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Design and Finite Element Analysis of a Tire Shredding Machine for the Materials Laboratory

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Abstract

The article presents the design calculations of the elements of a tire shredding machine, using mathematical formulas, analyses the influence on the parameterization of mechanical stresses when using rubber that will be cut in its natural state compared to the same material with a natural bio-coating through triglyceride fatty acids present in palm oil, which acts on the surface without altering the chemical composition. The materials used for the machine are A36 structural steel for the blades, bedplates, spacers, and structure, AISI 4340 steel being the most suitable for use in a shaft. Free body diagrams were also made for the designed shaft to observe the bending and torsional moments acting on it. Using CAD-CAE software, a finite element analysis was carried out in which we can find the Von Mises stresses and the safety factor of each designed part. The findings include increasing factors, efficiency as well as optimizing the time needed to obtain the tire shredding with the optimal particle size and reducing final costs in the production of the raw material. It is proposed to replace the glycerine used as a rubber degrader with a palm oil-based bio coating with a thickness of ± 50 nm containing triglycerides and natural fatty acids.

Keywords

Nano-coating, Fatty acids, Mechanical analysis, Tire shredding machine, Mechanical analysis, Optimization

Introduction

Used rubber tires from vehicles are materials that should be reused. It is estimated that around 2,400,000 tires are discarded each year, which take around 500 years to disintegrate. These tires are often not disposed of properly and accumulate in landfills, derelict land, and roads.

This project aims to determine which biodegradable substance will help to modify the surface level of rubber and use it in the pretreatment processes of the raw material recyclable tire rubber, within an alternative design of a shredder [1]. Considerations will be considered in the respective design calculations, starting with gear calculations [2], applying the corresponding stresses on the gear tooth and carrying out a simulation in finite element software [3]. For the design of the shaft, the loads, shear force and moments acting on it will be considered, represented in a free body diagram [4]. Subsequently, the simulation of the shaft will be carried out in CAD-CAE software [5], in which the Von Mises stress and the safety factor of the shaft will be observed [6]. For the design of the blades, the geometric shape [7] will be considered, in addition to selecting the material, which is an A36 structural steel according to the strength and deformation of the blade [8]. For the results of the tire shredding machine, we will tabulate them based on the granulometry of the rubber [9], with respect to the number of passes of the end-of-life tire.

Experimentation

Characterization of the nanotechnological bio-coating

The analysis of the molecular morphology was performed by scanning electron microscopy in incubation periods of 3 to 21 days to dehydrate the cells, taking as reference a sample of glutaraldehyde (Sigma Aldich) at 2.5% diluted in phosphate buffered saline for 2 h at 4 °C using Philips XL 30 and Jeol JSM6700 microscopes.

Three hypothesis of biological coatings based on palm, soybean and coconut oil were proposed, reaching greater efficiency in the superficial degradation of the tire rubber, the palm oil, due to the higher concentration of triglycerides in its enzymes, In addition to this, it has natural fatty acids that adhere to the rubber forming a high density surface layer in the order of ± 50 nm in its upper limit and ± 40 nm in its lower limit, providing a factor of 10% for use in different types of tires with multiple characteristics in their physical structure.

Design development

The main materials for the execution of the project are classified as: axle, blades, separators, structure, bedplates, gears, and standardized elements.

We will start with the pre-design calculations that it is necessary to consider several factors such as the maximum rim of the tires to be shredded, in this case 235/70 R16, the diameter of the wire from which the rim is made and the resistance of the rim's cable.

An average diameter value of 0.259 mm is obtained, then we obtain the area of the steel wire rope, obtaining a value of 0.052685 mm^2. The area is multiplied by the stress of the wire (1388.54 MPa) [10] to obtain the force, in turn we multiply by 64, which is the number of wires in 2 cm of the tread and as we have a shear cut of 2 blades at the same time, we multiply the value by 2 obtaining a result of 9363.8656 N.

To calculate the required motor torque, we multiply the force obtained by the distance from the shaft to the blade tooth (0.082 m), giving a value of 767.8369 Nm (Figure 1).

Gear design

Gear design is mentioned in table 1.

Axle design

The following data must be considered for the design of the axle (Figure 2): Material: AISI 4340, Sy = 800 Mpa, Sut = 1000 Mpa, Weight of all blades = 17.3481 N * 8 = 138.7848 N, Weight of all separators = 5.6048 * 8 N = 44.8384 N, Axle weight = 142.34 N, Gear weight = 24.4652 N, Maximum shear force = 9363.8656 N, and Torque = 383.9185 Nm.

The forces and moments applied on the shaft are P1 = 325.963 N, which represents the sum of the weight of the shaft, blade, and spreader, M1 = 1405.4254 Nm is the input torque, P2 = 31.1376 N corresponding to the weight of the catarina, M2 = 767. 8369 Nm is the torque needed to cut the wire, P3 = 24.4652 N represents the weight of the gear, M3 = 383.9185 Nm refers to the output torque of the blade at the end of the cases.

Figure 1: Measuring test on 2 cm of tire.

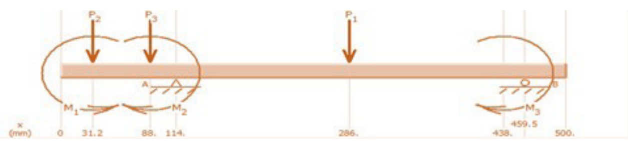

Figure 2: Free body diagram for the shaft.

Table 1: Gear design.

Circular passage (in)	Base step (in)	Diametral pitch (teeth/in)	Distance between centres (in)	Reason for contact	Tangential component (lb)
$P_c = \frac{\pi d}{N}$	$P_b = P_c \cos\phi$	$P_d = \frac{N}{d}$	$C = r_p + r_g$	$m_p = \frac{Z}{P_b}$	$W_t = \frac{T_p}{r_p}$
0.6374	0.5989	4.9283	5.2756	1.6013	2576.3626
Radial component (lb)	**Resultant force (lb)**	**Bending stress (kpsi)**	**Surface stresses (kpsi)**	**Uncorrected bending fatigue strength (kpsi)**	**Factor of safety against bending failure**
$W_r = W_t \tan\phi$	$W = \frac{W_t}{\cos\phi}$	$\sigma_b = \frac{W_t P_d}{FJ}\frac{K_a K_m}{K_v} K_s K_B K$	$\sigma_c = C_p\sqrt{\frac{W_t}{FId}\frac{C_a C_m}{C_v} C_s C_f}$	$S_{fb} = \frac{K_L}{K_T K_R} S_{fb}$	$N_{bengrane} = \frac{S_{fb}}{\sigma_b}$
937.7193	2741.7078	21.2638	224.2905	54.0938	2.5439

We shall section the shaft into 4 sections as shown by the diameter data and corresponding length in figure 3.

It is analyzed at point D, where the bending moment is greatest.

$$T_{m_1} = 1405425.4\,Nmm;\ M_{a_1} = 2578.1933\,Nmm$$

Because the shaft is rotating, the constant bending moment will cause a fully reversible bending stress.

Creep safety factor

Von Mises maximum stress:

$$\sigma_{\max} = \left[\left(\sigma_m + \sigma_a\right)^2 + 3\left(\tau_m + \tau_a\right)^2\right]^{1/2} = 52.7206\,MPa$$

Then

$$n_y = \frac{Sy}{\sigma_{\max}} = 15.1743$$

Blade design

Details of blade design are mentioned in table 2.

Calculation of motor - gearbox power

The selection of this component is influenced by power, torque and output speed, based on previous research the output speed required for a tire shredding machine is approximately 30 - 40 rpm.

$P = T \times w$

$P = 3.0555\ KW \approx 4.095\ HP$

Results and Discussion

Gear simulation

As can be seen in figure 4a, it has been found that the gearing will experience a maximum stress of 9.804e+01 N/mm^2 and a factor of safety of 2.5 can also be seen in figure 4b.

Axis simulation

As can be seen in, it has been found that the shaft will experience a maximum stress of 8.047e+01 N/mm^2 and in figure 5b a factor of safety of 8.8.

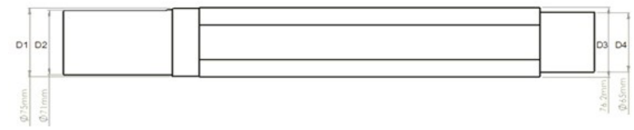

Figure 3: Axle sectioning.

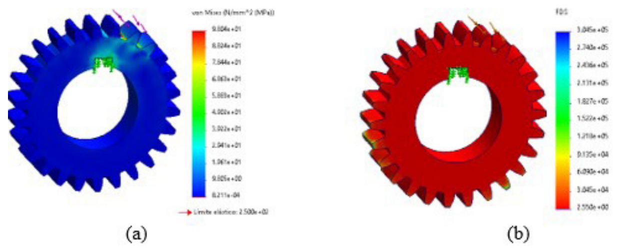

Figure 4: The **(a)** Von Mises stress and **(b)** the safety factor of the gearing.

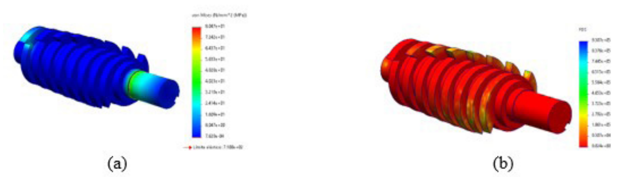

Figure 5: The **(a)** Von Mises stress and **(b)** the shaft safety factor.

Blade simulation

As can be seen in figure 6a, the blade will be subjected to a maximum stress of 4,152e+01 N/mm^2 and in figure 6b it has a safety factor of 6.

Assembly simulation

As can be seen in figure 7a, it has been found that the assembly will experience a maximum stress of 2.669e+02 N/mm^2, figure 7b shows a maximum displacement of 4.313e-02 mm.

Simulation of the structure

It was found in figure 8a that the structure will experience a maximum axial and bending stress of 1.295e+01 N/mm^2. Furthermore, in figure 8b, it was observed that it has a factor of safety of 19.

Table 2: Blade design.

Alternating and medium force (N)	Shear stress (MPa)	Von Mises efforts (MPa)	Fatigue strength limit (MPa)
$F_a = \frac{T}{r}$	$\tau_a = \frac{F}{A_{cortante}}$	$\sigma_a' = \sqrt{\sigma_x^2 + \sigma_y^2 - \sigma_x\sigma_y + 3\tau_{xy}^2}$	$Se' = 0.5 * 550MPa$
4681.9328	11.7048	20.2733	275
Modified fatigue strength limit (MPa)	**Fatigue safety factor**	**Maximum blade stress (MPa)**	**Factor of safety for contact pressure failure**
$Se = Ka * Kb * Kc * Kd * Ke * Kf * Se'$	$Nf = \frac{1}{\frac{\sigma_a}{Se} + \frac{\sigma_m'}{Sut}}$	$\sigma_{\max} = \frac{F_m + F_a}{A_{contacto}}$	$Ns = \frac{Sy}{\sigma_{\max}}$
108.0173	4.4355	23.4097	10.6793

Figure 6: The **(a)** Von Mises stress and **(b)** the blade safety factor.

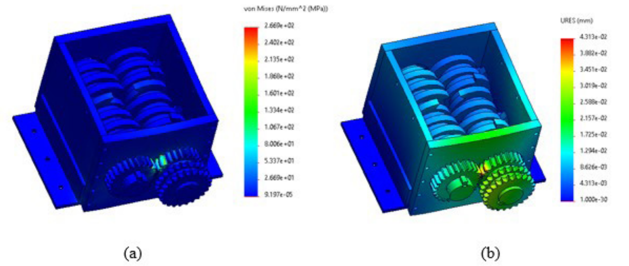

Figure 7: The **(a)** Von Mises stress and **(b)** the resulting assembly displacements.

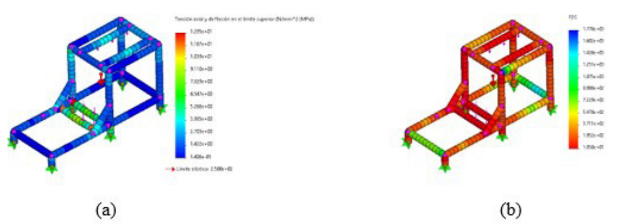

Figure 8: The **(a)** Von Mises stress and **(b)** the safety factor of the structure.

To obtain the granulometry, a grinding process is carried out which consists of several passes through the machine, i.e., depending on how fine or coarse the rubber grain is desired, more, or fewer passes through the machine will be needed.

To obtain the granulometry, a grinding process is carried out, which consists of making several passes through the machine, i.e., depending on how fine or coarse the rubber grain is desired, more, or fewer passes through the machine will be needed. For the tests, treads of 235/70 R16 tires were used on which the different types of bio-coatings considered as nano-coatings containing natural fatty acids present in table 3 were applied.

Conclusions

The geometry was determined according to the crushing capacity it would have, in this case it was for a tire tread smaller or equal to 235/70 R16, since preliminarily the machine should be easy to assemble and disassemble as well as not being fixed to the ground. Pre-design calculations were considered by the authors of this paper considering the torque required to cut the steel tire cords, the area and maximum tensile strength of the cords, the angular velocity at which the shaft rotates to select a suitable geared motor to meet the expectations of the pre-design requirements.

For the detailed design, we started from the gear, to obtain

Table 3: Uncoated pneumatic tire shear results vs nano-coated pneumatic tire.

	Tire in its natural state	Nano-coated treated tire		
		Palm oil	Soybean oil	Coconut oil
Rubber grain (mm)	30	22	28	28
Time (s)	42	35	41	41
Volume of raw material (m^3)	0.41	0.41	0.41	0.41
Maximum blade stress (MPa)	23.4097	19.659	22.352	22.352

the distance between centers, an economically accessible material was selected so that it is within the reach of the general population, considering the design factors, the loads applied on the gear teeth with their respective bending and fatigue stresses with which the respective safety factors were obtained.

The results obtained from the CAD/CAE software simulation using finite elements show the Von Mises stresses and the safety factor of each element, which validates the results obtained from the design calculations. It was found that the granularity of the tire decreases with the number of passes if a fine grain is desired, the number of times the tire must undergo shredding increases.

An approximate cost study was carried out with the materials that were used, so an estimated cost for the construction of the machine is 6000 USD, this cost varies depending on the locality and availability of materials.

This project contributes to goal 12 of the Sustainable Development Goals, because it contributes to improving sustainable consumption and production patterns by integrating nanotechnology with biodegradable products with a proportional reduction of 16.67% with respect to time and maximum allowable stress in the cutting blade and a reduction of 26.66% in the raw material cut with the use of triglyceride fatty acids present in palm oil with a range of thicknesses from ± 50 nm to ± 40 nm on the tire surface.

Acknowledgments

None.

Conflict of Interest

None.

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