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Structural Analysis Evaluationfor Rollers of Chassis DynoUsing Finite Element Software

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ABSTRACT
This paper discusses the design considerations for a chassis dynamometer, focused on the rollers system used for measuring the power and torque of a vehicle in a university-level workshop. The design must prioritize factors such as safety, ease of use, adaptability to diverse types of vehicles, and the inclusion of safety systems to protect students and the vehicle. The work suggests adaptative features for the design, such as an adjustable fastening system and a compact and portable design, to adapt the structure into a university laboratory. The study also describes common methods for measuring engine power and torque using a chassis dynamometer, and the accurate and reliable measurement systems used for this purpose. This concludes with a discussion of the manufacturing methods used for the chassis dynamometer and a static analysis of the maximum effort capacity in each roller using Von Mises and Goodman fatigue theories. Developing a chassis dynamometer, in accordance the current patents available for workshop practices in a university is essential for providing students with comprehensive training.

Keywords-Chassis dynamometer, conceptual design, finite element software, power and torque measurement, structural analysis.

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I. INTRODUCTION

This Accurately measuring a vehicle's power and torque is crucial in the fields of mechanical engineering and automotive technology. Chassis dynamometers are widely used devices for measuring the power and torque generated by an engine. However, designing and constructing a chassis dynamometer requires considering several factors, including the static analysis of the rollers.

Static analysis of the rollers is crucial to evaluate the forces generated in a chassis dynamometer since these rollers support the vehicle's weight and transmit the engine's force to the dynamometer. A detailed analysis of the stresses on the rollers can ensure the safety and reliability of the device while improving its design and performance.

The study aims to provide a detailed review of the static analysis of rollers on a chassis dynamometer for the evaluation of generated forces. The article will discuss the fundamental principles of static analysis and present simulations using the finite element method to evaluate stresses in the rollers. Additionally, the factors to consider in designing and constructing chassis dynamometer rollers will be analyzed.

Thispaper provides valuable information for mechanical engineers and automotive technicians involved in the design, construction, and maintenance of chassis dynamometers.

II. DESIGN CONSIDERATIONS

A chassis dynamometer is an instrument used to measure the tractive force or power of a moving vehicle. The chassis dynamometer consists of two rotating rollers that are placed under the wheels of the vehicle and connected to a measuring system. When the vehicle moves on the rollers, the force generated by the wheels is transferred to the rollers and the tractive force of the vehicle is measured.

The design and analysis of stresses in the rollers of a dynamometer is a critical aspect to consider when developing a chassis dynamometer for workshop practices at a university. The dynamometer consists of two rotating rollers that measure the tractive force or power of a moving vehicle. The focus of the design should be on the evaluation of the rollers' axis and the stresses generated in them, considering the steel types 1018 and 4140. Additionally, it should have an accurate and easy-to-read measurement system to help students understand the measurement results [1].

When designing the dynamometer, existing patents in the field of force and tension measurement were taken into consideration. One of the reference patents was US20160109520A1 titled "Force measuring device and method,"[2] which describes a force measuring device that utilizes a spring structure to measure the force applied to an object. Another reference patent was US20100236595A1 titled "Tension sensor,"[3] which describes a tension sensor that uses an elastic material and a transducer to measure the tension in a rope or cable.

These patents were utilized to develop a force and tension measuring reference device that meets the specific requirements of the project. The spring structure described in the first patent and the use of an elastic material in the second patent were considered to design a device that is precise and dependable in measuring force and tension.

A design proposal could include features such as an adjustable fastening system and a compact and portable design. Different measurement methods can be used to measure the power and torque of an engine using a chassis dynamometer, such as the electric current method.

The design of the rollers is a critical aspect of the chassis dynamometer. The rollers must be designed to withstand the forces generated by the wheels of the vehicle during testing. Therefore, it is essential to evaluate the stress and deformation of the rollers under different loading conditions.

In addition, the design must incorporate safety features to ensure that the vehicle can be safely stopped in the event of a transmission system failure. An overload protection system should also be included to prevent any potential damage to both the dynamometer and the vehicle being tested [4].

One approach to designing the rollers is to use finite element analysis (FEA). FEA is a numerical technique used to predict how a component or structure behaves under different loading conditions[5]. In this case, FEA can be used to analyze the stress and deformation of the rollers under the force generated by the wheels of the vehicle.

The FEA model should consider the material properties of the rollers, which can vary depending on the specific type of steel used. For example, steel 1018 and 4140 are commonly used in the manufacture of rollers due to their excellent mechanical properties. Steel 1018 is a low carbon steel that is easy to weld and has good machinability, while steel 4140 is a high-strength steel that is widely used in applications that require high strength and toughness.

The FEA analysis can help to optimize the design of the rollers, examining factors such as the material properties, roller diameter, and roller length. By analyzing the stress and deformation of the rollers, the design can be modified to ensure that the rollers avoid plastic deformation in the roller during the test [6]. This will ensure the accuracy of the measurement results and the safety of the students and the vehicle.

III. MEASUREMENT METHODS AND STRUCTURAL ANALYSIS

In a chassis dynamometer, the rollers are subjected to different types of stresses, such as tensile force, torque, shift, deformation, and fatigue [7]. It is important to consider these efforts to properly design the rollers and ensure their strength and durability. Static analysis of the rollers makes it possible to evaluate these stresses and determine the safe operating conditions for the chassis dynamometer.

The maximum load capacity of the proposed chassis dynamometer is a critical design consideration. The dynamometer must be able to handle the maximum weight of the heaviest vehicle it will be used to test. The dynamometer must be able to handle the weight of the heaviest vehicle it will be used to test. For this reason, the proposed design will have a minimum load capacity of 10791 N.

By applying the equation of equilibrium to the shaft, we can determine the moment at the center of the tire. Also, assuming a maximum torque of 3876.45 N-m based on data from the vehicle's computer.

According to this equation, the moment between two points on the beam is equal to the area under the shear curve between those same two points. In Fig. 1, it shows the bending moment under the shear curve between points x = 80.00 mm and x = 650.00 mm is 5,013.66 N-m.

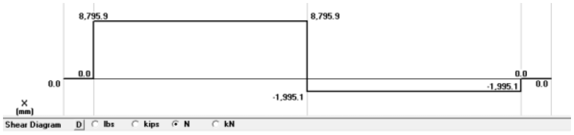

Figure 1. Bending moment on shaft

The von Mises criterion is a widely used failure theory in material science that predicts a material's resistance to plastic deformation equation 1. The criterion postulates that a material fails when the equivalent von Mises stress at a point exceeds a critical value. The von Mises equation (1) is employed to calculate the equivalent stress in a material under complex loads. It is expressed as follows:

$$\frac{1}{n} = \frac{\sigma'_a}{S_e} + \frac{\sigma'_m}{S_{ut}} \quad \text{Eq. (1)}$$

Where:

n = fatigue safety factor

Sut: Ultimate tensile strength of the material

Se= Fatigue strength limit of the material.

σa': Alternating equivalent effort.

σm': Average equivalent effort.

. The dynamometer rollers experience complex loads, and determining the equivalent stress at each point on the rollers was crucial. By utilizing the von Mises equation, the equivalent stress at critical points of the dynamometer rollers was determined and ensured that these stresses did not exceed the safe limits for the material utilized in constructing the rollers.

By developing structural calculations, the maximum permissible load that can act on the dynamometer rollers was theoretically evaluated. The components, such as the torque transmission shaft and the loads supported on the rollers, were analyzed using computer-aided analysis software to verify the load absorption capacity.

The reference data used in this study was based on the dimensions of a 4634 mm X 1834 mm vehicle. The rollers in question have a diameter of 406 mm, which implies a radius of 203 mm. To calculate the load per unit length equation (2), the total load is divided by the roller circumference using the formula:

$$Load\ per\ unit\ length\ = \frac{Total\ load}{Roller\ circumference} \quad \text{Eq. (2)}$$

By substituting the values, we get:

Load per unit length = 10791 N / (pi * 406 mm)

Load per unit length = 8.41 N/mm

The moment of inertia of a cylinder is a crucial property for the design and analysis of rotating systems. For the analysis, a pair of rollers with a diameter of 406 mm and a length of 1000 mm was considered. To calculate the moment of inertia, the density of the roller material must be known. Taking in count the roller is made of steel, which has a density of 7800 kg /m^3, the volume of the cylinder can be calculated using the equation (3):

$$V = \pi * r^2 * h$$ Eq. (3)

Where "r" is the radius and "h" is the length of the cylinder. Replacing the values, we get:

V = π * (0.203 m)^2 * 1 m = 0.1302 m^3

The mass of the cylinder can be calculated using density and volume in equation (4):

$$m = \rho * V$$ Eq. (4)

Therefore:

m = 7800 kg/m^3 * 0.1302 m^3 = 1013.16 kg

The moment of inertia can be calculated using the equation (5):

$$I = \left(\frac{1}{2}\right) * m * r^2$$ Eq. (5)

Substituting the values obtained, we get:

I = (1/2) * 1013.16 kg * (0.203 m)^2 = 20.57 kg*m^2

The maximum cutting stress on the rollers is an essential factor in determining the strength and durability of the equipment. To calculate this stress, the maximum shear force can be divided by the cross-sectional area of the rollers. Assuming the rollers have a diameter of 406 mm and a thickness of 17 mm, the radius can be calculated, and then the formula of the area of a circle can be used to calculate the cross-sectional area of each roller. The radius of the rollers is half the diameter, which gives:

r = 406 mm / 2 = 203 mm

The cross-sectional area of a roller can be calculated using the equation (6) for the area of a circle:

$$A = \pi * r^2$$ Eq. (6)

By substituting the values, we get:

A = π * (203 mm)^2 = 129,602 mm^2

If the maximum shear force (weight per roller) is 5000 N, taken from initial data in Table1 the maximum cutting stress on the rollers can be calculated by dividing the shear force by the cross-sectional area of the rollers in equation (7):

$$\tau_{max} = \frac{F_{shear}}{A}$$ Eq. (7)

By substituting the values, we get:

τmax = 5000 N / 129,602 mm^2 = 0.0385 MPa

Table 1. Input data from vehicle data

Input data	
Power engine HP	200
Torque	3500 N*m
Max. Speed	80 Km/hr
Mass (Total)	2200 Kg
Weight (per roller)	5000 N
Car dimensions	2.1 mts X 4.7 mts

Compared to the strength limit of steel, the maximum cutting stress value on the rollers of 0.0385 MPa is low, indicating that the roller material can withstand the load and forces applied in the use of the dynamometer. Therefore, the maximum cutting stress on the dynamometer rollers is approximately 0.0385 MPa, which is well within the limits of the steel.

With these input values, the total effort capacity in each roller was analyzed to prevent ensuring an adequate selection of materials for its future construction.

Through the evaluation of the components by means of computer-aided analysis software, it is possible to obtain that the maximum deformation generated by the weight of the vehicle reaches 3.75 mm, this considering the contact point of the wheels of the vehicle on the rollers of the structure [8].

Continuing with the static analysis, to obtain the maximum effort of the roller shaft, it is necessary to evaluate certain characteristics of the material, as well as the amount of load and the type of force form applied, which in this case used the torque defined by the technical sheet of the vehicle. Considering Goodman's failure criteria Fig.1[9], it involves determining the possible scenarios that induce a failure caused by fatigue in some mechanical system.

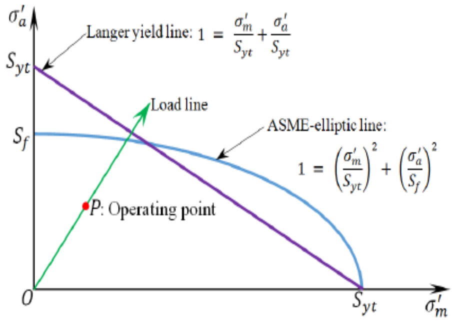

Figure 1. Goodman's failure criteria

This criterion supports the design of mechanical elements. Those are attributed within the analysis that, for the resolution of this criterion, the maximum effort generated through the calculations must be compared with the creep stress of the material.

IV. ROLLER'S SIMULATION

The implementation of an innovative chassis dynamometer for workshop practices in a university is essential to provide students with a complete training.

Regarding manufacturing methods, chassis dynamometers are usually built using welding and assembly techniques. The steel frame is cut and welded to create the basic structure of the dynamometer, and then electronic components are added [8].

Dynamometer rollers are made of strong and durable materials, such as hardened steel. The rollers are machined to ensure a uniform contact surface with the wheels of the vehicle and are mounted on the dynamometer frame using high-quality bearings [10].

Considering that the failure of the material will occur if the yield stress is less than the maximum effort generated in the study.

Considering the requirements by a dynamometer roller, to avoid any failure in its mechanical elements, the work was focused on the design of the components and their evaluation through their natural frequency, according to the suggested torsion force of use[11].

4.1 Boundary conditions

The development of the components arises from the general idea of a system that supports a specific load of force, considering the weight of the vehicle indicated in the structure Fig.2. reviewed the critical components of the dynamometer structure, such as the axle, the rollers and the wing of the mechanism, as critical points of analysis.

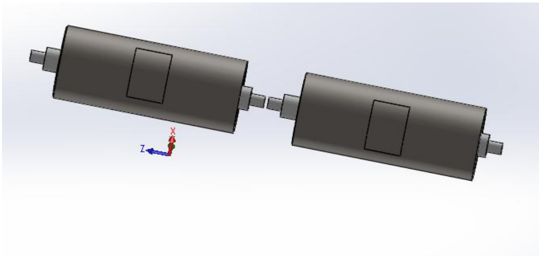

Figure 2. Dynamometerroller's structure prototype

For the design of the components, the most used materials of this type of devices are evaluated Table2, and this led to the conception of the design through virtual prototypes.

Table 2. Dynamometer structure´s materials

Material	Qty.
PTR tube 51X51 6mm	3
PTR tube 51X51 152mm	8
Carbon steel sheet	2
Rowlock P2B 208-SRB-CLH (SKF)	2
SY55TF ball bearing support	2
Roller shaft L=330cm D=55.6mm	1
Telma turbine AC 50-55 Retarder	1
Pulley Dint= 5.56cm Dext=13	2
Flange Dint=5.56cm	1

Flywheel	1
Angular tubular 35X35 8cm	3
Pulley belt type 5V	1
Steel tube with406.4 mm diameter	2
Electric brake for dynamometer	1

In Fig. 3, the rollers shownwith the shaft, will perform the absorption of failure in device. This element will be where the weight and torque from the reference vehicle will be applied to obtain the deformation values, as well as the maximum axle forces, and to prevent any failure.

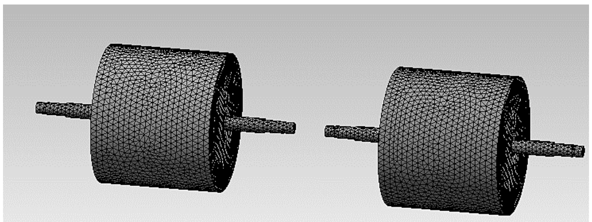

Figure 3. Dynamometer rollers and shift assembly from structure prototype

The study compares two materials for the shaft (shown in Table3), AISI 1018 steel with a tensile strength (Ultimate) of (440 MPa)[12], and AISI 4140 steel with a tensile strength (Ultimate) of (655 MPa)[13].

Table 3. Materials conditions for shaft and rollers.

Component	Material	Yield Strength	Tensile Strength
Roller	Steel A-36	250 MPa (36 ksi)	400 MPa (58 ksi)
Shaft	Steel 4140	415 MPa (60 ksi)	655 MPa (95 ksi)
	Steel 1018	370 MPa (54 ksi)	440 MPa (64 ksi)

And for the rollers, an A36 steel with a tensile strength of 400,shown at Table3 [14].

To perform the power evaluation of the device, the following Table4is conducted, with which the maximum amount of torque to be used in the system elements is analyzed.

Table 4. Output torque per gear transmission

Car speed by each gear transmission	Torque (Output) N*m
1ª: 6.9 km/h a 1.000 rpm	5245
2ª: 9.8 km/h a 1.000 rpm	3693
3ª: 14.9 km/h a 1.000 rpm	2429
4ª: 21.3 km/h a 1.000 rpm	1699
5ª: 29.3 km/h a 1.000 rpm	1235
6ª: 37.4 km/h a 1.000 rpm	968
7ª: 47.9 km/h a 1.000 rpm	756
8ª: 60.7 km/h a 1.000 rpm	596
R: 82 km/h a 1.000 rpm	441

4.2 Study results on dynamometer rollers

To conduct the study using finite element software, a system was established to evaluate the power generated by a vehicle of maximum power of approximately 200 HP.

The purpose of the study is to find the maximum tensile that can be supported by the dynamometer, to avoid fatigue failures in the structure of the device.

Considering the characteristics of the vehicles that will generate the power load that will vary during the study, depending on the type of car, limiting as maximum power load 45HP.

Through the analysis of finite elements in the evaluation of the static load a value of 240.02 MPa was obtained using Von Mises theory, with which it is observed that rollers are capable of supporting stress of up to 6.249 Pa per each one shown in Table5.

The results of a static study performed using finite element analysis for different materials used in a roller and shaft. The results are presented in terms of total deformation and equivalent stress for AISI 1018, AISI 4140, and the combination of A36 roller with 1018 and 4140 shafts.

The data is divided into two sections: one for the deformation and equivalent stress results for

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the roller shaft, and another for the deformation and equivalent stress results for the roller cylinder.

In the first section, the minimum, maximum, and average values of total deformation and equivalent stress are shown in Fig. 5 for AISI 1018 and AISI 4140 materials. In the second section, the minimum, maximum, and average values of total deformation and equivalent stress are presented for the combination of A36 roller with 1018 and 4140 shafts shown in Fig. 7.

Table 5. Static study results by finite element analysis

		Results			
		Total Deformation		Equivalent Stress	
Roller Shaft	Mat.	AISI 1018	AISI 4140	AISI 1018	AISI 4140
	Min.	2.9954 x10^-9 m	2.8897 x10^-9 m	139.4 MPa	123.6 MPa
	Max.	3.8432 x10^-4 m	3.7077 x10^-4 m	242.02 MPa	243.08 MPa
	Avg.	1.9272 x10^-4 m	1.8592 x10^-4 m	22.259 MPa	14.959 MPa
		Total Deformation		Equivalent Stress	
Roller cylinder	Mat.	A36			
	Min.	1.6903x10^-3 m		0.006249 MPa	
	Max.	3.7521x10^-3 m		240.02 MPa	
	Avg.	2.7163x10^-3 m		22.259 MPa	

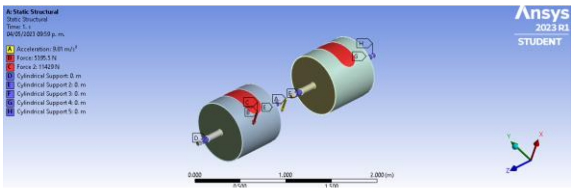

Figure 5. Dynamometer rollers and shaft assembly in static analysis, with green areas marked as the location of the main forces.

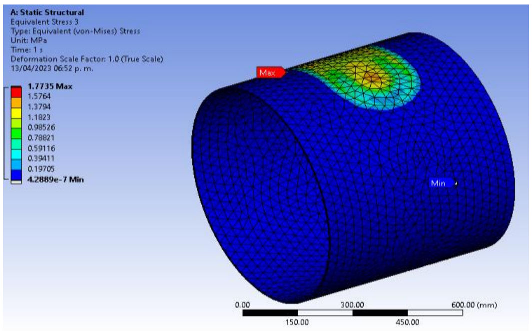

Figure 6. Dynamometer rollers and location of the main forces

The calculation of the safety factor under conditions of fluctuating load, according to equation 1, requires the evaluation of the values of AISI 1018 material, with an ultimate tensile strength (Sut) of 440 MPa, and AISI 4140, with a Sut of 655 MPa. To do this, the alternating equivalent stress must be considered, calculated using equation (8), as well as the mean equivalent stress, according to equation (9). In this way, the following results are obtained:

$$\sigma a' = \frac{\sigma max - \sigma min}{2} \qquad \text{Eq (8)}$$

$$\sigma m' = \frac{\sigma max + \sigma min}{2} \qquad \text{Eq (9)}$$

Replacing, we obtain:

For the AISI 1018 shaft:

σa'=(243.08 MPa)-(123.65 MPa) / 2 = 59.715 MPa

σm'=0 MPa. The value is zero due to the presence of an alternating load during the study.

For AISI 4140 shaft:

σa'=(243.08 MPa)-(123.65 MPa) / 2 = 59.715 MPa

σm'=0 Mpa. The value is zero due to the presence of an alternating load during the study.

The equation (1) is essential to define the fatigue strength limit of the material, which can be obtained by applying equation (10).

Se=Se'(Cm)(Cst)(CR)(Cs) Eq (10)

The coefficient e data will be obtained according to the material type, for both 1018 and 4140 steels. The following data will be utilized for the analysis:

Se' = modified fatigue strength for steels = 0.5*Sut

Cm = carbon steel material coefficient = 1

Cst = axial load coefficient = 1

CR = reliability coefficient = 1

Cs = size coefficient = 0.859-0.000837D

The diameter of the shaft is known to be 55.6mm according to Table 2.

Therefore, the coefficient is obtained:

CR=0.859-0.000837DEq. (11)

CR=0.859-0.000837*55.6mm = 0.85816

Substituting the coefficients into equation (10) yields the following result:

For steel 1018

Se=(440MPa)(0.5)(1)(1)(0.85816)(1)= 188.795MPa

For steel 4140

Se=(655MPa)(0.5)(1)(1)(0.85816)(1)=281.0474MPa

Therefore, by applying equation (1), the following fatigue safety factor values are obtained for each material:

For steel 1018:

n=1/((51.305MPa/188.795MPa)+(0MPa/440MPa))

n=3.6798

For steel 4140:

n=1/((σa'/Se)+(σm'/Sut))

n=1/((59.715MPa/281.0474MPa)+(0Mpa/655MPa))

n=4.7064

These safety factors, it is confirmed that the shafts will not suffer any type of damage, since they are greater than the minimum value is 1.

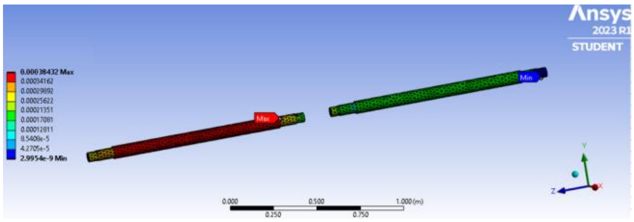

Figure 7. Dynamometer shaft in static analysis, with red areas marked as the location of the main forces.

V. CONCLUSION

The study has demonstrated that the selection of the material used for the shaft and roller-shaft combination plays a crucial role in determining the equivalent stress and total deformation of the system. The results reveal that the use of AISI 4140 material results in the highest equivalent stress, indicating that it may not be the best choice for applications where high stress is a concern.

Developing a conceptual design of a chassis dynamometer for workshop practice is essential to provide students with a direct learning experience that enables them to acquire in-depth knowledge of vehicle performance and diagnostic and performance improvement techniques. The use of this equipment in universities will help students gain practical skills and experience, allowing them to prepare for real-world scenarios.

The implementation of an innovative chassis dynamometer in a university offers many benefits. First, it significantly improves the efficiency of the equipment, allowing for accurate and reliable results. Second, it enhances safety by minimizing the risk of accidents, thereby reducing the possibility of injuries.

The findings can be applied in various areas, including the development of new materials for roller-shaft combinations, optimization of the design of the chassis dynamometer, and improving the overall performance of the system. Furthermore, the results can help manufacturers in selecting the appropriate materials for their products to ensure that they meet the desired performance requirements.

The fatigue safety factor calculation has been carried out for the steel shafts of AISI 1018 and 4140, using the corresponding equations and appropriate coefficients. The obtained results demonstrate that both shafts comply with the established safety requirements, as the obtained safety factors are greater than the minimum value required. This indicates that the shafts are adequately designed and are expected to operate without problems throughout their service life.

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