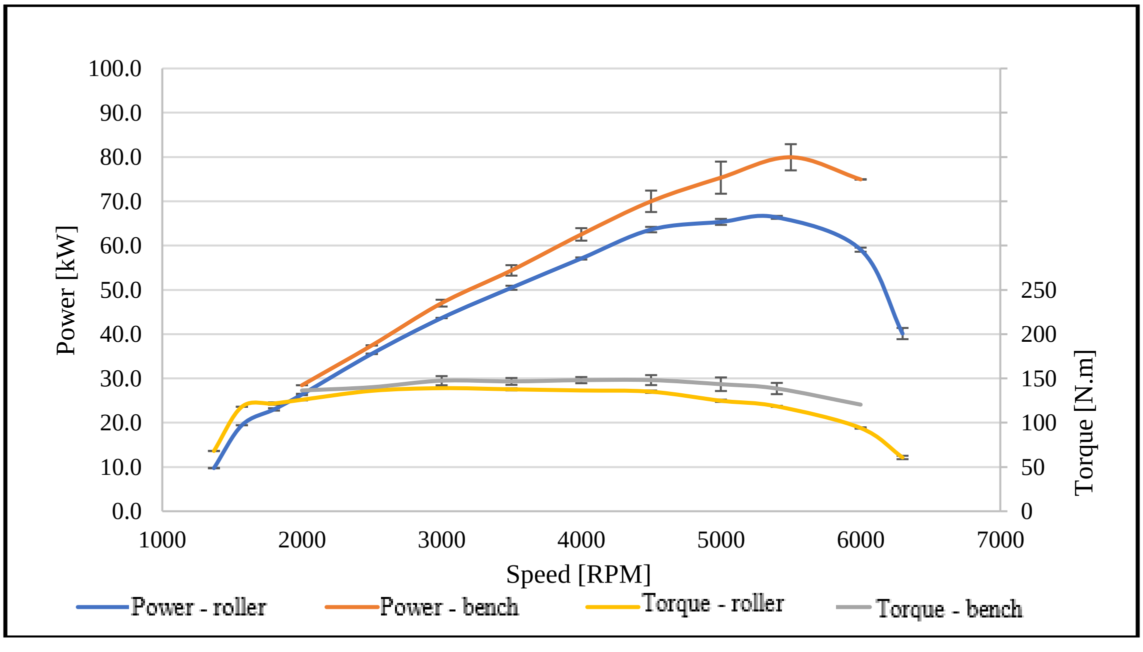

Figure 7. Comparison between result tests on bench and roller dynamometer.

The engine had a power difference to less than 5.1 kW, when compared to that specified by the manufacturer (85 kW), which is justified by the wear of the engine over time. A loss of power transmission of 16.9% is seen when compared to the measurements from flywheel and directly to the wheels. Kuniyoshi et al. (2010) found transmission losses of up to 21% in a formula SAE racing vehicle. Fact that occurs due to friction between gears, inertia of the entire rotating assembly as a clutch, transmission shaft and wheel, tire and bearing assembly.

Figure 8 shows the impact caused by the turbocharger assembly mounted on the engine, with a considerable increase in torque and power for all evaluated fuels. However, the E100 sample was the one that achieved the best result.

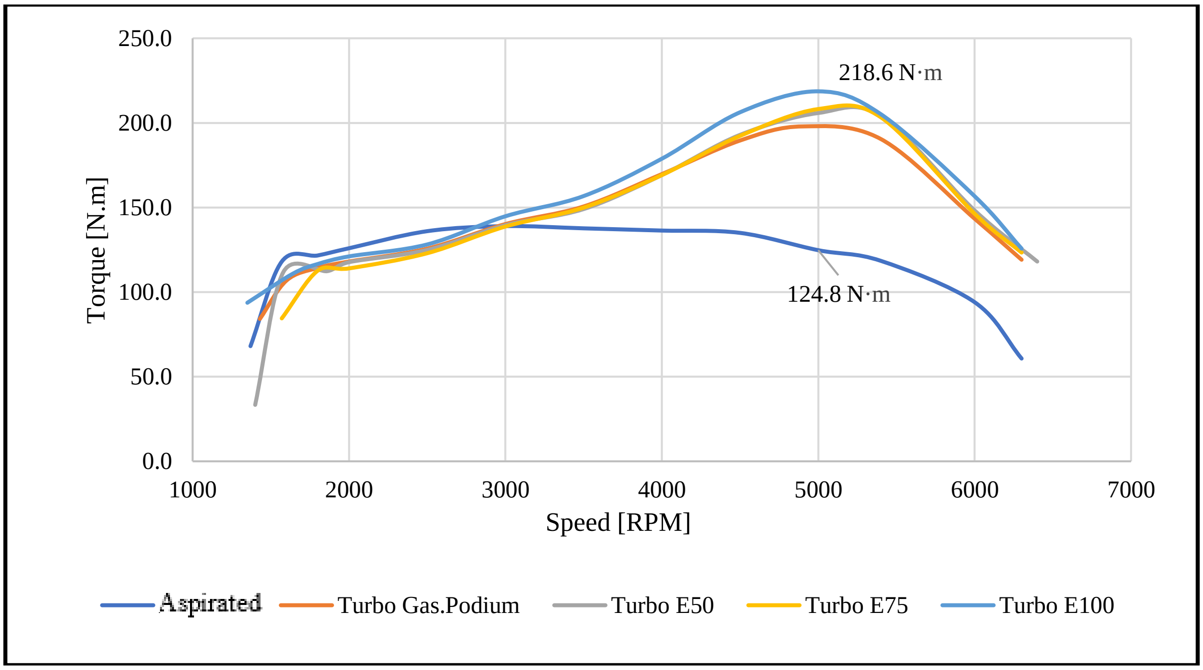

Figure 8. Torque curves per speed.

The aspirated engine has a higher torque than the supercharged engine up to 2700 RPM, due to the energy required to rotate the turbine that still insufficient, generating a back pressure in the cylinder with a turbocharger system. Similar to the torque, the power curves were plotted according to Fig. 9.

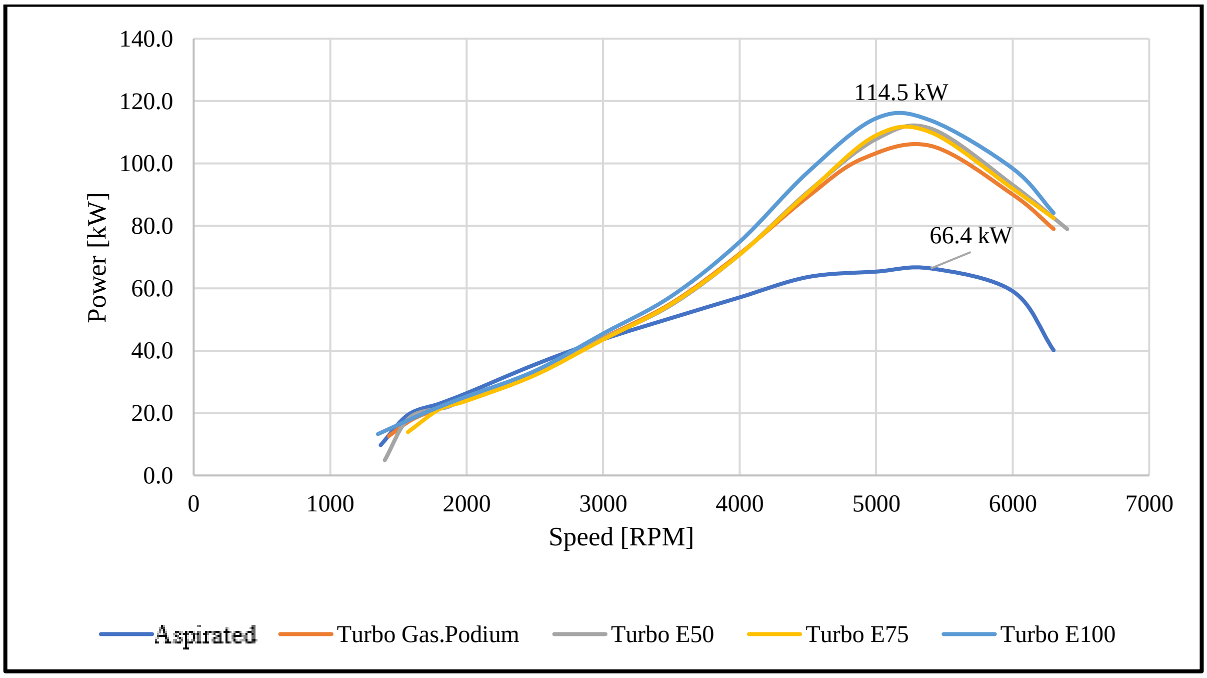

Figure 9. Power curves versus speed .

The highest percentage of power increase was observed when using E100, which was of 72.4%. As noted, the turbocharger increased the torque and power curves due to the increase in air mass flow. From 4000 RPM, the torque curve of the aspirated vehicle has decayed, and the turbocharger system compensates for this drop, making the engine able to promote greater torque up to around 5000 RPM.

Table 3 shows the main results obtained of torque and power for the engine tested in the roller dynamometer. Only the aspirated engine was measured on the bench dynamometer and found with this a percentage of transmission loss which was 16.9%. The transmission loss value was achieved by comparing the tests with the bench engine and afterwards with the vehicle in a roller dynamometer, obtaining the difference of 13.4 kW. The values of lambda were analyzed only at full load.

Table 3. Torque and power results in the roller dynamometer.

Condition	Fuel	Torque [N·m]	Power [kW]	Weight / Power[kg/kW]	λ at full load
Aspirated	Podium gas.	139.0	66.4	15.96	0.78
Turbo-charged	Podium gas.	197.9	105.7	10.03	0.85
	E50	205.8	111.2	9.53	0.85
	E75	204.2	109.2	9.64	0.85
	E100	218.6	114.5	9.26	0.85

The mass/power ratio is another important factor in vehicles that aim to increase performance. According to Nakano (2007), vehicles such as the Volkswagen Golf 2002 1.8 20 valves, turbocharged, originally had a ratio of 11.77 kg/kW, reaching a speed of 100 km/h in 8.2 s. Another vehicle analyzed was the Fiat Marea 2.0 20 valves, turbocharged, with a mass/power ratio of 10.84 kg/kW and taking 7.9 s to reach 100 km/h.

The vehicle under study reached the value of 9.26 kg/kW as the best result, less than those described in the previous paragraph. This is due to the fact that the total mass of the vehicle is only 1060 kg according to the manufacturer.

3.2 Emission index results

The emission levels are directly associated to the lambda factor. Higher the lambda factor, lower are the CO and HC indexes, however, higher CO_2 indexes were found. In order to obtain the lowest level of emissions, it was necessary to test mixtures with lambda factor closest to one at low engine loads. Table 4 shows the results obtained in emissions tests according to the CONAMA standard (2009) at 1000 RPM, and in Table 5 are identified the emission results for engine running at 2500 RPM.

Table 4. Emission indexes at 1000 RPM.

Condition	Fuel	CO [%]	CO_2 [%]	HC [PPM Vol]	λ
Aspirated	Podium gas.	0.92	14.0	128	0.97
Turbocharged	Podium gas.	1.06	11.1	183	0.96
	E50	1.27	6.6	204	0.95
	E75	1.34	12.0	129	0.95
	E100	0.96	10.6	107	0.97

Table 5. Emission indexes at 2500 RPM.

Condition	**Fuel**	**CO [%]**	**CO_2 [%]**	**HC [PPM Vol]**	λ
Aspirated	Podium gas.	1.03	14.1	119	0.97
Turbocharged	Podium gas.	2.72	12.1	184	0.91
	E50	2.61	11.9	141	0.92
	E75	1.81	12.3	119	0.94
	E100	2.16	12.1	93	0.93

As shown in the results of Tab. 3, regardless of the fuel used, there were great increases in torque and power in all tests with engine supercharged. In addition, there were increases in the emission levels with increasing speed, however, all were acceptable according to the CONAMA standard (2009).

4. CONCLUSION

Based on the results obtained in the dynamometer tests, it is evidenced experimentally that the supercharging system generates a considerable increase in power, without significantly increasing the levels of emissions. This is due to the fact that the turbocharger delivers a higher air mass flow to the cylinders, resulting in extra power when fully loaded.

With the results obtained experimentally, it is concluded that the greatest increase in torque and power occurred with the supercharged engine operating with E100. The measured torque was 218.6 N·m and power was 114.5 kW (155.7 cv) obtained in the roller dynamometer. However, considering the loss of transmission power as found with the aspirated engine, 137.74 kW (187.3 cv) was obtained in the supercharged engine operating with E100.

5. REFERENCES

Bell, C. *Maximum boost*. Cambrige: Bentley Publishers, 1997.

Brunetti, F. *Motores de combustão interna*. São Paulo: Blucher, v. 1, 2012.

Brunetti, F. *Motores de combustão interna*. São Paulo: Blucher, v. 2, 2012.

Bosch, R. *Manual de tecnologia automotiva*. Edgard Blucher, 2005.

CONAMA. Conselho Nacional de Meio Ambiente. Resolução n° 418, 25 de novembro de 2009. Publicado D.O.U, de 26/11/2009, Pág. 81-84.

Cruz, C.R. Desempenho de sondas lambda no monitoramento de motores do ciclo Otto alimentos por etanol e GNV. 2015. 109 f. Dissertação (Mestrado em Engenharia Mecânica) - UNESP - Universidade estadual paulista, Bauru, 2015.

Davis, E., Perkins-Davis, D. *Supercharging, Turbocharging, & Nitrous Oxide Performance*. Motorbooks, Minnesota, 2002.Fox, R.W., Mcdonald, A.T., Pritchard, P.J. *Introdução à Mecânica dos Fluidos*, 7° edição, LTC Editora, 2010.

GARRET®. *Manual Técnico: Performance catalog V.8*. <https://www.garrettmotion.com/wp-content/uploads/2018/05/Catalog-Volume-8_Full-Version_FINAL_web-version_pages.pdf>.

Gheorghiu, V. *Ultra-Downsizing of Internal Combustion Engines*. SAE Technical Paper, 2013.

Kuniyoshi, A., Almeida, M. C., Terra, R.T., Capato, R. M. *Perda de potência em um de transmissão veicular*. Revista Ciências do Ambiente On-Line, v. 6, n. 2, 2011.

Martins, J. *Motores de combustão interna*. Publindústria, 2006.

Nakano, D.G. *Estudo Sobre Instalação De Um Turbocompressor Em Automóvel Nacional*. 2007. Dissertação para obtenção de título de Mestre Profissional em Engenharia Automotiva. Escola Politécnica da Universidade de São Paulo–USP.

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