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Transmition System Design for a Green Cane Conveyor

Diseño del sistema de transmisión para una transportadora de caña en verde

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Abstract

This work proposes a design of a conveyor for a prototype of mechanized harvester of green sugar cane. This way of harvest represents a technological opportunity that reduces costs, improves the harvest and minimizes the negative impacts to the environment generated by the traditional method of burning. The design of a green cane harvester consists of several subsystems that start from cutting, defoliation, transportation and a temporary storage. The present work shows the methodology to design the transmission system for a bucket conveyor that will be responsible for the transfer of cane to a container. Within the design, operating factors are taken into account, such as the amount of cane to be transported, the weight of the components, speeds, the power required to move the conveyor, and the type of chains, sprockets, are defined material and diameter of the transmission shaft.

Shaft, Design, Simulation

Resumen

Este trabajo propone un diseño de una transportadora para un prototipo de cosechadora mecanizada de caña de azúcar en verde. Esta forma de cosechar representa una oportunidad tecnológica que reduce costos, mejora la cosecha y minimiza los impactos negativos al medio ambiente que genera el método tradicional de quema. El diseño de una cosechadora de caña en verde consiste de varios subsistemas que parten desde el corte, deshoje, transportación y un almacén temporal. El presente trabajo muestra la metodología para diseñar el sistema de transmisión para un transportador de cangilones que se encargara del traslado de caña hacia un contenedor. Dentro del diseño se toman factores de trabajo en operación como la cantidad de caña a transportar, el peso de los componentes, velocidades, la potencia requerida para mover el transportador, además que se definen el tipo de cadenas, catarinas, y se concluye definiendo el material y diámetro del eje de trasmisión.

Ejes, Diseño, Simulación

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Introduction

Mexico produces around 5 million tons of sugar, with a value of more than 3 billion dollars per year, which represents about 12% of the value of the primary sector. The cane production is carried out in 664 thousand hectares that supply 57 sugar mills in 15 sugarcane states. (Colombia, Ceni Caña, 2002)

The traditional practice in the sugar mills for harvesting cane is to burn the plantation to facilitate the harvesting of the stems. However, the burning of a single hectare of cane plantation means that more than 160 kg of carbon dioxide and carbon dioxide are released into the atmosphere. During the harvest season, the sugarcane areas tint their skies of reddish tones due to the large amount of smoke and pieces of burnt leaf that the wind carries to the nearby towns, causing problems with drainage.

By harvesting the cane in green and avoiding combustion in the sugarcane fields, the impact on water pollution is reduced and the environment, the flora and fauna of the sugarcane agro-system is protected. In addition, less application of herbicides is required in the initial stage of crop growth that would translate into higher sugar yields.

The mechanized harvesting of sugarcane represents a window of technological opportunity that not only reduces costs and streamlines the harvesting and delivery of cane to the mill, but also minimizes the negative impacts on the environment, contributing to the productivity of the mills and the improvement of the standard of living of the producers and inhabitants of the sugarcane areas of Mexico.

The technology for the management and harvesting of cane in green is not yet fully developed, due to the fact that its disadvantages are that the soil must be uniform and socially it would not require a large amount of labor. An important step in the design of a harvester in green is the transportation of the cane after a process of cutting and defoliation.

This project determines the necessary design characteristics that the power transmission system must have for a conveyor, which is part of a subsystem of a green cane harvester.

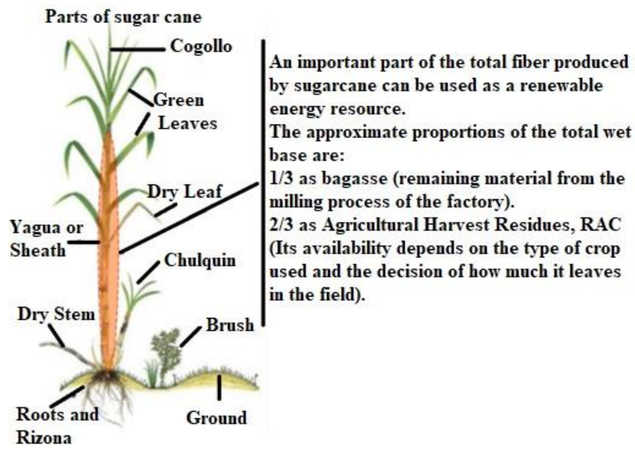

Figure 1 Parts of sugar cane

Description of the Method

The conceptual design of the power transmission system for a sugarcane transporter will be carried out. This design will be applied in a subsystem of transportation, cutting of the bud and storage within the process of harvesting cane in green.

The conveyors are machines of horizontal, vertical or sloping design that are used for the continuous transport of materials in a determined trajectory, until the final point or discharge. They consist of a surface belt that circulates in rollers and pulleys, by a propulsion motor, and all arranged in a structure or support. (Selesiana, 2011). There are conveyors that are driven by gravity and others by motive power such as belt conveyors, slats, drag, tires, vibratory, rollers, screws and bucket elevators. Among the common transporters in the agroindustrial branch we have:

1. Roller conveyors.
2. Band conveyors.
3. Bucket or shingle conveyors.

The roller conveyor is not suitable for transporting cane since the space between rollers will cause waste of the product, as well as encrustations may occur, making it difficult to move the load.

The disadvantages of the belt conveyor is that the material of the textile belts has a short duration when they work with abrasive materials or large pieces and they are more used in the transport of loose material and moderately small pieces. (Budynas, 2012).

The bucket conveyor has the following properties:

– Carries large and heavy loads
– It's slow-moving
– Resists large tensile forces
– The buckets are used to transport different loads
– Supports tilt angles greater than 18°.

The harvest technology in green cane implies the presence of large amounts of foreign matter and buds. By these considerations it is defined that the best option to transport cane in green is the transporter of buckets.

Initial parameters

The initial parameters to develop the design calculations of the bucket conveyor are:

– Amount of sugarcane in green to transport.
– Bucket dimensions.
– Total length of the conveyor.
– Density of the bucket material.
– Density of the cane in green.

(Torn, 2018), establishes that the subsystem 1 of the harvester in green has a mass flow of cane at the output of 75,000 kg / hra. (Mott, 2004) considers a design factor of 1.15 to 2 for structures under static loads with a high degree of confidence in the design data. The mass flow of cane in green for design was calculated as shown:

$$\dot{m}_d = 75{,}000(1.15) = 86{,}250\,\frac{kg}{hra.} \qquad (1)$$

Torn also developed a procedure to determine the density of sugar cane in green. This procedure consists of cutting equal pieces of cane and making dimensions in different sections. The results that were obtained can be seen on table 1.

Measurements made in sections of cane	Values (averages)
Transverse diameter $D_{caña}$	0.035 m
Mass $m_{caña}$	1 *kg/m*
Density $\rho_{caña}$	1058.8996 kg/m^3

Table 1 Average values in cane sections in green

Torn defines that the length of cane at the exit of the cutting subsystem is 2 meters on average. Therefore, with the data from Table 1, the mass flow was calculated according to cane numbers per second.

$$\dot{m}_d = 86{,}250\frac{kg}{hra.}\left(\frac{1\ caña}{2kg}\right)\left(\frac{1\ hra}{3600\ seg}\right) = 11.979\ \frac{cañas}{seg} \cong 12\ \frac{cañas}{seg.}$$

The bucket was designed so that it can transport 3 rods in its interior, therefore, in the area of admission of the transporter it is necessary to pass 4 buckets per second, so that it receives 12 rods in this time. The bucket is composed of 2 rectangular sections of wood. The dimensions of the bucket can be seen in figure 2.

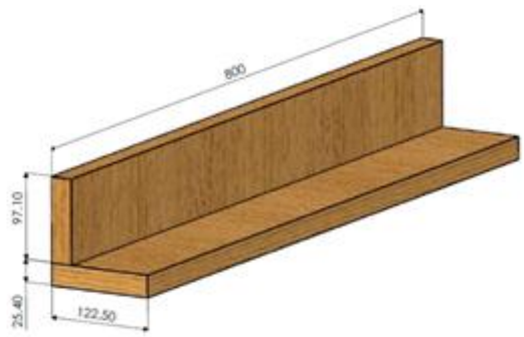

Figure 2 Dimensions of the bucket of matera for the cane transporter in green

The linear speed of the conveyor was calculated according to the dimensions and number of buckets that must pass per second.

$$V = \frac{122.5\ mm\times 4}{1\ seg.} = 490\,\frac{mm}{seg} \cong 0.5\,\frac{m}{seg}$$

Calculation of the force in the periphery of the drum of the driving head

The buckets are made of oak, (Reforestal, 2012) establishes that the density of this material is 650 kg/m^3.

The conveyor belt will have 14 buckets to ensure proper operation. The buckets will be transported by means of a commercial steel chain. Figure 3 shows the distribution of the buckets on the chain.

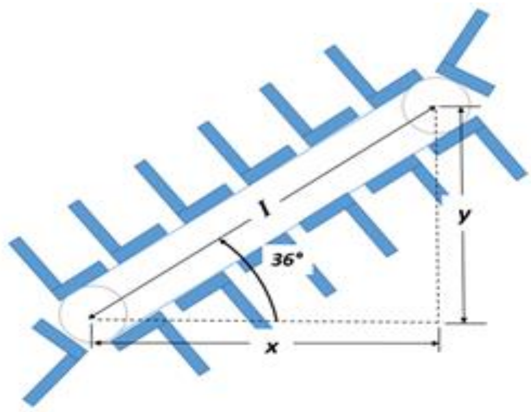

Figure 3 Distribution of the buckets on the roller chain

The total weight of the buckets Q_{p1} is 40.6042 kg., while the approximate length of the chain is calculated according to the number of buckets:

$$L_{Aprox} = 122.5mm \times 14 = 1715mm = 1.715m$$

For design purposes, it is proposed to use an ANSI 50 chain. Table 2 shows some characteristics for standard American roller chains.

Chain Number ANSI	Pitch in (mm)	Width in (mm)	Minimal Resistence to Stress lbf (N/m)	Average Weigth lbf/ft (N/m)	Roller Diameter in (mm)	Spacing of Multiple Standars in (mm)
25	0.250 (6.35)	0.125 (3.18)	780 (3470)	1.09 (1.31)	0.130 (3.30)	0.252 (6.40)
35	0.375 (9.52)	0.188 (4.76)	1760 (7830)	0.21 (3.06)	0.200 (5.08)	0.399 (10.13)
41	0.500 (12.70)	0.25 (6.35)	1500 (6670)	0.25 (3.65)	0.306 (7.77)	———
40	0.500 (12.70)	0.312 (7.94)	3130 (13920)	0.42 (6.13)	0.312 (7.92)	0.566 (14.38)
50	0.625 (15.88)	0.375 (9.52)	4880 (21700)	0.69 (10.1)	0.400 (10.16)	0.713 (18.11)

Table 2 Characteristics for standard American roller chains (Budynas, 2012)

The weight of the chain Qp2 is calculated as follows:

$$Q_{p2} = \left(10.1\frac{N}{m}\right) * \left(\frac{1KG}{9.81N}\right) * (1.715\ m) = 1.7657\ kg$$

The weight of the belt and the rolling parts Q_p is obtained from the sum of the weight of the buckets Q_{p1} and the weight of the chain Q_{p2}

$$Q_p = 40.6042kg + 1.7657\ kg = 42.37\ kg$$

Once Qp has been determined, the weight of the tape is calculated according to the length qp defined by the following equation:

$$q_P = \frac{Q_P}{I} * g \qquad (1)$$

Where I is the wheelbase of the conveyor pulleys. Considering 6 canjilones and 5 separations according to figure 3 we have I = 0.8575 m approximately. Substituting the values in equation 1 you get:

$$q_p = \frac{42.37\ kg}{0.8575m} = 49.411\ \frac{kg}{m} * \left(\frac{9.81m}{s^2}\right) = 484.722\ N/m$$

The distance between centers is divided horizontally and vertically as shown in figure 3, because the conveyor has an inclination angle of 36°

$$\frac{0.8575\ m}{sen\ 90} = \frac{x}{sen\ 54} \quad x = 0.6937\ m$$

$$\frac{0.8575\ m}{sen\ 90} = \frac{y}{sen\ 36} \quad y = 0.5040\ m$$

A fictitious assumption is calculated to increase the distance between axes I0 with equation 2.

$$I_0 = \frac{50-0.2\ x}{2} \qquad (2)$$

In this case, we have the approximate distance defined $I_0 = 0$.

The force required to move the belt in vacuum is calculated using the following equation:

$$P_1 = f' q_p(I + I_o) \qquad (3)$$

To find the value of P_1, a coefficient of friction or friction is selected, using $f' = 0.03$ (Mott, 2004).

Substituting in equation 3 you get:

$$P_1 = (0.03)\left(484.722\ \frac{N}{m}\right)(0.8575\ m) = 12.4694\ N$$

Force needed to move the material P_2

The equation to determine the force needed to move the material P_2:

$$P_2 = f' q_m(I + I_o) \qquad (4)$$

Where q_m is the weight of the material transported per linear meter of the conveyor (kg/m).

$$q_m = \frac{\dot{m}_d}{3.6\ V} * g \qquad (5)$$

$$q_m = \frac{23.9583\,\frac{kg}{s}}{3.6\,\left(0.5\,\frac{m}{s}\right)} * (9.81\,\frac{m}{s^2}) = 130.571\,\frac{N}{m}$$

With the results obtained, it is substituted in equation 4 to obtain P_2.

$$P_2 = (0.03)\left(130.571\,\frac{N}{m}\right)(0.8575\;m) = 3.358\;N$$

Force needed to move the material vertically P_3

Because the conveyor has an angle of inclination, not only transport the material horizontally but also vertically, that is why the force required to move the material P_3 vertically with the following equation is calculated:

$$P_3 = \frac{\dot{m}_d y}{3.6V} * g \qquad (6)$$

$$P_3 = \frac{\left(23.9583\frac{kg}{s}\right)(0.5040m)*(\frac{9.81m}{s^2})}{(3.6)(0.5m/s)} = 65.8055\;N$$

Finally, adding the three loads P_1, P_2 and P_3, we obtain the value of the total force P_T in the periphery of the sprocket wheel:

$$P_T = P_1 + P_2 + P_3 \qquad (7)$$

$$P_T = 81.636\;N$$

Calculation of power absorbed by the conveyor

Known the efforts in the periphery of the wheel Catarina and the speed of the tape, it is immediate the calculation of the respective powers absorbed by the transporter defined by the following equation:

$$N_T = \frac{P_T}{75} V \qquad (8)$$

Substituting the values obtained in equation 8 is obtained:

$N_T = 0.5442\;CV$ = 0.537 hp

It is defined that the commercial engine to be selected will have a power of 3/4 Hp.

Selection of the chain

The next step was to determine the type of roller chain that supports N_T power to move the conveyor properly. The speed of entry and exit of the conveyor remains constant, (Budynas, 2012) proposes a procedure for the selection of chain at different speeds of rotation. The design data is shown in table 3.

Ratio of speeds n1 / n2	1
Power absorbed by the N_T conveyor	0.537 Hp
Service factor Ks	1.3
Design factor nd	1.5
Number of teeth of the driving Catarina for low speeds N	17
K1 preextreme power	1
Number of strands K_2	1

Table 3 Design data for the selection of the roller chain

Equation 9 determines the necessary power capacity. Thus

$$H_{TAB} = \frac{\eta_d k_s H_{nom}}{k_1 k_2} \qquad (9)$$

$$H_{TAB} = \frac{(1.5)(1.3)(0.537)}{(1)(1)} = 1.047$$

According to the compiled information of the ANSI B29.1-1975 standard described by (Budynas, 2012) it is defined that the chain to be used is ANSI 50 with a step p of 15.88 mm since it supports a nominal capacity of 1.34 hp at a 100 rpm speed.

The selection of the roller chain is a function of the revolutions per minute at which it rotates. The diameter of Catarina D was calculated as shown:

$$D = \frac{p}{sen(\frac{180}{N})} = 8.642\;cm$$

To determine the angular velocity ω the approximation of the linear velocity V was taken into account, considering the passage of 4 buckets per second.

$$\omega = \frac{2V}{D} = \frac{2 \cdot 0.5\,\frac{m}{s}}{0.0864m} = 11.5741\,\frac{rad}{seg} = 110\;rpm$$

In this way it is checked that the roller chain selection is correct.

The commercial engines of 0.75 hp rotate at higher speeds, to reach the speed ω it will make use of a speed reducer.

(Mott, 2004) suggests a maximum of 50 steps for the distance between centers C of the ladybirds so that the chain does not fail. If we consider that the approximate center distance I is 85.5 cm, then we can define the distance between centers C of 48 steps.

The length of the chain is defined by:

$$L = 2C + \frac{N_1+N_2}{2} + \frac{(N_2-N_1)^2}{4\pi^2 C} \tag{10}$$

$$L = 2(48) + \frac{17+17}{2} + \frac{(17-17)^2}{(4\pi^2)(48)} = 113\ pasos$$

The appropriate type of chain to use for the design of the conveyor is the ANSI 50 chain with a single strand with 113 steps, in addition to the use of type B stainless steel 17-tooth model 50B17SS. (Gear, 2013)

Design of axes

The conveyor shaft will be subject to the following loads:

- Weight of the ladybirds
- Load to transport
- Weight of the buckets
- Weight of the chain
- Tensions in the chain

The correct diameter of the arrow must be designed to ensure satisfactory resistance and therefore correct operation in work operations.

Calculation of the tensions of the chains

At the moment of transmitting the power, F1 is generated in the chain on the taut side. Unlike a band, on the loose side the chain does not exert any force F2 on the loose side. The strength of the taut side is calculated with the following equation (Mott, 2004):

$$F_1 = \frac{33000(H)}{V} \tag{11}$$

The power to transmit H is the result of the power of the commercial motor selected. Substituting data in equation (11) we have:

$$F_1 = \frac{33000(0.75\,Hp)}{98.5\,\frac{ft}{min}} = 251.269\ lb = 1117.7N$$

Calculation of all the loads that act on the axis

Calculation of all the loads that act for calculation purposes, a resultant force F_C is considered, which is the sum of the forces produced by the 14 buckets, the 2 chains and the cane that is transported as shown in figure 4 on the axis.

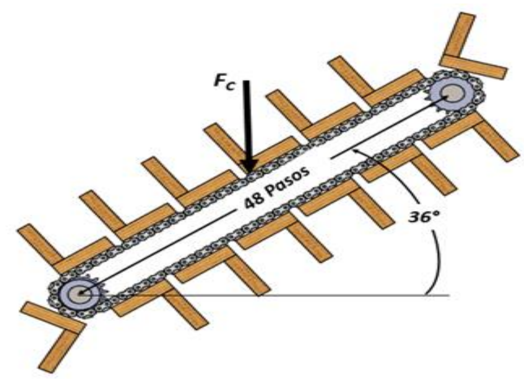

Figure 4 Resulting force applied to the centroid of the conveyor

Table 4 shows all the forces that apply on the centroid of the assembly.

Buckets (14 pieces) F_D	398.5 N
Chain F_E	18 N
Cane F_F	358.5 N

Table 4 Forces produced by several elements

For the lower shaft the force produced by the chain counteracts the resultant force F_C. Therefore, the design calculations are based on the upper axis, which receives the highest load. Figure 5 shows the free body diagram. The force F_G corresponds to the weight of the Catarina that is of 8 N. The forces R_{AX}, R_{AY}, R_{EX}, R_{EY} correspond to the reactions that are generated in the zone of support of the axis.

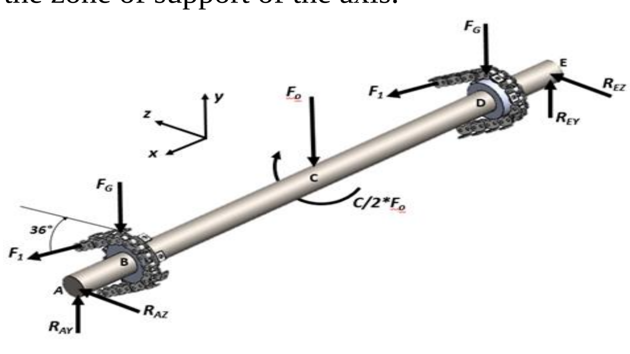

Figure 5 Reactions that are generated on the upper axis of the conveyor

Applying the equations of static equilibrium, the reactions in the supports were determined which are shown in table 5.

R_{AY}	1052.47 N
R_{AZ}	904.238 N
R_{EY}	1052.47 N
R_{EZ}	904.238 N

Table 5 Values of the reactions in the supports of the arrow

Critical point of the axis

To determine the critical point of the axis, the diagrams of the distribution of the bending and torsion moments were made. These diagrams are shown in figures 6-8.

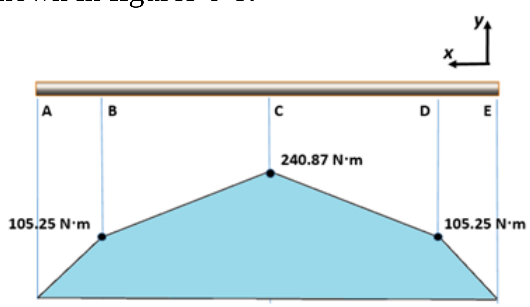

Figure 6 Diagram of bending moments in the flat xy

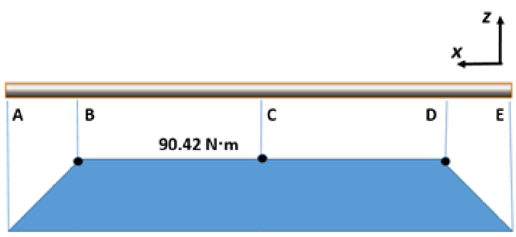

Figure 7 Diagram of bending moments in the xz plane

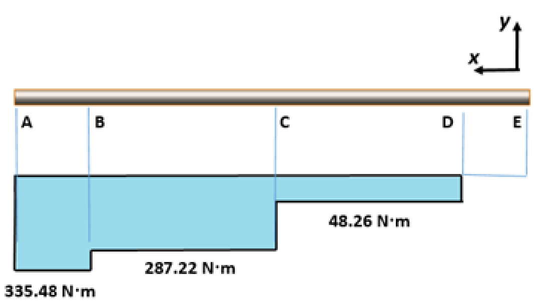

Figure 8 Diagram of torsional moments.

According to the diagrams, it can be seen that the critical zone is point C, since it is the place where the bending moment is maximum. Determining the magnitude of the bending moment.

$$M_c = \sqrt{240.87^2 + 90.42^2} = 257.282\ Nm$$
$$T_c = 287.22\ N \cdot m$$

Shaft design under static loads.

At moments at point C, a safety factor of 2 will be added because this mechanism will operate in non-uniform places and will be subject to vibrations and therefore increase impact efforts.

Many axles are made of low carbon steel, cold drawn steel or hot rolled steel such as ANSI 1020-1050 steels (Budynas, 2012). The material selected for the axis is an ANSI 4140 steel, with yield strength Sy of 665 MPa and ultimate tensile stress Sut of 758 MPa.

The minimum diameter required for the arrow is calculated as shown:

$$d = \left(\frac{32ns}{sy\,\pi}\sqrt{(M_c)^2 + (T_c)^2}\right)^{\frac{1}{3}} \tag{11}$$

$$d = \left[\frac{32(2)}{\pi(655 \times 10^6)}\sqrt{(257.282)^2 + (287.22)^2}\right]^{\frac{1}{3}} = 0.0228\ m \approx 22.8\ mm$$

The SolidWorks software was used to verify by means of a finite element analysis the results obtained analytically. Figure 9 shows the results of the safety factor.

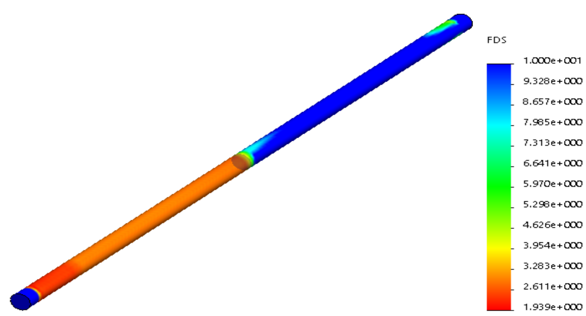

Figure 9 Distribution of the safety factor in the arrow

It can be seen that the safety factor approaches 2 as established in the calculations.

Shaft design under dynamic loads.

The behavior of parts of machines is different when subjected to varying loads. The case of the transporter arrow has a sinusoidal stress ratio completely reversed because it rotates at a constant revolution.

The minimum diameter of the arrow can be determined by combining the forces by the energy of the distortion and using elliptical ASME as the criterion of failure.

$$d = \left\{ \frac{16n}{\pi} \left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}} \quad (12)$$

In the design of the arrow is considered the absence of irregularities or discontinuities, such as holes, grooves or notches that significantly increase the efforts. Therefore, the factors of axial stress concentrator Kf and shear Kfs are equal to 1.

In the case of a rotating shaft with constant bending and twisting, the bending stress is completely reversible and the torsion is constant. Equation 12 can be simplified by matching Mm and Ta to 0, which simply eliminates some of the terms (Gear, 2013).

Considering a factor of surface, size and modification per load, it is determined that the limit of fatigue resistance is 254,466 Mpa. If you take a safety factor of 1.5 for the design and substitute in equation 12 you have:

$$d = \left\{ \frac{16(2)}{\pi} \left[4\left(\frac{(1)(257.282(2))}{254.466\times10^6}\right)^2 + 3\left(\frac{(1)(287.22)(2)}{655\times10^6}\right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}}$$

$$d = 0.0282m = 28.2\,mm$$

The result obtained represents the minimum value for the diameter of the arrow to resist the dynamic loads.

Shaft design under critical speeds

When an axis rotates, the eccentricity causes a deflection due to the centrifugal force that is resisted by the flexural stiffness of the EI axis. As long as the deflections are small, no harm is caused. However, another potential problem is the critical speeds where at certain speeds the axis is unstable and the deflections increase without an upper limit (Budynas, 2012).

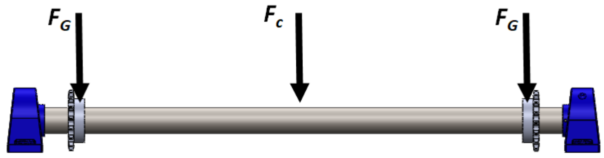

Figure 10 Loads on the arrow for calculating the critical speed

For calculating the critical speed of the arrow, the weight of the FG sprockets will be disregarded, since it is small compared to the resulting load F_c.

For an axis with a single mass, where the mass of the axis is small compared to the mass that is attached, the first critical velocity can be calculated approximately as a function of gravity g and the deflection of the axis at the location point of the mass δ:

$$\omega_c = \sqrt{\frac{g}{\delta}} \quad (13)$$

To determine the deflection at the location of the force Fc, the method of moments area was used.

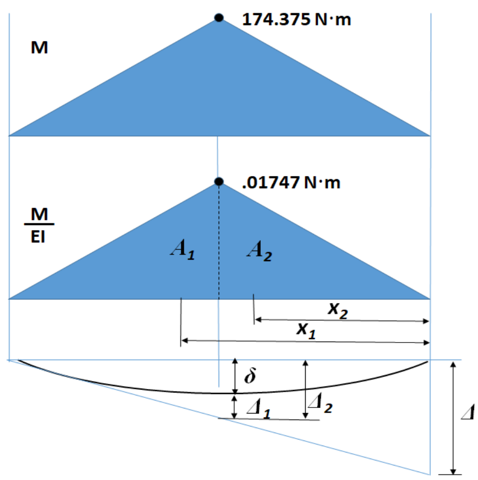

Figure 11 Moments area

The results are shown in table 6.

A₁, A₂ → A_1, A_2	3.9307×10^{-3} m^2
x_1	0.6 m
x_2	0.3 m
Δ	3.5376×10^{-3} m
Δ_1	5.8960×10^{-4} m
Δ_2	1.7688×10^{-3} m
δ	1.1792×10^{-3} m

Table 6 Moments Area Values

By replacing the value of the deflection in equation number 13, the value of the first critical velocity is obtained:

$$\omega_c = \sqrt{\frac{9.81\frac{m}{s^2}}{101.1792x10^{-3}m}} = 91.20\frac{Rad}{s} \; = 870.9\ rpm$$

It is observed that the critical speed is above the operating speed ω = 110 rpm. This confirms that there will be no large deflections of the arrow in normal operation of the machine.

A frequency analysis was carried out in the SolidWorks software, in order to check the results obtained analytically.

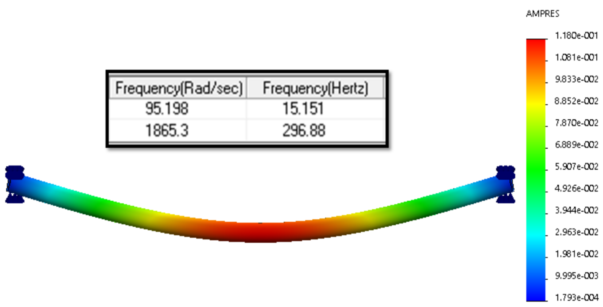

Figure 12 Frequency simulation results

As shown in figure 12. A natural frequency of 95,198 rad / sec is obtained result very similar to that obtained analytically.

Results

To obtain a correct design of the power transmission system it is necessary to design with static and dynamic loads and analyze the critical speeds. It is observed that the diameter obtained by static loads would not be able to support the dynamic loads, since the piece would fail after a certain cycle of workloads. Therefore, the minimum diameter will be 28.2mm.

Commercially the steel bars are in standardized measures, so the diameter approaches a dimension of 1.25 inches (31.75mm) that is common to find in establishments selling steel.

Figure 13 shows the results of the safety factor considering a diameter of 1.25 inches, it can be observed, as is logical, that the safety factor increases to 4.79 in the critical zone.

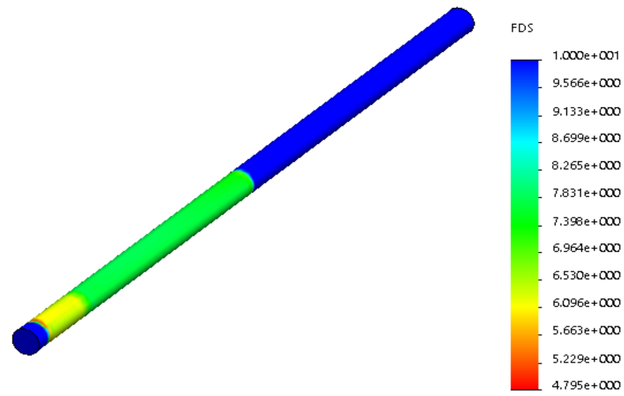

Figure 13 Distribution of the safety factor in the arrow with a diameter of 1.25 inches

A dynamic analysis was developed to check the damage that can be generated in the arrow. It is considered 1x106 work cycles under normal load conditions. Figure 14 shows that there will be no damage produced with these considerations.

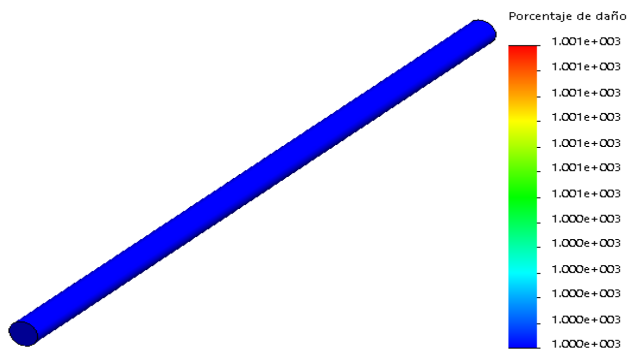

Figure 14 Percentage of arrow damage produced by dynamic analysis

It is verified that the arrow does not fail due to static loads or dynamic loads. Based on this study, the virtual model of the cane transportation system has been developed, the commercial components were adapted and the structure on which the power transmission system will be assembled was developed as shown in figure 15. It is observed that the transmission system was hidden with the side hoppers avoiding any contamination to the cane

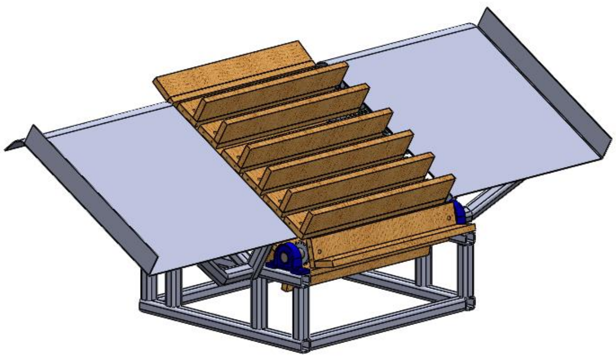

Figure 15 Virtual model of the cane harvester conveyor in green

Conclusions

The correct design of the sugarcane transporter in green was determined based on research and development of calculations. They went determining parameters of entrance, according to the special characteristics that have in a process of mechanized harvest.

The selection of materials and components are the result of calculation memory, coupled with the characteristics and mechanical properties provided by suppliers.

To determine the diameter of the arrow it is fundamental to make the static and dynamic analysis, since a variation in the result was observed, which can cause a bad design in the mechanical component. The analytical model by static charges was verified by a finite element analysis, concluding that the result of the determination of the minimum diameter by static charges is correct.

Recommendations

For future work we have the following points:

1. The prototype must be built in real scale to verify the results obtained in this present work.
2. It is proposed to perform different simulations in CAD by changing the type of material.
3. Complement the study with the analysis of selection of other mechanical components such as bearings, couplings, pins, etc.
4. Select the commercial motor 0.75 hp and a speed reducer to reach the value of $\omega = 110\ rpm$

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