Design of a Test Bench for Micro Combustion Engines

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Abstract

This study is about the design, construction and instrumentation of a test bench for testing micro combustion engines used in model vehicles. Several sensors and actuators were used to obtain its performance specifications. There is no standard test bench for micro engines, so this equipment intends to present drivers with viable information, something that is missing. Several technical requirements were considered for the test bench to be safe, effective and practical to use. Some of the components were manufactured while others were off-the-shelf and, at a final stage of the construction, the test bench was assembled with all the required model parts. One of the objectives is to characterise a 3.5 cc micro engine. For that, the test bench was designed to obtain the power and torque diagrams and the fuel consumption rate. These results are displayed on a PC through a data acquisition board able to conduct all the necessary readings and processing for characterisation.

First a conceptual design was made to assess the idea behind the project. After a thorough research on test benches and micro engines, the concept gave way to the actual design of an inertia dynamometer. A program in *LabView* was developed for the data acquisition system including the actual control of the actuators. Components were produced in Instituto Superior Tecnico and, finally, the test bench was assembled with the required flywheel and all the necessary electronic components. Tests were made for an electric motor driving a gyroscope to validate the instrumentation and then the 3.5 cc glow plug engine was tested, whose results of the power curves validated the whole design of the test bench.

The inertia dynamometer is viable and flexible, able to test any kind of micro engine from 2.0 to 20.0 cc. The program running the acceleration tests is fully automatic, allowing the test bench to have good repeatability which is essential to get realistic data.
Keywords: Power curves, inertia dynamometer, micro engines, performance, data acquisition.

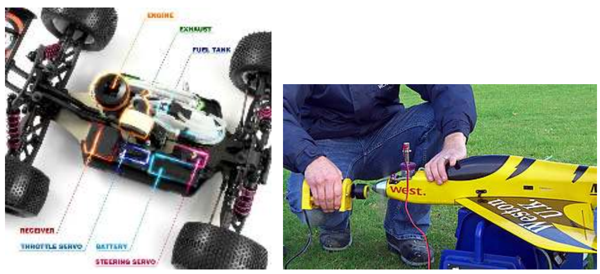
1. Introduction

There is a growing need to characterise micro engines. At this point, there is no standard test bench for testing them. The manufacturers fail to provide realistic performance specifications and the objective of this thesis is to develop an equipment capable of doing it for any kind of micro engine.

1.1. Micro Engines

Micro engines are internal combustion engines [1] that go up to 20.0 cc of displacement and are largely used in radio-controlled model vehicles (Figure 1), mostly for racing purposes. There are three types of micro engines: spark ignition, diesel and the glow plug, which offers the highest power to weight ratio and is the primary choice for most of the model racing car drivers.

These engines can go up to 45 000 rpm [2], so they have a very well-defined flow pattern within the combustion chamber characterised by a loop scavenging in the two-stroke engines. The admission of fuel and air is accomplished by a variable-venturi carburetor with a slide that is actuated by an electric servo responsible for throttle control.



(a) Model racing car with assembled micro engine.
(b) Starting the Glow Plug engine of a model airplane.

Figure 1: Applications of micro engines.

1.2. Test Benches

There are two main types of engine test benches for power assessment, commonly known as "dynos" (from dynamometer): the inertia dynamometer and the steady-state dynamometer. The inertia dynamometer works by having the engine accelerate an inertial mass (often called flywheel) during a certain amount of time until the torque evolution is fully collected [3]. Steady-state dynamometers use a brake (also named absorber or retarder) to apply a load to the engine and hold it at a constant speed against the open throttle, where an electronic "load cell" reads the torque transmitted to the brakes [4]. To characterise micro engines, it will be used an engine inertia dynamometer. It conducts an acceleration test more similar to the real world, while cheaper and simpler (no need for large braking systems). The flywheel simulates the inertia experienced by the engine [5]. The torque is given by

$$T[N.m] = I[Kg.m^2] \times \alpha[rad/s^2]\,, \tag{1}$$

where I is the moment of inertia of the flywheel and shaft all together and α is its angular acceleration.

The main goal is to obtain the power (P) diagram, which is proportional to torque (T) and rotation speed (ω) of the drive shaft. By measuring both, power is calculated as

$$P[W] = T[N.m] \times \omega[rad/s]\,. \tag{2}$$

2. Background

Before beginning the design of the inertia dynamometer, technical requirements must be defined accordingly to the objectives for this test bench.

2.1. Engine Specifications

The type of micro engine to be tested for this dissertation is the glow plug of a car. According to the *O.S. Engines* manufacturer [6], the 4.5cc for truggies is one of the most powerful engines and yet largely used, having a maximum crankshaft power output of 3.26 HP (2.43 kW) at 32 000 RPM, resulting the estimation of maximum torque

$$T_{engine} = \frac{P \times 60}{2 \times \pi \times N} = 0.725 N.m \tag{3}$$

A safety factor of 1.5 will be used to ensure that the test bench withstands the test of any kind of micro engine from 2.0 to 20.0 cc.

2.2. Technical Requirements

The drive shaft must have a given amount of inertia for the engine to complete a run in, approximately, 10 seconds (recommended value for Inertia dynamometers) [3].

For practical reasons, the flywheel should not exceed 300mm of diameter and 10Kg of mass. It must be perfectly balanced to ensure maximum stability at the highest RPMs.

For safety and balancing reasons, the rotational speed of the drive shaft must not exceed 10 000 rpm. A reduction gear needs to be implemented, given the engine maximum RPM greatly exceeds this value.

The technical requirements covered in this section are summarized in Table 1.

Description	**Parameter**	**Value**
Engine displacement vol.	D	2.0 - 20.0 cc
Maximum power	P_{max}	2.5 kW
Maximum torque	T_{max}	0.750 Nm
Safety factor	n	1.5
Engine max rot. speed	N_{max}	40 000 rpm
Engine idle rot. speed	N_{idle}	10 000 rpm
Engine max temperature	T_{max}	130 C
Run acceleration time	Δt	10 s
Flywheel diameter	$d_{flywheel}$	$\leq$ 300 mm
Flywheel mass	$m_{flywheel}$	$\leq$ 10 Kg
Drive shaft rot. speed	$N_{driveshaft}$	$\leq$ 10 000 rpm

Table 1: Technical requirements for the engine dynamometer.

2.3. Auxiliary Systems Required

A braking system has to be implemented in the test bench to stop the flywheel after a run. It should also stop it fast enough to save time between runs, allowing the dynamometer to be more efficient.

The temperature at the engine cylinder head should not exceed 130C, according to some drivers reports. As such, the engine needs to have a cooling system. The micro engines are cooled by air, so a fan cooler needs to be considered.

A device to actuate the throttle must also be implemented, to control the engine load.

Micro engines are started by a 12V electric motor connected to its crankshaft allowing it to rotate and start burning fuel [2]. A device must be set to make the coupling on the crankshaft wheel and allow the electric motor to kick back at the moment the engine starts. At the same time, the glow plug must be heated by a 1.5V glow starter. This requires electric power for both the electric motor and the glow starter.

2.4. Drive Shaft and Components Configuration

The drive shaft for the test bench, comprising the reduction gear and the flywheel, will be a solid steel shaft to prevent deflection, keeping the diameter as small as possible [7]. Three ball-bearings supports will secure it, two of them set close to the flywheel to minimize deflection. To attach the flywheel, an hub will be used with a cubic shape. A screw is used to lock it on the shaft in a section where a groove is set to receive the screw, preventing it to slide.

The flywheel will be a balanced disc with an inner-disc and a thicker outer-ring. This way it will be lighter for a given radius (R) than it would be if designed as a regular disc. The moment of inertia for a flywheel this shape is given by

$$I = \frac{1}{2}\, m_{disc}\, {R_1}^2 + \frac{1}{2}\, m_{ring}\, ({R_1}^2 + {R_2}^2) \quad (4)$$

being R_1 the radius of the inner-disc, and R_2 the outer radius of the flywheel.

The reduction is accomplished by a spur gear that reduces the speed of the drive shaft according to a gear ratio given by

$$i = \frac{\omega_1}{\omega_2} = \frac{\phi_2}{\phi_1}\,, \quad (5)$$

where ω_1 and ϕ_1 are the engine crankshaft rotation speed and diameter, and ω_2 and ϕ_2 are the drive shaft rotation speed and diameter.

To prevent engine overheating, a one-way bearing is used. This way, the flywheel overruns the engine crankshaft after a run, when the throttle is closed for deceleration, preventing engine braking that could damage it [8].

2.5. Auxiliary Systems Configuration

The braking system is similar to the one used in model cars. A small disc is attached to the drive shaft with two pads right next to it. One pad is static, the other one is actuated by a rotating pin, located by the shaft support, which will cause the pad to press the disc and, eventually, stop it within the two parallel pads. In Figure 2 is possible to see such system.

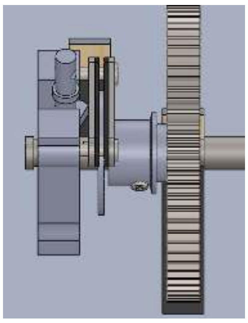


Figure 2: Braking system.

To give the required degree of freedom for positioning the engine in the test bench, it has to comprise adjustable supports.

For throttle and brake control, the electric servo used in model cars will be applied, which directly controls the position of both the throttle and the brake pads by actuating the mechanical arm connected to the carburetor and the rotating pin on the other end, respectively.

In Figure 3 is possible to see a previous layout of the test bench.

3. Design, Assembly and Wiring Layout

The material chosen for the drive shaft was the cold drawn AISI 1020 for its good mechanical properties

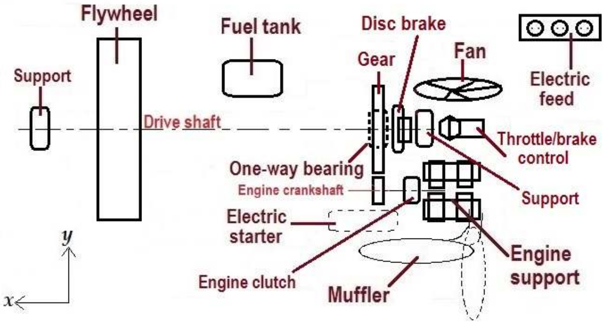


Figure 3: Layout of the test bench.

and relatively low cost. It has a yield strength of 390 MPa and a tensile strength of 470 MPa [7].

The flywheel will be made of steel. Other materials with lower densities would cause it to be much bigger to reach the required inertia. The frame to help support the ground components of the test bench will be a stainless steel sheet mounted on a wooden table. The supports for the drive shaft and the gear itself are made of a hard and light composite, implemented from *XRay* model vehicles [9].

3.1. Drive Shaft

To size the drive shaft, it is important to know the maximum torque applied to it. Remembering the maximum of N = 10 000 rpm for the drive shaft, and that the maximum rotation speed of micro engines is N = 45 000 rpm, the minimum gear ratio (i) given by Equation (5) is

$$i_{min} = \frac{45000}{10000} = 4.50\,.$$

To prevent the drive shaft to reach its maximum speed, it is being used a more conservative gear ratio of **i = 5**. Therefore, the applied torque on the drive shaft is given by

$$T_{transmitted} = T_{engine} \times i = 3.75Nm\,. \quad (6)$$

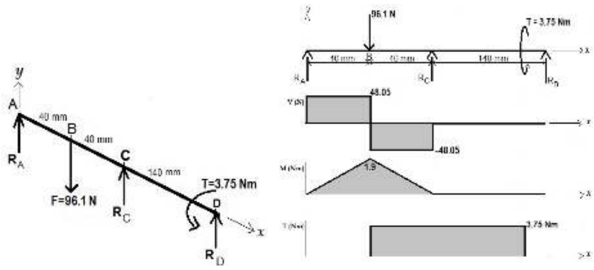
The AISI 1020 CD steel has a yield strength of $\sigma_y = 390MPa$. Considering the safety factor $n = 1.5$, the maximum stress σ_{max} is given by

$$\sigma_{max} = \frac{\sigma_y}{n} = 260MPa \quad (7)$$

The torque transmitted from the engine causes a torsional moment on the drive shaft and, due to the flywheel weight, it will also experience a bending moment. The shaft will have an effective length of **L = 220mm**. It needs to be this long to ensure the flywheel does not interfere with other components, namely a 180 muffler or the starter below the main frame.

Conducting an equilibrium analysis, it is possible to draw the load diagrams and assess the maximum torque and moment suffered by the shaft. From Section 3.2 in this chapter, it is known that the flywheel will have a mass of 9.81 Kg, which corresponds to a Weight of **W = 96.1 N**, considering $g = 9.8\, m/s^2$. The shaft is supported by simple ball bearings (points A, C and D) restraining it only in the y direction. The flywheel is set at just 40mm from supports C and D. This way it causes a low bending moment.

The weight of the gear is being neglected because this is made of a light composite. Besides that, the bending moment is minimal because it is set right next to support D. The free-body diagram can be seen in Figure 4 a).



(a) Free-body diagram. (b) Shear, bending and torsional moments.

Figure 4: Load diagrams.

Using the equilibrium equations for the applied loads to acknowledge R_A and R_C, it is possible to calculate the shear forces (V), torque (T) and bending moment (M) to draw its load diagrams,

$$\sum M_C = 0 \Leftrightarrow R_A = \frac{0.04}{0.08} \times 96.1 \Leftrightarrow R_A = 48.05N$$

$$\sum F_y = 0 \Leftrightarrow R_C = 96.1N - R_A \Leftrightarrow R_C = 48.05N$$

The load diagrams for the drive shaft can be seen in Figure 4 b). Torque is constant between the gear and the flywheel and equal to 3.75 Nm and the maximum bending moment is given by $M = R_A \times 0.04 \Leftrightarrow M_{max} = 1.9Nm$.

There is no thrust load and since the shaft complies to the long shaft criteria ($\frac{L}{d} > 10$), shear forces (V) are negligible.

Considering both T and M in the most solicited section of the shaft (B), the minimum diameter for static analysis, according to Von Mises theory [7], is given by

$$d_{min} = (\frac{32\, n}{\pi}\, \sigma_y \sqrt{M^2 + \frac{3}{4} T^2}\,)^{\frac{1}{3}} = 5.3mm \quad (8)$$

where d_{min} is the minimum diameter, σ_y is the yield strength, M is the maximum Moment, T is the maximum Torque. As such, a shaft with 6mm of diameter satisfies the criterion.

After a thorough research regarding these model vehicles, a solution for the drive shaft and the attachment of its components is found. The drive shaft must have 8mm of diameter to comply with the components found for the braking system. It is a AISI 1020 steel provided by *Poly Lanema*.

As such, the shaft diameter exceeds the previously suggested 6mm. The safety factor should be much higher than the original value of 1.5,

$$n = d^3 \times \frac{\pi\, \sigma_y}{32\sqrt{M^2 + \frac{3}{4}\, T^2}} = 5.2 \quad (9)$$

In Figure 5 it is possible to see the solution for the drive shaft of the test bench.

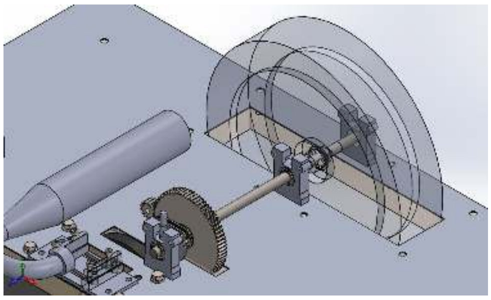


Figure 5: Drive shaft and its components.

3.2. Flywheel Specifications

To calculate the required inertia of the flywheel, the angular acceleration (α) must be estimated from ω_0 to ω_f.

$$\omega_f = \omega_0 + \alpha \times \Delta t \quad (10)$$

From the technical requirements, it is known that the common micro engine goes up to 40 000 rpm and that the idle speed is, usually, 10 000 RPM. Considering the gear ratio, i=5, $\omega_0 = \frac{10000}{5} = 2000\, RPM = 209.44\, rad/s$, $\omega_f = \frac{40000}{5} = 8000\, RPM = 837.76\, rad/s$.

Considering $\Delta t = 10\, s$ from the technical requirements, results an average angular acceleration of $\alpha = 62.83\, rad/s^2$.

Consequently, the desired moment of inertia of the flywheel is given by Equation 1,

$$I = \frac{T}{\alpha} = 0.0597\, Kg.m^2$$

The flywheel can have up to 300 mm of diameter and the mass must not exceed 10 Kg. It will be sized down until a fair equilibrium of mass and radius is found. From Equation (4), an iterative process is made to assess the required dimensions of the outer ring to ensure the maximum of 10 Kg for the flywheel. The results are shown in Table 2. The density of the steel is $7850 Kg/m^3$, accordingly to the supplier *Ramada Acos*.

Thickness e_o	45 mm
R_1	80 mm
R_2	105 mm

Table 2: Dimensions for the outer ring of the flywheel.

The mass of the outer ring is given by

$$m_{ring} = \rho\, e_o\, \pi\, ({R_2}^2 - {R_1}^2) = 5.12\, Kg \qquad (11)$$

and $m_{disc} = 4.71\, Kg$. So the total mass of the flywheel with a radius of 105 mm is equal to 9.83 Kg.

3.3. Braking System

It will be used a disc brake similar to the system used in model vehicles. Knowing the brake pads actuate at uniform pressure $p = p_a$, and simplifying the contact area to $A_c = \frac{\pi}{4} \times (R^2 - r^2)$ for an equivalent annular pad of 90, the braking force [7] is given by

$$F = p \times A_c = p_a \times \frac{\pi}{4} \times (R^2 - r^2) \qquad (12)$$

where p_a is the applied pressure by the pads, R the outer radius = 15mm, and r the inner radius = 7mm.

In Figure 6 a) is possible to see the dimensions of the pads used for the braking system and its equivalent annular pad actuating the disc.

The contact area for each pad is $A_c = \frac{\pi}{4}(R^2 - r^2) = 138.23\, mm^2$. The larger it is, the lower is the pressure for the required force. Integrating the product of the friction force and the radius, the Torque is found by

$$T = \frac{\pi}{2}\mu p_a \times \int_r^R r^2\, \mathrm{d}r = \frac{\pi \mu p_a}{6} \times (R^3 - r^3) \qquad (13)$$

$$T = \frac{\pi \mu p_a}{6} \times (R^3 - r^3)$$

where μ is the friction coefficient. Considering both materials (ferodo and steel), $\mu = 0.4$ at 150 C.

Since there will be used two pads on either side of the disc, equation (13) is multiplied by two (the number of pairs of mating surfaces).

For T = 3.75 Nm, $p_a = 2.95 MPa = 29.5 bar$.

The equation can be written to relate torque with the braking force,

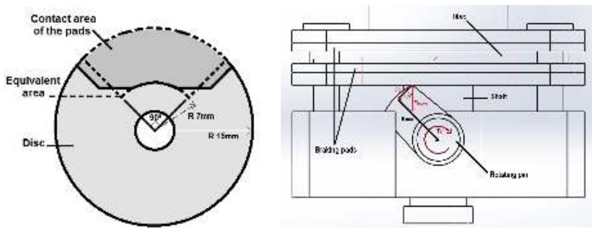
$$T = \frac{4}{3} F \mu \times \frac{R^3 - r^3}{R^2 - r^2}\,, \qquad (14)$$

Resulting for the ideal force

$$F = \frac{3T}{4\mu} \times \frac{R^2 - r^2}{R^3 - r^3} = 408N\,. \qquad (15)$$

This force of 408 N would be capable to stop the flywheel immediately, which will not be the case. A smaller force does the work by stopping it in a few seconds, while the generated friction in the brake pads is absorbing the kinetic energy dissipated by heat through the disc.

The electric servo for throttle and break control generates up to 13.6 Kg.cm of torque (T). This quantity is transmitted to the brake pads by a small arm connected to the rotating pin that actuates the brake (Figure 6 b)).



(a) Dimension of the braking pads.

(b) Braking force.

Figure 6: Braking system.

Knowing $1\, Kg.cm = 0.098\, N.m$, the torque generated by the servo is equal to 1.33 Nm (T_1). The servo arm has a radius (R) of 20mm, so the respective force is given by

$$F_1 = \frac{T_1}{R} = \frac{1.33}{0.02} = 66.64N\,. \qquad (16)$$

The force F_1 actuates the pin arm (R = 10mm) transmitting a torque (T_2) to the rotating pin given by

$$T_2 = F_1 \times 0.01 = 0.6664Nm\,. \qquad (17)$$

The rotating pin actuates the braking pads at a 45 angle with $F_2 = \frac{0.6664}{0.006} = 111.1N$. The torque T_2 causes a moment in the 6mm ledge of the pin that actuates the brake pad. Projecting F_2 to a normal force experienced by the ledge, it is possible to obtain the braking force,

$$F_{brake} = \frac{F_2}{cos(45)} = 157.1N \qquad (18)$$

This is 38.5% of the ideal 408 N. It is sufficient to stop it in a few seconds, so it is viable to use the braking system of model vehicles on the test bench. Note that the braking must be continuous to prevent an abrupt stopping of the flywheel, which could cause the test stand to flip over.

3.4. Engine Support
The engine support blocks will be set on the table through two parallel slots. The design takes into account the technical requirements for the engine dimensions. In Figure 7 it is possible to see the solution for locating the support blocks to lock the engine with a 10mm freedom on both x and y directions, in case different gear meshes are used or larger engines are tested.

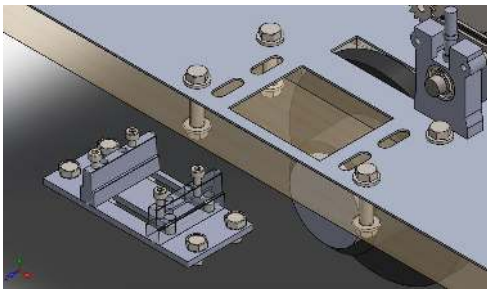


Figure 7: Engine support.

3.5. Starting System
For setting the starter, a pedal is designed to accomplish the contact between the electric motor (Figure 8 a)) and the engine crankshaft. It is set below the test stand, secured by an hinge locked on the wooden table. The operator actuates it by pulling an handle set on the opposite side of the micro engine.

To heat the glow plug, it will be used a portable "glow starter" set at 1.5V, as shown in Figure 8 b).

(a) Electric motor (starter).

(b) Glow starter.

Figure 8: Starting system.

3.6. Deflection Analysis
The shaft is designed to withstand the weight of the flywheel with a conservative safety factor, so the displacement should be low because the material is stiff. To conduct the analysis, a simulation in *SolidWorks Simulation* was made by a finite element method. This static analysis intends to assess the maximum displacement experienced by the shaft. A force of 96.1 N is applied at the flywheel section (y direction).

In Figure 9 it is possible to see the deflection experienced by the drive shaft. Note that graphical representation has a magnifying factor of 200 for visualization purposes.

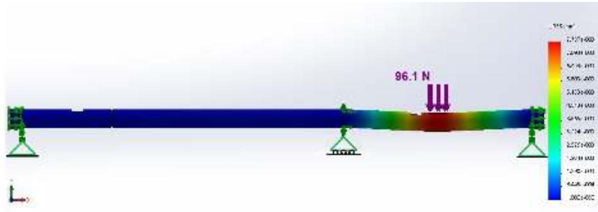


Figure 9: Displacement on the drive shaft.

The maximum displacement occurs, naturally, in the flywheel section. This value is equal to 0.008 mm, as it can be seen in Figure 9.

3.7. Structural Assembly
The flywheel was manufactured in Instituto Superior Tecnico using both turning and milling.

The first components to assemble on the drive shaft are the supports, locked on the test stand, the braking system and the one-way bearing securing the gear. In Figure 10 a) it is possible to see the assembled braking system.

(a) Braking system.

(b) Flywheel assembled.

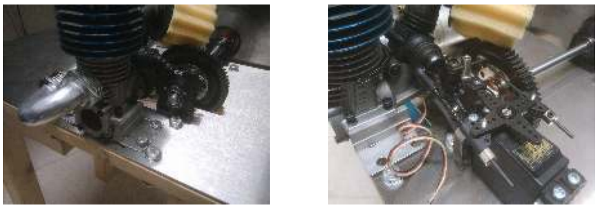
Figure 10: Drive shaft completed.

Next, the flywheel is assembled (Figure 10 b)). After it is in position, the hub must be locked ensuring the flywheel does not slip.

The engine support is assembled on the respective slots cut on the main frame. Its two support blocks are locked by four M4 bolts (Figure 11 a)) and the engine is now locked in position as well. The muffler is secured by a wire, previously set on the main frame very close to the flywheel but without interference.

The electric servo for throttle and brake control is set on the side of the engine as illustrated in Figure 11 b). The fuel nipple angle, connected to the throttle arm, can be adjusted ensuring both arms are at the same height.

The electric starter is set below the test stand (Figure 12 b)).



(a) Engine support. (b) Servo for throttle and brake control.

Figure 11: Assembling of the engine and servo.

(a) Assembly completed. (b) Assembled starter.

Figure 12: Test bench fully assembled.

3.8. Selection of Sensors and Actuators

The required sensored quantities and their measuring ranges are summarized in Table 3.

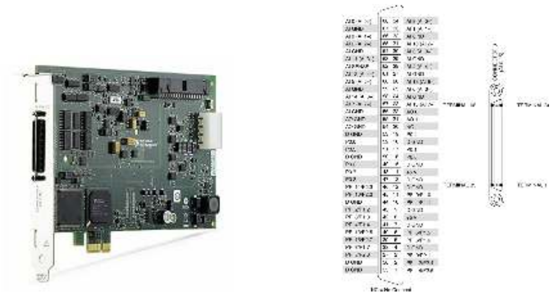
Physical quantity	**Range**
Rotation Speed	3 000 - 45 000 RPM
Volumetric Flow	20 - 80 mL/min
Engine Temperature	0 - 150 C
Room Temperature	10 - 40 C
Atmospheric Pressure	800 - 1100 mbar

Table 3: Sensored quantities and their measuring ranges.

To read all the sensors outputs and record them, a data acquisition (DAQ) system will be used. This system will be comprised of a data acquisition board connected to a computer. The DAQ board, previously acquired, is the National Instrument NI PCIe-6321 [10] (Figure 13), which has 16 analog +/-10 V inputs and outputs, connected to a computer running LabView, to read, process and record the data acquired from the sensors, and to control the cooling fan and throttle servo.

To measure the RPM of the crankshaft wheel and the flywheel,**CNY70**, a reflective sensor was chosen (Figure 14 a)). The phototransistor measures a frequency (Hz) which multiplied by 60 (seconds in a minute) will give the exact number of RPM of the engine.

To measure the volumetric flow in the fuel line to assess fuel consumption, a flow meter was chosen from *Omega*. The flow sensor to implement is the **FTB311-EU**, capable of measuring flow rates from



(a) Board NI PCIe-6321. (b) Pinout schematic for the NI PCIe-6321.

Figure 13: Data acquisition board for the test bench (PCIe-6321).

30 to 300 mL/min (Figure 14 b)). It also measures a frequency (Hz) which multiplied by a factor will give the volumetric flow in the required unit.

(a) RPM sensor. (b) Flow sensor.

Figure 14: Frequency sensors.

One temperature sensor is necessary to measure the temperature of the engine outer head and another one to measure room temperature. The chosen was the **LM35DZ** (Figure 15 a)), a thermocouple able to measure temperatures from -40^0C to 150^0C from *Texas Instruments*.

To measure room pressure, an analog absolute pressure sensor is required to connect to an analog input of the DAQ board. The chosen was the *KP236N6165* from *Infineon* (Figure 15 b)) that can measure absolute pressures from 60KPa (600mbar) to 165KPa (1650mbar).

(a) Temperature sensor. (b) Pressure sensor.

Figure 15: Voltage sensors.

There are two electrical actuators required for the test bench to work: the electric servo for throttle and brake control and a relay to switch the fan on and off.

The servo for throttle and brake control is similar to those used in model cars. It was chosen the **Futaba S9302**, a brushless electric servo with a torque of 13.6 Kg.cm and a speed of 0.11s/60. It can be seen in Figure 11 b).

For fan control a relay was implemented. It was chosen the **EC-500**, a DC5V relay module (Figure 16 a)). It is high level TTL (5V) and needs a trigger current of not less than 5 mA, which complies with the drive current of the data acquisition board. The fan used for engine cooling is a *Corsair AF120* with a dimension of 120 x 120 x 25mm (Figure 16 b)).

(a) 2-channel 5V relay.

(b) Fan for engine cooling.

Figure 16: Components for engine cooling.

3.9. Cost breakdown

In Table 4 it is possible to see the cost of the whole equipment including mechanical components and data acquisition system.

	Component	**Cost** [*e*]
Mechanical components	Flywheel	45
	Drive shaft, supports	25
	Electric starter and accessories	60
	Hinge	5
	Hubs, gear and one-way bearing, bearings	55
	Fuel tank	20
	Main frame sheet	40
	Bolts, nuts, rings	15
Data acquisition system	DAQ board, shielded cable, connectors block	1240
	Servo	50
	Cooling fan	15
	12V DC battery	10
	5V Relay	5
	Flow sensor	185
	Rotation sensors	5
	Temperature sensors	5
	Pressure sensor	10
	Breadboard and cables	10
	TOTAL	**1800**

Table 4: Cost breakdown of the test bench.

3.10. Wiring Layout

The sensors are supplied by the DAQ board itself (5V) (except for the flow sensor (12V). For external supply of the actuators and the flow sensor a 12V DC battery is used. The detailed wiring layout of all the sensors and actuators, implemented in the test bench, can be seen in Figure 17.

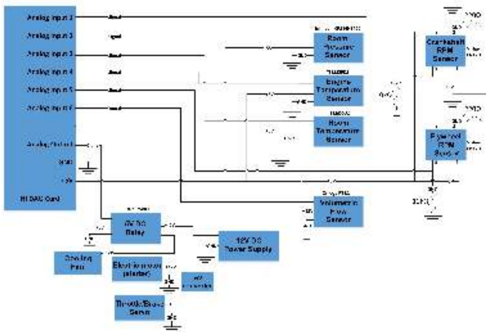


Figure 17: Electrical layout of the test bench.

3.11. User Interface

The operator of the inertia dynamometer has to be aware of the data being collected from the test bench. So a graphical interface on the software *LabView* was created to show the power and torque diagrams, the fuel consumption rate and other sensored quantities for the acceleration run at hand. This interface can be seen in Figure 18.

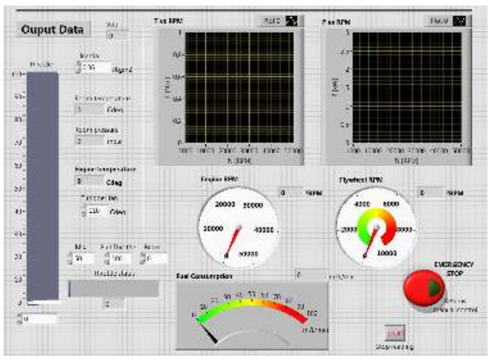


Figure 18: LabView interface for the dyno operator.

There is a series of graphical and numeric indicators, from which the dynamometer operator can assess: **maximum power**, **maximum torque**, **maximum rotation speed**, **clutch losses**, **fuel consumption at idle speed and full throttle**, and the **time to warm up the engine**.

At the end of a run, the operator must select the Stop button for *LabView* to cease data collection. The flywheel RPM must never reach 10 000 RPM for safety issues. In that case, the operator must press the "emergency stop" button immediately, which will actuate the brake and return manual control for the operator to cease braking when he sees fit.

The throttle control was programmed to be fully automatic. The operator can decide the duration of the run, meaning when to start full throttle, when to end it and even the duration of braking. The throttle position alternates between 0 (break), 50 (idle) and 100 (full throttle) as it shows in the throttle slide of the interface (Figure 18).

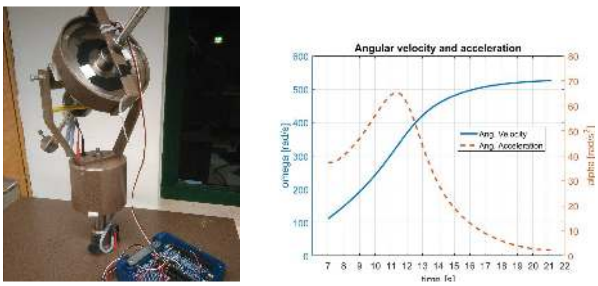
To control the relay that actuates the fan, a simple True/False condition was implemented. If the engine temperature is equal or higher than the "T trigger fan" (110^oC), the program sends out a signal of 5V that actuates the relay and turns the fan on. If not, the relay stays open and the fan off.

3.12. Data collection

LabView records the data in an *Excel* file where each column contains the recorded value at each millisecond. The collected data is then post processed in *MatLab* to generate the **torque and power diagrams**, and the **fuel consumption rate** to save the results already illustrated in the *LabView* interface.

4. Results

Before running the engine on the test bench, a previous test was made for the electric motor of a gyroscope. The engine rotation speed sensor was set on the flywheel (Figure 19 a)) and tests were made using the *LabView* program to confirm the required data is being collected. In Figure 19 b) it is possible to see the acquired data of angular speed (ω) and angular acceleration (α).



(a) Assembly of the rotation speed sensor.

(b) Collected data.

Figure 19: Rotation speed test of the gyroscope electric motor.

Knowing the inertia of the gyroscope flywheel is approximately $0.004Kg.m^2$, post processing was made in *MatLab* to assess the power and torque diagrams of the electric motor (Figure 20).

These results validate the instrumentation of the test bench. In this case, the gyroscope motor was characterised with a maximum power of 90 W , a maximum torque of 0.26 N.m and a maximum rotation speed of 5 000 RPM.

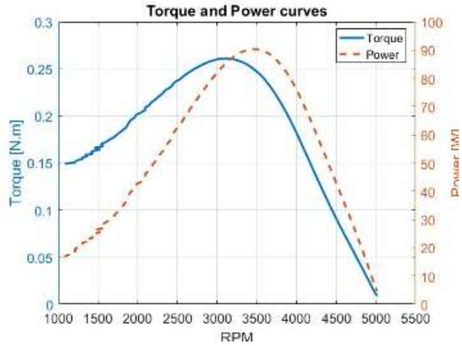
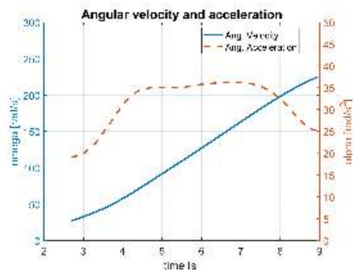


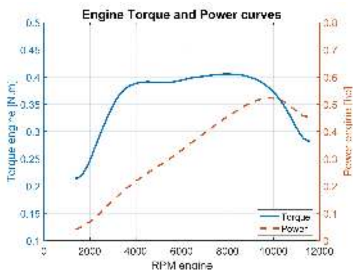
Figure 20: Torque and power of the electric motor driving the gyroscope.

4.1. Testing the 3.5cc Glow Plug Engine

A test was made for a 3.5cc glow plug engine, the *Novarossi Rex P5*, to validate the design. Besides demonstrating the results, these runs ensure the structural integrity of the test bench. The results of the power curves are shown in Figure 21.



(a) Flywheel angular velocity (rad/s) and acceleration (rad/s^2).



(b) Engine torque and power curves.

Figure 21: Results for the glow plug engine.

This run was accomplished by measuring the rotation speed at the flywheel. The engine rotation speed was obtained by multiplying those values by the gear ratio of $i = \frac{70}{13}$. Maximum speed, torque and power obtained for the for the *Novarossi Rex 3.5cc* engine are given in Table 5.

The run represents the typical behaviour of these engines where maximum power is achieved before maximum speed and never at the same time. Accordingly to the results, the engine reaches a maximum rotation of N = 11 310 RPM and it achieves a maximum power of 0.53 hp at 10 000 RPM.

Max RPM	Max Torque [N.m]	Max Power (hp)
11 310	0.41	0.53

Table 5: Performance of the 3.5cc glow plug engine on the test bench.

The expected value for maximum rotation speed was around 35 000 RPM. Possibly, some slipping occurred on the engine centrifugal clutch. Further tests have to be made measuring the rotation speed directly at the engine crankshaft. These engines are expected to achieve a maximum power between 1.5 hp and 2.5 hp.

5. Conclusions

The objective of the dissertation was to design, construct and instrument a test bench for micro combustion engines to characterise them in terms of torque, power and fuel consumption. It was built an engine inertia dynamometer because it was the best solution to obtain a dynamic test close to the real conditions as on the road. The program running the acceleration tests is fully automatic, allowing the test bench to have good repeatability which is essential to get realistic data. The cost breakdown shows a possible price of 2000 euros for this equipment. It is an acceptable value considering a model vehicle can cost up to 800 euros. One of the greatest challenges was the structural assembly of the test bench in IST, because the required materials were not always available, like special end mills to cut through stainless steel or specific turning tools. Also, some electronic components like the flow sensor were difficult to acquire. This had to be imported from United Kingdom because there were no economic solutions in Portugal.

The main achievements of the dissertation were:

- Answer the growing need for a standard testing of micro combustion engines.
- The test bench is able to test several types of micro engines.
- A user guide was developed explaining the testing procedure.
- Several sensors and actuators were set up to allow a fully automatic engine testing, ensuring a good repeatability of the acceleration runs.
- A *LabView* program was created with a graphical interface for the dynamometer operator to read the collected data live.
- A 3.5cc glow plug engine was tested to demonstrate the results and validate the whole design of the test bench.
- This prototype is a new testing equipment to use in IST laboratories for academic purposes.

5.1. Future Work

There are some things than can be further improved. Safety has a lot of issues to be resolved, for example, a heavier brake can be designed to stop the flywheel faster (never too fast) and a mechanical brake system should be implemented for backup in case of emergency like power failure.

Although the design was validated, the test bench was just assembled with a car engine. Other engines for boats or aircraft should be tested as well, and even larger engines closer to 20.0 cc, to confirm if the adjustable supports are suitable. Other properties can be assessed, such as fuel emissions, and further studies can be made to reduce the cost of the test bench.

The engine dynamometer built is a prototype that can always be improved.

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