DESIGN AND CONSTRUCTION OF A DINAMOMETER BENCH TO 5.5 CV ENGINES

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Abstract. *The main equipment of any engine test lab or specialized workshops is the dynamometer. The engines are subjected to tests or trials, so that it can be ensured its power and other constructing characteristics. A power test is the main test realized in any kind of new engine production after its assembly line in factories. In repair shops and / or remanufactured, it is recommended that the reconditioned engines are subjected to potency tests before being put back into service. Thus, in this study, it was analysed the parameters that are needed to build and conduct tests to determine the torque and power of engines ranging up to 5.5 cv. It was also analysed a building design, to improve a didactic learning in motor learning.*

Keywords: *Dynamometer, Otto cycle, Power, Torque*

1. INTRODUCTION

The main equipment of any engine test laboratory or specialized workshops is the dynamometer. The engines are subjected to test or trial so that you can ensure your power and other constructive characteristics (Claudio, 2006). potency test is the main test done in any kind of new engine coming out of its assembly line in factories. Repair shops and/or remanufactured, it is recommended that reconditioned engines are subjected to potency tests before they are placed back into service.

Presently, there are many versions, types and models of dynamometers available, but all are based on physical principles adopted in first kind of device, known as PRONY BRAKE, created in 1821 by French engineer Gaspard Clair François Marie Riche De Prony (1755-1839). This system can be observed in Figure 1a (Claudio, 2006).

The oldest device concept that is still used to this day of today to measure power of the engine comprises a flywheel surrounded by a connected strap to an arm one end of which rests on the platform a balance. The steering wheel, engine-driven, has its movement constrained by the pressure applied to the belt, which transmits the effort to the arm supported on the scale. From the readings of the scale, calculate the effort spent by the motor (Claudio, 2006).

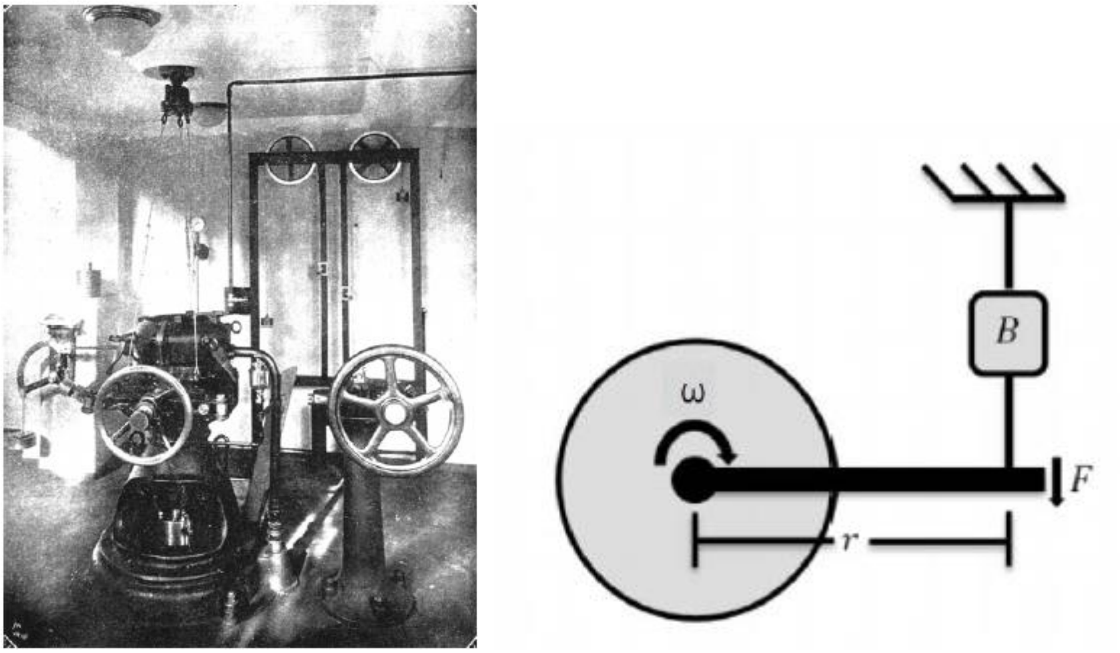
This type of measurement has a major disadvantage, since at the time measuring each change of speed or load causes the system to loose balance; therefore it is necessary to adjust the weights. This type of concept It can be seen in Figure 1b, which shows a composition which uses a scale for reading the force provided by the shaft.

Presently, there are two types of dynamometers, linear and rotary. Linear are used for working force measurements about something It has linear motion or no motion at all, meaning that there is only gravitational force acting on it. Since the rotating turn, are used for devices that have rotational movement (Turra, 2014).

With the evolution of technology and equipment, it was possible develop new models of dynamometers - models with readings electronic and more accurate than mechanical readings, for example, electric dynamometers and inertia dynamometers - however, the cost of manufacturing a dynamometer such is high, precisely because of its electrical parts involved.

The dynamometers are composed of several parts, which is commonly found in rotary dynamometers - which was the object of study of this work - a rotating part and a torque reading element. In some cases a lever is used for transmission spin move to the reading equipment (Dynamometer Prony) other uses electric current for carrying out reading element rotating (electric dynamometers) and there are also models that use hydraulic braking systems for this (hydraulic dynamometers) (Turra, 2014).

This paper presents the design and construction of a portable dynamometer for motors up to 5.5 hp (4.4 kW) using a brake disc as brake system. The project aims to solve problems arising from availability of space and cost for the construction of dynamometers professionals.



(a) (b)

Figure 1. (a) Prony brake. (b) Prony brake scheme

2. MATERIALS AND METHODS

The knowledge of the properties of materials used in the manufacture of products is important, as each material has a behavior that differentiates the manufacturing process to which it shall be submitted. To make a choice, the designer must take into account the properties of the material before and after processing, because in the meantime, major changes occur and may affect the project. Some properties such as mechanical strength, density, thermal conductivity and / or electrical, plasticity among others, are important, as the cost and availability (Van Vlack, 1970).

To design a dynamometer bench, the main challenge is the heat dissipation, which occurs through two modes: conduction from the friction at the disk and convection from the disk to the air.

2.1 Conduction

The ability of a material to conduct heat is called thermal conductivity (Incropera *et. al.*, 2008). The Table 1 exposes some material's thermal conductivities. The helical fins were composed of steel and the inside tube of copper.

Table 1. Materials and its thermal conductivities

Material	Thermal Condutivity (W/mK)
Steel	50.20
Copper	385.00

2.2 Convection

This heat exchange mechanism happens mainly between liquid and gas. Its function is to transfer heat within a fluid, through their own movements. Convection occurs as a result of differences between the densities of air. When the heat is conducted from the relatively hot surface for the overlying air, this air then becomes warmer than the surrounding air (Kreith, 1998).

The equivalent to the thermal conductivity is the convection coefficient. To evaluate the convection coefficient it is used a similarity approach. It is defined the dimensionless Nusselt number as the Eq. (1) (Incropera *et. al.*, 2008).

$$Nu \equiv \frac{hD}{k_f} \quad (1)$$

This equation does not define the convection coefficient, but the Nusselt number can be related to other dimensionless numbers using correlations as the Dittus-Boelter equation exposed in Eq. (2) (Incropera *et. al.*, 2008).

$$Nu = C \cdot Re^m \mathrm{Pr}^{1/3} \quad (2)$$

Re is the Reynolds number and Pr is the Prandtls number. C and m are determined using tables. Therefore, the convection coefficient can be determined using these equations.

The flow outside a heat exchanger is an example of convection phenomena.

2.3 Thermal deformation

The heat generated by friction in devices such as brakes and clutches induces thermal deformation, which associated with the deformations caused by tensions during sliding between insert and brake disc, can lead to the development of localized areas of contact areas and high temperature known as “hot spots.” “Hot spots” are areas with large temperature gradients in the contact surface. The existence of these areas is considered one of the most dangerous phenomena and can cause a device (brake system, for example) premature fracture, permanent distortion, and other damages, such as vibrations. It has been demonstrated that a thermo-mechanical stress associated with these “hot spots” can cause a cycle of compression and tensile stress variation with deformation plastic, which is directly related to the formation of cracks (Panier, 2004) (Choi and Lee, 2004).

Panier (2004) proposed a classification for the types of “hot spots” observed experimentally in disc brakes (exposed in Figure 2):

- clear from discrete asperities contacts. The temperature rises quickly but little, in small areas of the contact surface (1).
- hot bands gradients correspond to regions of small Contact appearing along a unique path (2).
- hot bands in the radial direction appear as small areas contact disc brake pad. They are seen on the disk as rings narrow high temperature in the direction of sliding. They can move along the radial direction during braking (3).
- macroscopic “hot spots” (MHS) are large temperature gradients evenly distributed on the disc surface. This one phenomenon drastically reduces the area of the contact surface with high local temperatures (4).
- “hot spots” with small temperature gradients distributed throughout the disk surface, not associated with a cooling homogeneous. This distribution appears at the end of braking associated with thermal diffusion (5).


(1) (2) (3) (4) (5)
Figure 2. Hot spots classification

The most dangerous gradients of temperatures for the structure of a Brake disc are those shown in Figure 10 (2), (3) and (4).

The type (2) has thermoelastic instability (TEI), which is based on variation of the contact friction due to interactions between expansions thermal, frictional heating, heat conduction out of the upper zone temperature and wear. According to the theory of the TEI “hot spots” appear only at a critical sliding velocity which depends on the thermal properties of the material.

The type (3) occurs by a reduction in the contact area of the tablet with disk, caused by thermal distortion of the components, wear of the insert and thermomechanical behavior of materials.

The type (4) shows large temperature gradients, commonly considered the main failure mechanisms on the disk. It is shown that MHS are found on both sides of the disc toward Slipping. The anti-symmetry of MHS and temperature levels therein found indicate a circumferential strain with plastic deformation and local changes in the crystalline structure of the metal (Panier, 2004).

Reducing the temperature difference between the surface and the interior of disc, is in the first instance, an effective method for the prevention of appearance of cracks on the surface. The thermal conductivity is the property essential to reduce the temperature gradient. The increase in resistance to stresses and fatigue of the material would also be a possible alternative (Jimbo *et. al.*, 1990) (Mackin, 2002).

2.4 Power and torque

This power definition is related to the ability of an engine or other machine to perform mechanical work or generate energy in a given time interval. (Gray, Constanzo and Plesha, 2014)

According Bosch (2005), the power refers to the amount of work performed per unit time. Therefore, the power equivalent to a change in speed occurs in a power system or a time taken to perform work.

According to Beer and Russell (1991), the power is the criterion in selecting a motor or engine since, although a job can be effected by motors of various sizes, the time for a larger engine would run a job It would be short compared a smaller engine would need a longer time to reach the same result.

Everything that generates or tends to cause rotation or twist, and the intensity is measured by the product of force and the perpendicular distance the force line of action of the rotation axis.

The Torque always occurs in a plan and applied around an axis. The greater the distance of the force application point to axis of rotation is greater the torque. If the force is acting on the shaft alignment, the value of the torque produced will be zero. (Training Division Publication, 2004).

The equation used to relate the power and the torque was the Equation 1, as Taylor (1988) stated.

$$P = B.r.\omega.\frac{\pi}{2250} \quad (1)$$

3. RESULTS AND DISCUSSION

Since the desired dynamometer capacity was 5.5hp, or 4.4kW, it was necessary to dissipate this power in the brake system. Considering a motorcycle disk, the first option for a brake system, the disk area was measured as 0.0233m^2. The measured perimeter was 0.5m and the hydraulic diameter was 0,18m. That implied in a flow of 188kW/m^2 transferred from de shaft to the brake system. Considering an estimated 400K temperature difference, the calculated necessary convection coefficient would be of 470W/m^{2}K.

This flow should be dissipated by convection. It was adopted a forced convection, since the brake disc would be rotating. Using an axial velocity of 3.75m/s (1500rpm for a 0.15m disc – a worst case scenario), the calculated Reynolds was $1.03 \cdot 10^5$ (turbulent) and the adopted Prandtl was 0,8. It was adopted C=0.228 and m=0.731 (vertical plate with constant thermal flow, as Incropera *et al.* (2008) suggested). The calculated Nusselt number was 980.73. The adopted conduction coefficient was 43 W/mK.

With these considerations, the necessary convection coefficient would be around $2.27 \cdot 10^5$ W/m^{2}K. That coefficient would exceed the necessary, however, it is possible that the adopted convection correlations are not sufficient to model the brake disc, it was decide to use an available disc brake to verify whether the "hot spots" phenomena would occur.

An experiment was set up. A 5.5hp motor was attached to a baja vehicle and the brake was activated and the disc brake was recorded with a thermal camera, as exposed in Figure 3.

Figure 3. Experiment to verify the occurrence of hot spots in a brake disc.

After 7 minutes of experiment, the image exposed in Figure 4 was acquired. It was verified that the brake disc tested had "hot spots" of type (5), one of the least hazardous, which means small temperature gradients distributed over the entire disk surface. After reaching this conclusion, the focus of problem became its braking efficiency and not its structure. The maximum measured temperature at the instant the disc and brake pads were in contact was 457.6ºC. As, according to the manufacturer, the brake loses performance after 300°C, therefore, it was decided to adopt a different brake disc. The chosen brake disc was a ventilated disc, commonly used in cars, as exposed in Figure 5.

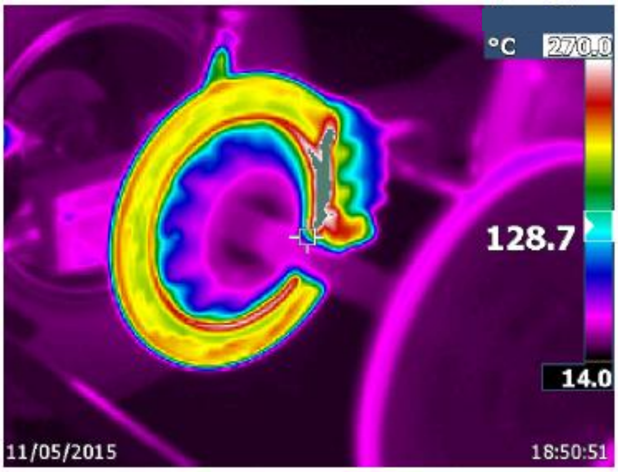
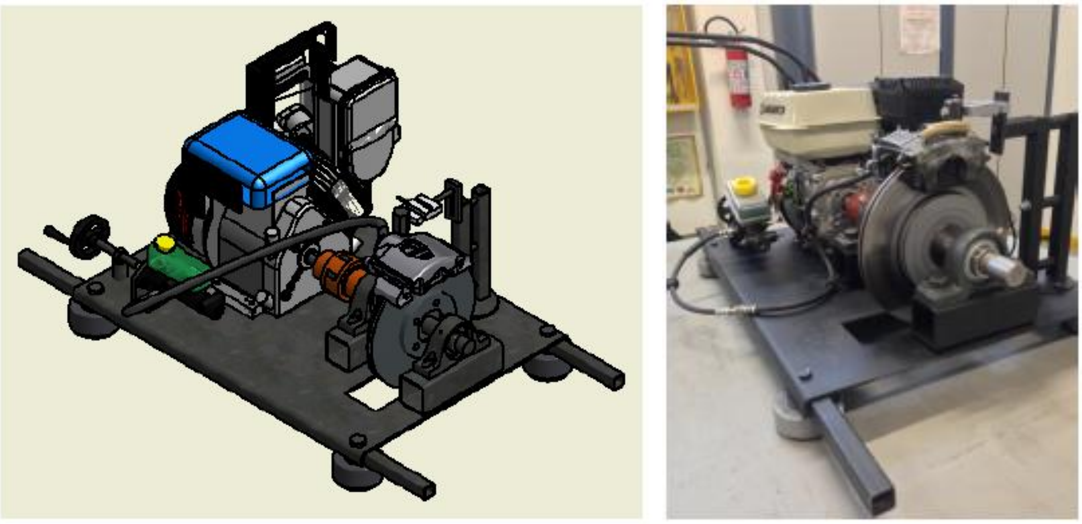


Figure 4. Temperature distribution for a brake system after 7 minutes of experiment.

Figure 5. Adopted brake disc

This brake disc had an area of 0,5m^2 – counting four exposed faces, since the disc has internal ventilations. This disc worked at 240°C temperature and did not lost performance when braking nor presented the “hot spots”. The engine selected for validation and testing was a B4T-5.5H of White engine manufacturer, and is available from Universidade Positivo. This engine has a rated power of 5.0 hp (3.68 kW) at 3500 rpm.

The proposed dynamometer bench was designed and implemented as exposed in Figure 6.



(a) (b)

Figure 6. Proposed dynamometer bench.

A load cell was used to measure the power. The load cell adopted in this project was the TS 20 with a capacity of 20 kg and with an accuracy of 4 grams and torque measurement capacity of 10 Nm. The load cell was calibrated with a 5kg mass and, after that, an experiment with the dynamometer bench was set up. The measurement results are exposed in Table 1. The tension was measured in the load cell terminals and is listed in the same table.

Table 1. Power and torque measurements

rpm	Tension (mV)	Torque (kgfm)	Power (hp)
2000	4.8	1.02	2.76
2100	4.85	1.03	2.96
2200	4.9	1.04	3.15
2300	4.95	1.05	3.35
2400	4.99	1.06	3.55
2500	5	1.06	3.65
2600	4.99	1.06	3.75
2700	4.98	1.06	3.94
2800	4.96	1.05	4.04
2900	4.93	1.05	4.14
3000	4.9	1.04	4.34
3100	4.86	1.03	4.44
3200	4.82	1.02	4.54
3300	4.78	1.02	4.63
3400	4.73	1.01	4.73
3500	4.67	0.99	4.73
3600	4.59	0.98	4.83

This data can be observed in comparison with the manufacturer manual (BRANCO, 2000) in Figure 7. Notice that the measured power and torque are in conformity with the manufacturer data.

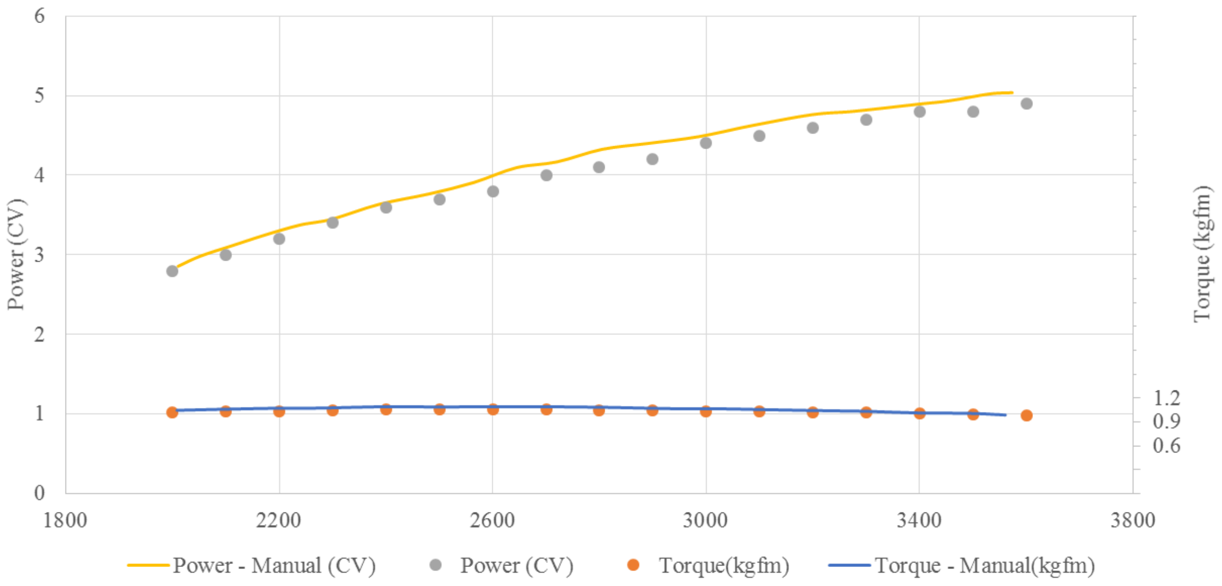


Figure 7. Dynamometer bench measurements in comparison with motor manual data

4. CONCLUSION

In this research it was analysed the parameters that are needed to build and conduct tests to determine the torque and power of engines ranging up to 5.5cv. A heat transfer study was performed and an adequate brake disc was selected to dissipate de engine power in the brake system through conduction and convection. It was also analysed different designs, allowing to choose the best design of more appropriate setting to the initial project with the goal of better learning in motor learning. The designed dynamometer bench is suitable to didactic purposes and to analyse commercial engines up to 5.5cv.

5. ACKNOWLEDGEMENTS

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