ISSN 2531-2189

Volume 6, Issue 17 – January – June - 2022

Journal of Mechanical Engineering

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Journal of Mechanical Engineering Volume 6, Issue 17, January- June 2022, is a journal edited semestral by RINOE. 38 Matacerquillas, Moralzarzal - CP-28411. Madrid - Spain. WEB: www.ecorfan.org/spain, revista@ecorfan.org. Editor in Chief: SERRUDO-GONZALES, Javier. BsC. ISSN-2531-2189. Responsible for the latest update of this number ECORFAN Computer Unit. ESCAMILLA-BOUCHÁN, Imelda. PhD, LUNA-SOTO, Vladimir. PhD. 38 Matacerquillas, Moralzarzal - CP-28411. Madrid - Spain, last updated June 30, 2022.

Journal of Mechanical Engineering

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Presentation of the Content

In the first chapter we present, *Partial Hand Prosthesis: Design and Simulation* by MENDOZA-MIRANDA, Juan Manuel, SALAZAR-HERNÁNDEZ, Carmen, SALAZAR-HERNÁNDEZ, Mercedes and RAMÍREZ-MINGUELA, José de Jesús, with adscription in the Instituto Politécnico Nacional and Universidad de Guanajuato, as next article we present, *Analysis of the mechanical behavior of the chassis of a vehicle driven by human power through the application of the FEM module of a commercial CAD Software* by TORRES-RICO, Luis A., MANRÍQUEZ-PADILLA, Carlos G., SÁNCHEZ-LERMA, Josué R. and VELAZQUEZ-BAUTISTA, Juan E., with adscription in the Universidad Politécnica de Juventino Rosas and Universidad Autónoma de Querétaro, as next article we present, *Development of design proposals for a test box for medium soils in direct shear test* by PESCADOR-GUTIÉRREZ, Mariana, with adscription in the Centro de Tecnología Avanzada (CIATEQ, A.C.), as the last article we present, *Evaluation of modal frequencies obtained with the impact hammer technique on an epoxy matrix composite material reinforced with glass fibers* by COCA-GONZALEZ, Juan Manuel, AVILA-HERNÁNDEZ, Sergio Albano, REYES-SOLÍS Alberto and TORRES-CEDILLO, Sergio Guillermo, with adscription in the SEPI-ESIME TIC. IPN and Centro Tecnológico Aragón, FES- UNAM

Content

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Partial Hand Prosthesis: Design and Simulation

Prótesis parcial de mano: Diseño y simulación

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DOI: 10.35429/JME.2022.17.6.1.6 Received: March 30, 2022; Accepted: June 30, 2022

Abstract

Nowadays, engineering has attempted to find a way to replace the loss of human limbs with unnatural devices that approximate their actual function. Current devices vary from hooks, mechanical prostheses, or simply cosmetic prostheses, where an important component when deciding which one to use is the cost of it, even above functionality. The aim of this article is the development of a mechanical prototype for partial hand prosthesis for patients who have suffered partial (middle and distal phalanges) or total (proximal, middle, and distal phalanges) finger mutilation. The results of this work show that the mechanism and the proposed design of the prosthesis are satisfactory to simulate the movement of a human finger, while the finite element analysis shows that the proposed design can adequately support the loads for extreme conditions.

Partial hand prosthesis, Functional prosthesis, Six-bar mechanism, Simulation, Distal, Proximal

Resumen

Actualmente la ingeniería ha intentado encontrar una solución de sustituir la pérdida de miembros humanos con dispositivos que aproximen su función real. Los dispositivos actuales varían desde ganchos, prótesis mecánicas, o simplemente prótesis cosméticas, donde un componente importante a la hora de decidir cuál utilizar es el costo de esta, incluso por encima de la funcionalidad. El objetivo de este artículo es el desarrollo de un prototipo mecánico de prótesis parcial de mano para pacientes que han sufrido una mutilación parcial (falanges medias y distales) o total (falanges proximales, medias y distales) de los dedos. Los resultados de este trabajo muestran que el mecanismo y el diseño propuesto de la prótesis son satisfactorios para simular el movimiento de un dedo humano, mientras que el análisis de elementos finitos muestra que el diseño propuesto puede soportar adecuadamente las cargas comunes.

Prótesis parcial de mano, Prótesis funcional, Mecanismo de seis barras, Simulación, Distal, Proximal

Citation: MENDOZA-MIRANDA, Juan Manuel, SALAZAR-HERNÁNDEZ, Carmen, SALAZAR-HERNÁNDEZ, Mercedes and RAMÍREZ-MINGUELA, José de Jesús. Partial Hand Prosthesis: Design and Simulation. Journal of Mechanical Engineering. 2022. 6-17: 1-6

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1. Introduction

The hand is extremely important and therefore exposed to a range of injuries that can affect its function, resulting in disabilities that lead to serious problems in daily life (Binvignat et al. 2012). In Mexico, limb amputation is the fourth cause of disability, with most victims being people of working age between 20 and 45 years (Sandoval 2011). The replacement of some of the human limbs with devices has been performed for more than two thousand years (Perez Romero 2011) and is aimed at enabling affected individuals to incorporate into their daily lives. Therefore, prostheses play a key role as it means an increase in the quality of life of people who have suffered amputation.

Prostheses can be classified into two categories: passive and active. The first ones, also known as cosmetic prostheses, aim to reestablish the appearance although they lack mobility (Pérez Romero 2011). These prostheses are commonly manufactured with silicone or chlorinated polyvinyl chloride and offer advantages and disadvantages in terms of durability, realism, and price. Among the active prostheses are the mechanical prostheses (Loaiza and Arzola, 2011; Torres-SanMiguel, 2022), which are devices that are used to control closing or opening at any time, thus partially recovering the functionality of the hand.

Some reference examples of finger designs are those used in Standford/JPL hands (Matthew and Kenneth, 1985), Utah MIT (Jacobsen et al. 1986), TUAT/Karlsruhe (Schulz and Pylatiuk, 2011), DLR (Butterfass et al. 1998), and those developed by NASA (Lovchik and Diftler, 1999) and those designs reported by Díaz Montes (Díaz Montes, 2013), Piña Quintero (Piña Quintero, 2010).

The recent invention that does not use electronic devices for its operation is called X-finger, which is specifically designed to address partial finger amputations. Each of these prostheses is individually manufactured to accommodate different amputation cases (Didrick D. 2005). This type of mechanical designs can present some disadvantages such as: limitations of movements caused by the restriction of the mobility of the proposed mechanisms, excessive weight due to the material with which they are manufactured.

In addition, it cannot be considered to make grips where too much force is printed and the main disadvantage is caused by its high cost, which makes them not accessible to all people of working age.

Therefore, the aim of this investigation is to design and analyze a mechanical prototype of a partial hand prosthesis for patients who have suffered mutilation of one or more fingers of the hand either partial (middle and distal phalanges) or total (proximal, middle, and distal phalanges).

2. Human hand anatomy

Anatomical knowledge of the human hand is a fundamental step in the design of prostheses, since it is important to know all the possible movements that it can perform. The skeleton of the human hand can be divided into three regions: i) the carpal region (wrist), ii) the metacarpal region and iii) the phalangeal region (fingers), the last of which can be further subdivided into proximal middle and distal phalanges (see Figure 1). Table 1 shows the dimensions of the phalanges of fingers and Table 2 the range of movements. Thus, it can be stated that the human hand is a complex structure with 22 degrees of freedom that allow multiple configurations of grasping and manipulation. Each finger, except for the thumb, has two hinge joints and a joint at the base with two degrees of freedom, where one of its rotation axes is parallel to the rotation axes of the joints and the second is perpendicular to it and normal to the palm (Gutiérrez T. 2010).

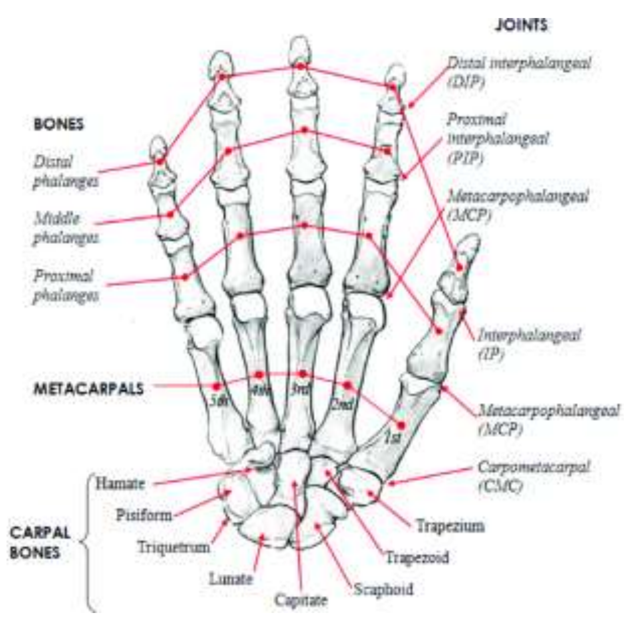


Figure 1 Human hand anatomy
(Nanayakkara et al., 2017)

Phalange type	Finger length (mm)			
	Index	Middle	Ring	little
Proximal	32.50±0.5	37.50±0.5	35.50±0.5	27.5±0.5
Middle	28.50±0.5	30.00±0.5	29.20±0.5	20.00±0.5
Distal	21.50±0.5	21.50±0.5	21.50±0.5	13.00±0.5

Table 1 Index, middle, ring, and minimum finger dimensions
(Gutiérrez, 2010).

	Fingers range of motion of flexion/extension			
	Index	Middle	Ring	little
Distal inter-phalangeal	72°/8°	71°/8°	63°/8°	65°/8°
Proximal Inter-phalangeal	102°/7°	105°/7°	108°/6°	106°/9°
Metacarpo-phalangeal	86°/22°	91°/18°	99°/23°	105°/19°

Table 2 Finger range of motion
(Gutiérrez, 2010)

3. Mechanism synthesis

The synthesis of mechanisms consists of finding a solution to the problems of trajectory generation, operation, and movement (Erdman and Sandor, 2007).

The proposed mechanism consists of six linkages joined together, which form two four-bar mechanisms, the first one is located at the base of the finger that provides movement to the middle phalange, which in turn, activates the other mechanism to move the distal phalange. Thus, the activation is performed by the L_1 and L_3 link which are connected to the middle phalange producing flexion in the distal phalange, as shown in Figure 2.

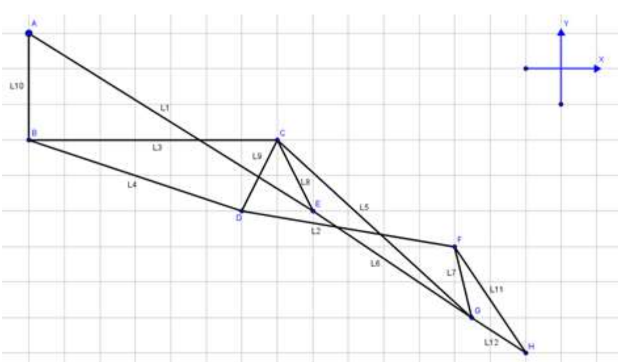


Figure 2 Six-bar mechanisms generating the mobility of the prosthesis prototype

To verify the mechanism's mobility and functionality, a position analysis is performed between the proposed linkages and types of joints.

The position of the prototype was obtained based on a kinematic analysis using the vectorial loops method, since there are two four-bar mechanisms generated from the 6-bar mechanism from Figure 2. The analysis shows a generalized method, since the mechanism is repeated in each of the fingers, changing only the dimensions of the linkages according to the dimensions of the fingers to which a prosthesis is to be made.

Equation (1) describes the position of the vector loop 1, shown in figure 3.

$$\overrightarrow{L_{10}} + \overrightarrow{L_3} - \overrightarrow{L_8} - \overrightarrow{L_1} = \vec{0} \qquad (1)$$

Dividing the vector loop 1 into its Cartesian components we obtain the following equation system for position:

$$L_1 \cos\phi_1 = L_{10}\cos\phi_{10} + L_3\cos\phi_3 - L_8\cos\phi_8 \qquad (2)$$

$$L_1 sen\phi_1 = L_{10} sen\phi_{10} + L_3 sen\phi_3 - L_8 sen\phi_8 \qquad (3)$$

Solving for ϕ_8 we have:

$$(A^2 - B^2)\tan^2\phi_8 + 2BC\tan\phi_8 + (A^2 - C^2) = 0 \qquad (4)$$

where:

$$A = L_1^{\,2} - L_8^{\,2} - L_3^{\,2} - L_{10}^{\,2} \qquad (5)$$
$$B = 2L_{10}L_8 sen\phi_{10} \qquad (6)$$
$$C = 2L_3L_8\cos\phi_3 \qquad (7)$$

On the other hand, Equation (8) describes the position of the vector loop 2, shown in figure 4.

$$\boldsymbol{\overrightarrow{L_2} + \overrightarrow{L_7} - \overrightarrow{L_5} - \overrightarrow{L_9} = \vec{0}} \qquad (8)$$

Calculating the components of each vector, and substituting in the vector loop 2 we have the following system of position equations:

$$L_2\cos\phi_2 = L_5\cos\phi_5 + L_9\cos\phi_9 - L_7\cos\phi_7 \qquad (9)$$
$$L_2 sen\phi_2 = L_5\cos\phi_5 + L_9\cos\phi_9 - L_7\cos\phi_7 \qquad (10)$$

Solving for ϕ_7 we have:

$$\left(A^2 - C^2\right)\tan^2\phi_7 - 2BC\tan\phi_7 + \left(A^2 - B^2\right) = 0 \qquad (11)$$

Where:

$$A = L_2^{\,2} - L_5^{\,2} - L_9^{\,2} - L_7^{\,2} - 2L_5L_9\cos\phi_9\cos\phi_5 \\ -2L_5L_9 sen\phi_5 sen\phi_9 \tag{12}$$

$$B = -2L_7L_5\cos\phi_5 - 2L_7L_9\cos\phi_9 \tag{13}$$

$$C = -2L_7L_9 sen\phi_9 - 2L_7L_5 sen\phi_5 \tag{14}$$

Equation (11) describes the motion of the second vector loop for the four-bar mechanism that is formed between point C, D, F and G (Figure 4) which provides the motions of the prosthesis prototype.

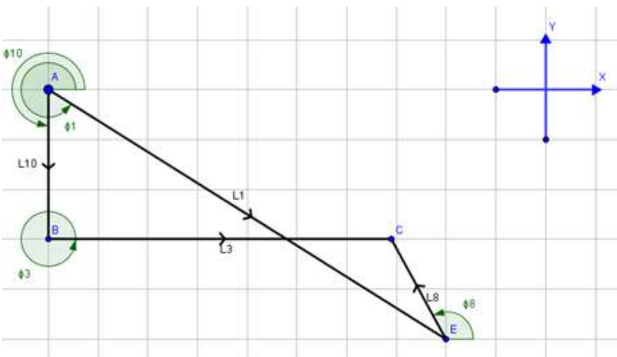


Figure 3 Analysis of the first four-bar mechanism

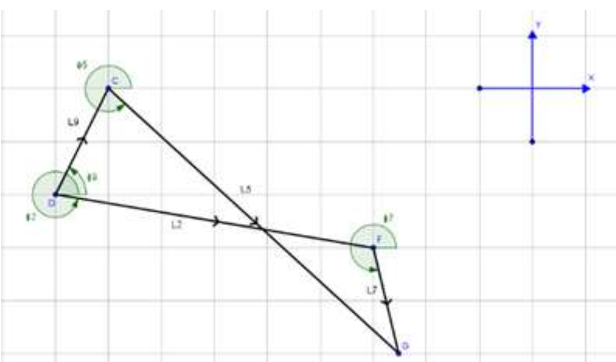


Figure 4 Analysis of the second four-bar mechanism.

4. Stress analysis

Once the mobility of the mechanism and the trajectories were determined, a stress and deformation analysis of the proposed mechanism was performed by means of finite elements (FEM), this allows proposing a suitable material for the subsequent manufacture of the prototype considering the weight and the redesign of the critical areas where stress concentrations are present. For this purpose, some critical movements of the prototype are considered, where the application of force on the phalanges is required.

5. Results

The analysis and synthesis of the proposed mechanism provides the necessary elements to determine the desired trajectories of the mechanism simulating the hand opening and closing. From this analysis and with the typical dimensions of the fingers, the prototype shown in Figure 5 was obtained. The trajectories followed by the distal phalange of the mechanisms of each finger of the hand are shown in Figure 6, where a similarity to the movement of the hand can be appreciated.

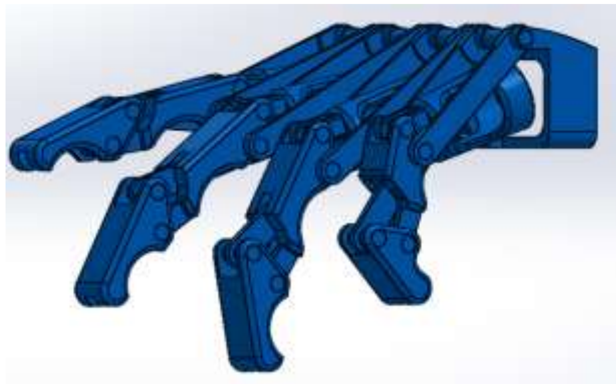
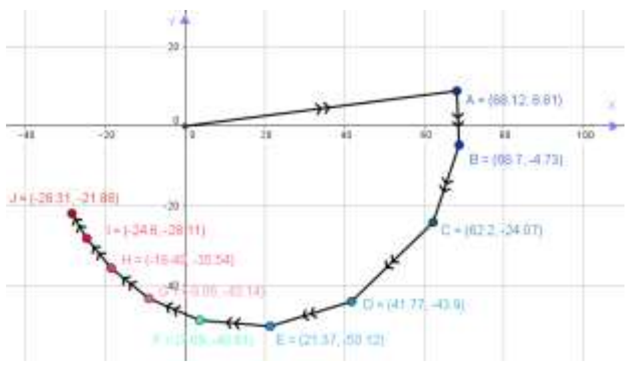
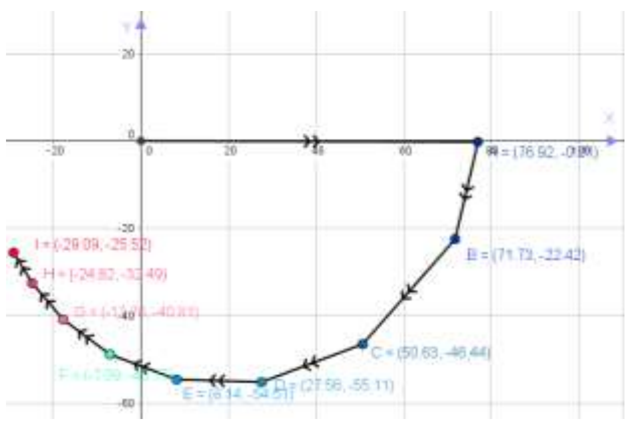


Figure 5 Prosthesis prototype for four fingers of the hand.



(a)



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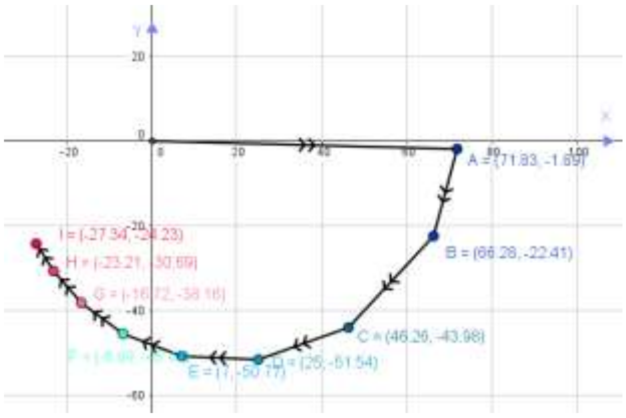

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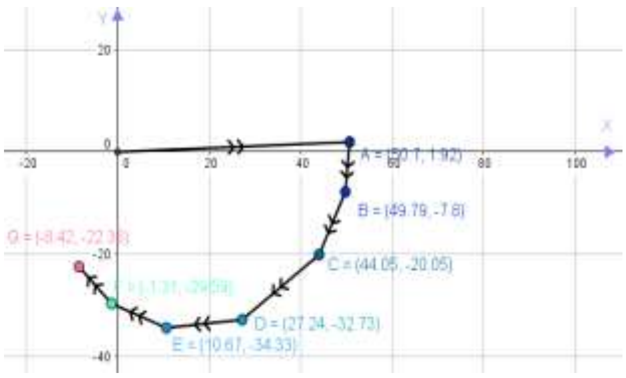

(d)

Figure 6 Prosthesis movement path for (a) index finger, (b) middle finger, (c) ring finger, (d) little finger

The results from the finite element analysis of the total deformation generated in the distal phalange is shown in Figure 7, having a maximum displacement of 1.15 mm. On the other hand, Figure 8 indicates the maximum Von Mises stress, produced in the bars supporting the prototype. The maximum value calculated was 156 MPa and was presented in the geometry changes of the same bars where the stress concentrators are located.

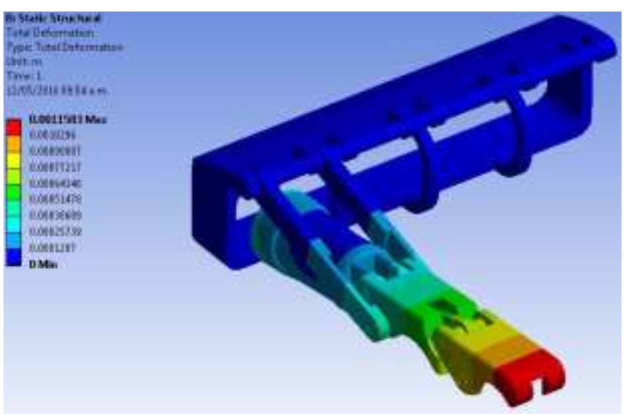

Figure 7 Deformation distribution of the prototype

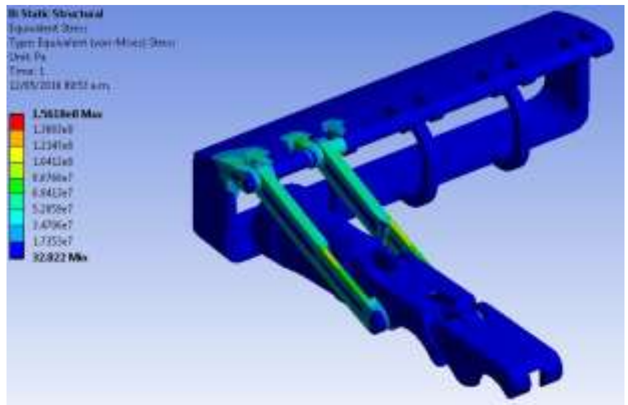

Figure 8 Von Mises Stress analysis

Acknowledgments

This work has been founded by Secretaria de Investigación y Posgrado (SIP) of the Instituto Politécnico Nacional [grant number SIP-20220359].

Conclusions

The proposed six-bar mechanism fulfills the opening and closing conditions of mobility performed by the fingers of the hand, in addition, we ensure with the position equations that there will be no interference between the movements of the bars and their types of joints.

The stress analysis on the design of the mechanism showed that the design of the proposed prototype adequately withstands the stresses, obtaining a minimum-security factor of 1.8. This provides assurance that the design for the phalanges is adequate to support extreme loading conditions.

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Article

Journal of Mechanical Engineering

June, 2022 Vol.6 No.17 7-13

Analysis of the mechanical behavior of the chassis of a vehicle driven by human power through the application of the FEM module of a commercial CAD Software

Análisis del comportamiento mecánico del chasis de un vehículo impulsado por potencia humana mediante la aplicación del módulo MEF de un Software CAD comercial

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DOI: 10.35429/JME.2022.17.6.7.13 Received: March 30, 2022; Accepted: June 20, 2022

Abstract

In this paper, a design of a vehicle called Human Powered Vehicle (HPV) will be developed, which must comply with the competition requirements of the ASME standard, since it has as purpose in the future to participate in the student competition organized by this association (ASME). In order to achieve compliance with the objectives of the vehicle design, a process was followed, in which the first step was to know the requirements that the model to be made must have, in accordance with the standard, that is, both in terms of form and materials to be used. Three different structures were made, which met the requirements of the model. These models were simulated with finite elements to determine their viability, which were subjected to various static loads, among which are loads due to overturning, impacts; Likewise, the fastenings to which the system is exposed were considered, these fastenings were considered of the fixed type. According to the results obtained, the model meets the needs of the ASME standard, in addition to being a light and safe vehicle for the driver.

Vehículo impulsado por potencia humana, Análisis por elemento finito, Simulación

Resumen

En el presente trabajo se desarrollará un diseño de un vehículo denominado Vehículo impulsado por potencia humana (HPV: Human Powered Vehicle, por sus siglas en inglés), el cual debe cumplir con los requerimientos de competición de la norma ASME, ya que se tiene como propósito a futuro participar en la competencia estudiantil organizada por esta asociación (ASME). Para lograr el cumplimiento de los objetivos del diseño del vehículo se siguió un proceso, en el cual como primer paso fue conocer los requerimientos que debe tener el modelo a realizar, de acuerdo con la norma, es decir tanto de forma como de los materiales a utilizar. Se realizaron tres diferentes estructuras, las cuales cumplían con los requisitos del modelo. Dichos modelos se simularon con elementos finitos para determinar la viabilidad de ellos, los cuales se sometieron a diversas cargas estáticas, entre las que se encuentran cargas debidas a volcaduras, a impactos; así mismo se consideraron las sujeciones a las cuales el sistema está expuesto, estas sujeciones se consideraron del tipo fijas. De acuerdo con los resultados obtenidos se tiene que el modelo satisface las necesidades de la norma ASME, además de que es un vehículo ligero y seguro para el conductor.

Human-powered vehicle, Finite element analysis, Simulation

Citation: TORRES-RICO, Luis A., MANRÍQUEZ-PADILLA, Carlos G., SÁNCHEZ-LERMA, Josué R. and VELAZQUEZ-BAUTISTA, Juan E. Analysis of the mechanical behavior of the chassis of a vehicle driven by human power through the application of the FEM module of a commercial CAD Software. Journal of Mechanical Engineering. 2022. 6-17: 7-13

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Introduction

Nowadays, the environmental situation and human needs have led to the development of various alternative vehicle models, in which environmentally friendly energy sources are used as the system's propulsion source. Such is the case of human-powered vehicles. A human-powered vehicle (HPV) is a transport system developed as an intercity mobility alternative for journeys of 30 minutes or less, which aims to reduce emissions of pollutant gases into the atmosphere and consequently reduce the exposure of the population to these pollutants, improving their health and raising their quality of life (Johansson et al. (2017)).

There are some models (HPV), reported in the literature among which are; the vehicle designed by Fegade et al. (2018), which was designed taking into consideration criteria such as the aerodynamics of the chassis and the overall weight of the vehicle, for its part, Knaus et al. (2010) developed an HPV according to both the design criteria requested to participate in the Human Powered Vehicle Challenge (HPVC) competition sponsored by the American Society of Mechanical Engineers (ASME) and to economic, ergonomic and low weight criteria, which resulted in the development of a lightweight, high-speed and easy-to-repair vehicle. Similarly, Choudhury et al. (2012) proposed a human-powered vehicle capable of moving over a variety of uneven terrain by transferring the power generated by the occupant to the transmission system using a biodegradable hydraulic fluid. Most of the aforementioned vehicles were developed using design criteria focused on improving the performance of the vehicle when moving over various types of terrain, however, some safety criteria in case of collisions were not considered.

In this work we aim to design a human-powered three-wheeled vehicle that possesses both performance characteristics similar to other vehicles of the same type reported in the literature and safety criteria to safeguard the integrity of the occupant in the event of a rollover or a lateral collision. To corroborate the mechanical behaviour of the proposed vehicle chassis under the aforementioned loading conditions, a series of simulations based on finite element theory were carried out.

Design criteria

In order to meet the objectives initially set out within the project, a series of design criteria were proposed based both on the rules of the Human Powered Vehicle Challenge (HPVC) competition (American Society of Mechanical Engineers. (2021)) and on geometric and ergonomic criteria based on the anthropometric dimensions of the Mexican population in an age range of 18 to 25 years (Carmenate et al. (2014)).

Performance criteria:

- Braking distance less than 6 m, from a speed of 25 km/h to rest.
- Turning radius less than 8 m.
- Minimum steady speed of 5 km/h.

Safety criteria:

- Minimum distance of 2 in from the chassis to the ground.
- Minimum distance of 3 in between the rider's helmet and the top of the safety cage.
- Design of seat belt harness brackets in accordance with the criteria set out in SAE Baja, "Mini Baja Regulation 2020", (2020).
- Roll cage capable of withstanding a state of loading as described in Figure 1, without permanent deformation or deformation greater than 5.1 cm.

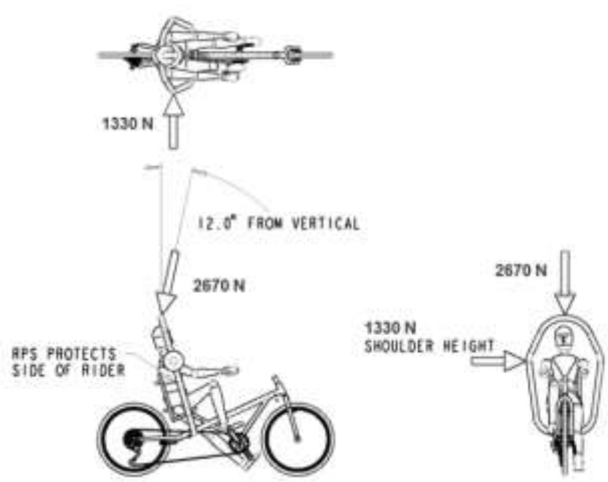

Figure 1. State of applied loads on the chassis
Source: (American Society of Mechanical Engineers (2021)

HPV design description

For the design of the human-powered vehicle proposed in this work, a three-wheeled vehicle model with the "tad-pole" configuration was selected, which, consists of a vehicle equipped with two wheels on the front axle and one wheel on the rear axle (Kosmanis et al. (2014)), as shown in Figure 2.

Figure 2 First prototype of the proposed HPV

The tad-pole configuration allows both to distribute the driver's weight evenly over the vehicle chassis and to provide the vehicle with a rear-wheel drive system and front steering, thus reducing the risk of rollover, improving the vehicle's performance when driving on steep slopes and achieving a minimum stable speed well below 5 km/h.

For the design of the chassis, two types of tubular sections were considered; a 3/4" circular section of schedule 40, and a square section of 7/8" x 7/8" x 1/12". Both tubular sections were commercially manufactured using 6063-T6 aluminium. The first chassis design is shown in Figure 3.

Figure 3 Proposed chassis design

Table 1 shows the chemical characteristics of the selected material in weight % provided by the manufacturer (Aluminium (1100, 1200, 3003, 6026, 6061, 6063, 7075, Ladders), (2022)).

Si	Fe	Cu	Mn	Mg	Cr	Zn	Ti	Otros	Aluminium
0.2-0.6	0.35	0.1	0.1	0.45-0.9	0.1	0.1	0.1	0.15	Rest

Table 1 Chemical characteristics of 6063-T6 aluminium in % weight

Similarly, Table 2 shows the mechanical properties of the material used as part of the boundary conditions of the simulation (Aluminium (1100, 1200, 3003, 6026, 6061, 6063, 7075, Stairs), (2022)).

Material	σ_U. (MPa.)	σ_y (MPa)	Elongation (%)	Hardness HB
Aluminium 6063-T6	245	215	14	75

Table 2 Mechanical properties of 6063-T6 aluminium

In the case of the wheels, transmission system and braking system, the commercial items described in Table 3 were selected.

Component	Quantity
Crankset Shimano RS2000	1
Shimano HG-31 cassette	1
26" rim, Double Fighter 3	2
20" rim, Double Fighter 3	1
Braking system Shimano BR-M6000	1
Brake disc Shimano Deore SM-RT56	2
Shimano shift lever SL-M315-8R	1
Shimano shift lever SL-M315-L	1

Table 3 Commercial HPV components

The seat was designed using a synthetic membrane attached to the chassis by means of a flexible fastening system consisting of coated steel cables.

Mechanical analysis of the structure

In order to analyse the mechanical behaviour of the proposed chassis under side impact and low speed rollover, a series of simulations based on the Finite Element Method (FEM) developed in the FEM module of a commercial CAD software were carried out. For this purpose, the Solidworks Simulation® module developed by Dassault Systèmes® was selected.

ISSN-2531-2189

According to the mechanical properties of the material, a mass of 9.44484 kg was considered for the chassis and taking as a reference the average mass of the Mexican population in an age range of 18 to 25 years, plus a safety factor due to the use of safety equipment and various accessories, a mass of 130 kg was considered for the pilot.

In the case of the side impact simulation, all degrees of freedom of the contact surfaces between the chassis and the tyres were restricted as shown in Figure 4.

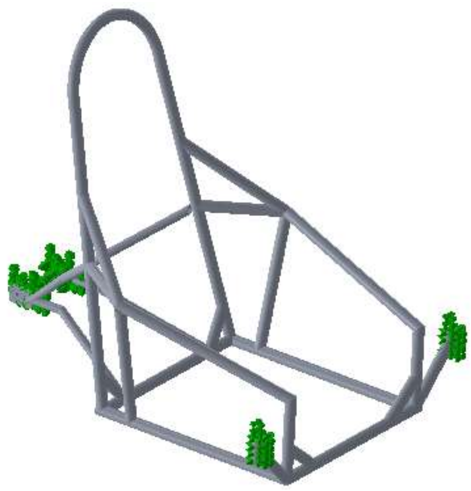

Figure 4 Constraints for side impact simulation

This means that both rotations and displacements of the chassis on the tyre contact surfaces are not allowed. Similarly, for the low-speed rollover simulation, all degrees of freedom of the surfaces where the seat belt supports are located were restricted. The surfaces where the seat belt mounts are located on the chassis are marked in green in Figure 5.

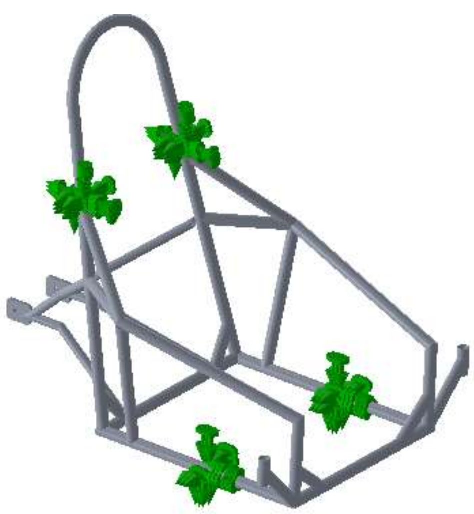

Figure 5 Constraints for low speed rollover simulation

Once the displacement restrictions were defined, the external forces acting on the chassis were introduced, taking as a reference the load state described in Figure 1. For the case of the chassis weight, the gravity force option, g, included in the simulation module was activated. For the weight of the rider, safety equipment and accessories, the total force, W, was distributed equally and applied to the surfaces where the seat supports are located as shown in Figure 6.

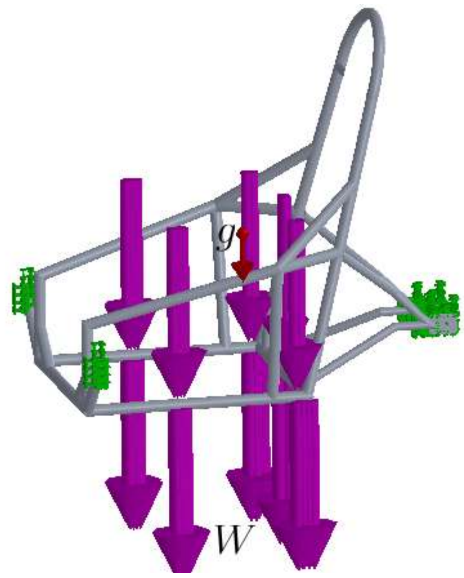

Figure 6 Application of pilot weight, safety equipment and accessories

The loading condition for the side impact simulation consists of the application of a horizontal lateral force, FL, of 1330 N as shown in Figure 7.

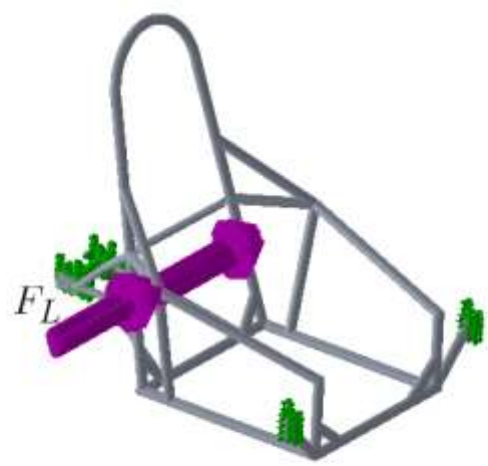

Figure 7 Loading state for the side impact simulation

Figure 8 shows the loading state used for the low speed rollover simulation, where the loading state described in Figure 1 is applied, consisting of a horizontal lateral force, FL, of 1330 N and a force, Fv, applied on the roll cage.

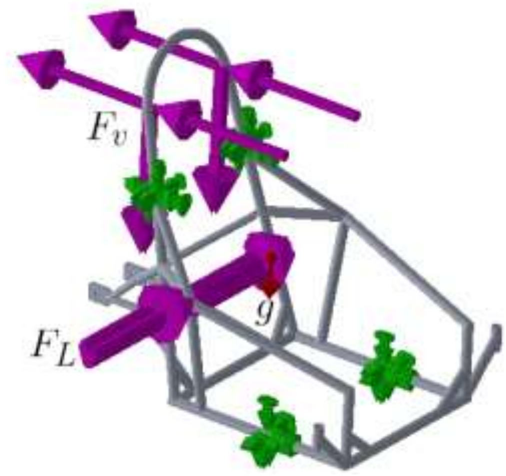

Figure 8 State of loads for the low speed rollover simulation

To generate the chassis mesh, a curvature-based mesh composed of high-order triangular solid elements with a maximum size of 28.9607 mm and a minimum size of 5.79213 mm was implemented. Figure 9 shows the resulting mesh, which consists of 148,003 nodes and 73,371 elements.

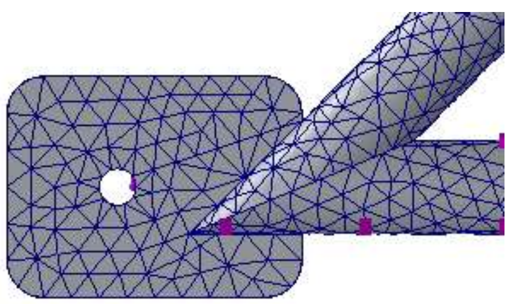

Figure 9 Fragment of the mesh used in the simulation..

Additionally, the option of large displacements and deformations was selected because the geometry analysed is non-linear. The "Direct Sparse" option was used as the solution method, which is recommended for systems with less than 100,000 degrees of freedom (Gómez-López et al. (2013)). As simulation results, the stresses and deformations to which the chassis is subjected under the load states described above were obtained.

Results

When performing the side impact and low-speed rollover simulations, it was found that the chassis design shown in Figure 3 presented deformations of more than 5.1 cm when subjected to a low-speed rollover, so a reinforcement was added to the roll cage as shown in Figure 10.

Figure 10 Final chassis design

With the final chassis design, simulations based on the finite element theory were again carried out considering the load states and displacement constraints described above. Both the maximum allowable deflection constraint and the Von Mises criterion (Von Mises (1913)) were used to analyse the results.

Lateral impact simulation

In the case of the simulation of the mechanical behaviour of the chassis when subjected to a side impact, a maximum deformation of 2.82 mm on the upper arch of the roll cage was observed as shown in Figure 11. This represents 5.52% of the maximum permissible deformation.

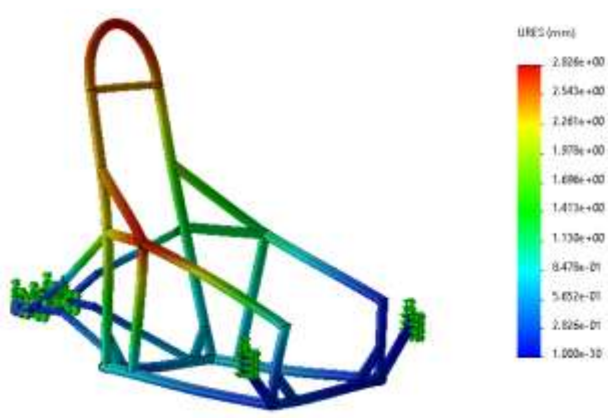

Figure 11 Chassis deformation under side impact

Similarly, Figure 12 shows the stresses to which the chassis is subjected during a side impact, the maximum stress value reached 103.1 MPa which is equivalent to 47.95% of the yield stress of the material.

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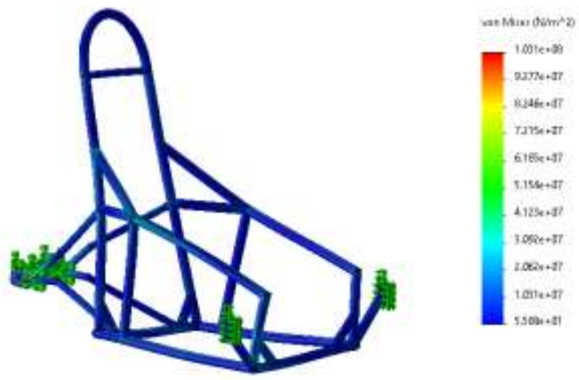

Figure 12 Chassis forces under side impact

Comparing the maximum stresses and deflections with the design criteria, it was found that the vehicle meets the safety criteria in the case of a side impact.

Simulation of low-speed rollover

The results of the simulation of the mechanical behaviour of the chassis in a low-speed rollover show a maximum deformation of 4.528 mm, equivalent to 8.87% of the maximum permissible deformation, as shown in Figure 13.

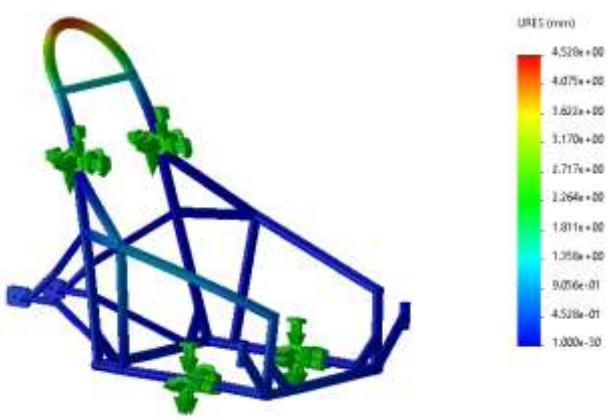

Figure 13 Chassis deformation in a low-speed rollover

The maximum stress shown in Figure 14, with a value of 132.0 MPa, corresponds to 61.39% of the yield stress of 6063-T6 aluminium.

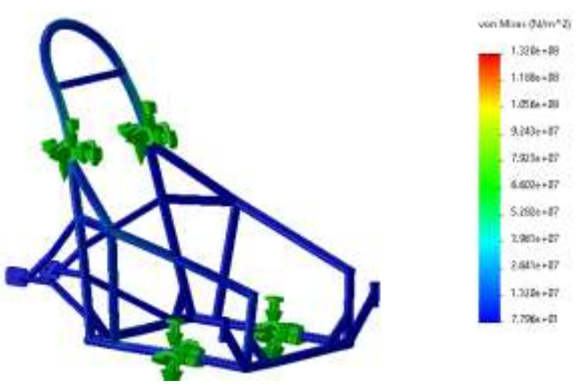

Figure 14 Chassis stress in low-speed rollover

Taking into consideration the values of chassis stresses and deformations, it can be said that the vehicle meets the safety criteria for a low-speed rollover.

Conclusions

This work presented the analysis of the mechanical behaviour of a chassis designed for a human-powered vehicle using a series of simulations based on the finite element theory by applying the Solidworks Simulation® module, which is part of the Solidworks® CAD software developed by Dassault Systèmes®. As a result of the simulations, it was found that the first design proposed for the chassis did not meet the safety criteria by presenting a deformation of more than 5.1 cm in the upper arch of the roll cage when subjected to a state of loads that emulates a low-speed rollover.

Based on the distribution of stresses and deformations, it was proposed to add reinforcement to the roll cage in order to reduce the deformations and alter the stress distribution and thus reduce the magnitude of the maximum stress. By simulating the new chassis design with the proposed material, it was found that it meets the safety criteria when subjected to both side impacts and low-speed rollovers. In both cases the stress magnitude is within the elastic range of the material and the deformations were much smaller than 5.1 cm.

Furthermore, taking as a reference the distribution of stresses and deformations in the chassis, their magnitude and the elastic limit of the material proposed for its manufacture, welding can be considered as a method of joining the various substructures that will form the chassis without significantly affecting the mechanical behaviour of the chassis.

Acknowledgements

This work has been funded by UAQ FONDEC-UAQ 2021 [FIN-2021-30].

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ISSN-2531-2189

Article
Journal of Mechanical Engineering
June, 2022 Vol.6 No.17 14-25

Development of design proposals for a test box for medium soils in direct shear test

Desarrollo de propuestas de diseño de un cajón de ensayos para suelos medianos en un análisis de corte directo

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DOI: 10.35429/JME.2022.17.6.14.25 Received: June 30, 2022; Accepted: December 20, 2022

Abstract

The direct shear test is performed in order to be able to measure the shear strength of soils and rocks in the laboratory. There are laboratories with equipment that performs tests for large materials with which greater reliability is achieved since the soil or granular medium with all its components is considered. The Laboratory of Quality Control and Engineering Consulting, S.C. (CCCI, S.C.) has a direct shear machine for large materials, although it contains both fine and thick materials, the machine is designed for materials with a diameter of 6 inches, therefore, it rules out analysis for materials of 3 inches. inches, that is, medium size. The objective of this research is to propose design alternatives for the test box of the direct shear machine to analyze the resistance parameters for 3-inch materials, which will speed up the shear test process of soils for this type of material. Consequently, obtain reliable data, at the same time, for future research on the mechanical properties of soils.

Direct Shear test, Shear Strength of Soil, Direct Shear Machine

Resumen

El ensayo de corte directo se realiza con motivo de poder medir la resistencia al esfuerzo cortante de los suelos y rocas en el laboratorio. Existen laboratorios con equipos que realizan ensayos para materiales de gran tamaño con lo que se alcanza mayor confiabilidad ya que se considera el suelo o medio granular con todos sus componentes. El laboratorio de Control de Calidad y Consultoría en Ingeniería, S.C. (CCCI, S.C.) tiene una máquina de corte directo para materiales de gran tamaño, a pesar que contiene materiales tanto finos como gruesos, la máquina está diseñada para materiales de diámetro de 6 pulgadas, por lo tanto, descarta análisis para materiales de 3 pulgadas, es decir, de tamaño medio. El objetivo de esta investigación es proponer alternativas de diseño del cajón de ensayos de la máquina de corte directo para analizar los parámetros de resistencia para materiales de 3 pulgadas, lo cual permitirá agilizar el proceso del ensayo de corte de los suelos para este tipo de materiales. En consecuencia, obtener datos fiables, al mismo tiempo, para futuras investigaciones sobre las propiedades mecánicas de los suelos.

Análisis de Corte Directo, Resistencia al Corte de Suelos, Máquina de Corte Directo

Citation: PESCADOR-GUTIÉRREZ, Mariana. Development of design proposals for a test box for medium soils in direct shear test. Journal of Mechanical Engineering. 2022. 6-17: 14-25

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Introduction

The shear strength is the most important aspect for the analysis of the stability of civil works, since it reflects the quality and bearing capacity of shallow and deep foundations, as well as the stability of slopes and the design of retaining walls.

The direct shear test is the most used analysis to perform this type of tests, due to its low cost and efficiency in terms of operation and concrete parameters in shear resistance of soils. On the other hand, this test is performed with standard equipment, which is limited because it does not consider granular media with all its components and the maximum applicable force of 2.5 N (ASTM 3080-03, 2003). This is solved with a larger equipment, since it considers granular media or large, medium and fine particles, with all their components.

In the laboratory of Quality Control and Engineering Consulting, Abasolo Guanajuato, direct shear tests are performed for fine materials with a conventional machine and for 6-inch materials, considered large materials, in a robust machine.

There are tests where the sample size is 3 inches thick, so, performing the cutting test with the giant machine entails the extraction of more material to fill the box, therefore, this process is postponed and laborious. Therefore, the reduction of the box in the direct cutting machine for medium granular media tests optimizes the filling time to 10% of the operating time of the direct cutting machine. Consequently, a precision is obtained in the resistance parameters of the materials to be analyzed, for the study of soil mechanics, as well as for materials that emerge to external environmental changes.

Next, the objectives to achieve the research study are described. Subsequently, the methods to be carried out to develop the design for 3-inch materials are described. Also, the results of the preliminary design proposals for the 3-inch material box and the static analysis for the design evaluation are presented.

Next, conclusions, acknowledgements and references used to support the development of the design proposals for the medium materials testing box.

1. Background

When structures are supported on the ground, they transmit stresses to the soil, which produce deformations in the soil, resulting in catastrophic failures in everyday structures. This can be avoided by knowing the shear strength of the soil.

An important aspect to consider for the correct validation of the study of soils according to geotechnical engineering is the shear stress. This aspect reflects, in addition to the quality of civil works, the bearing capacity of foundations, slope stability, design of retaining walls, among other civil works.

1.1 Shear stress

The shear stress of a soil is defined as the ultimate or maximum resistance that the soil can withstand or resist slip failure in any plane within it, (Das, B. M, 2002). This is determined by various methods: uniaxial compressive strength, direct shear and triaxial compression tests.

The latter are more effective; however, the direct shear test has a greater advantage, since it is a quick and economical analysis.

1. 2 Direct shear test.

The direct shear test consists of containing a portion of the soil to be analyzed.

The test begins when a normal force is applied to the sample container, where the first vertical deformations are recorded (Das, B. M, 2002). Subsequently, a horizontal force is applied to produce the horizontal displacement, so that the shear deformations of the analyzed material are recorded.

The interpretation and evaluation of the test results are left to the consideration or criteria of the engineer or the company requesting the test. Figure 1 below illustrates a test box used to perform direct shear analysis of materials and the general concept of this.

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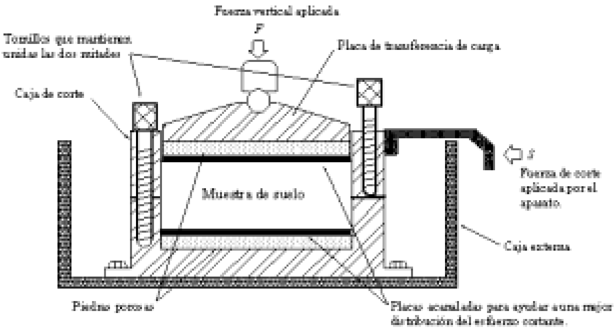

Figure 1 Shear box
(Das, 1997)

This type of direct shear tests are commonly performed by means of conventional equipment (see Figure 2), which have certain limitations, since they are governed by the ASTM 3080-03 standards. These limitations are: of the sample, i.e., fine particles, and the maximum application force, which is 2.5 N.

Figure 2 Direct cutting machine for fine material (CCCI.S.C. Abasolo Gto)

With the advance of mega structures and technological development in civil works, soil samples of different sizes, coarse, medium and fine, must be considered, in addition to all the elements that compose it. As long as the strength parameters of the materials are not reflected in a concrete way in conventional equipment, this is solved with the robust direct cutting machine, giant machine, so called because of the size of the test box.

The laboratory of Control de Calidad y Consultoría en Ingeniería, S.C. (CCCI, S.C.) has a direct cutting machine (Figure 3), developed by Dr. Julio Cesar Leal Vaca and Rafael Eduardo Leal Vaca, partners of the company.

The main features of the machine are: maximum application force of 50 tn and tests for 6-inch materials.

Figure 3 Direct cutting machine for large granular materials (CCCI.S.C, Abasolo Gto).

1.4. Objective

To develop proposals for the design of a test box of a giant direct cutting machine, so that it can perform tests on 3-inch materials.

1.4.1 Specific objectives

– Review the state of the art and investigate the specialized elements in direct shear testing.
– Characterize the current equipment.
– Develop design alternatives to reduce the size of the test box for 3-inch materials using CAD.
– Perform CAE analysis of preliminary proposals when exposed to the maximum test load of 50 tn in a normal plane.
– Perform a design feedback.

Methodology

For the development of the design of the test box, a compilation of information about means that perform direct shear tests for large materials was made. As well as the review of the state of the art and basic training in finite element analysis software.

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Subsequently, field visits were made to the geotechnical analysis laboratory CCCI.S.C in Abasolo, Guanajuato, in order to analyze the direct shear test for large materials, as well as to dimension, characterize and model the current direct shear machine, using the 3D modeling platform SolidWorks and ANSYS for finite element analysis, in order to observe the behavior of the equipment when it is subjected to 50 tn, the maximum load applied during the test.

On the other hand, two alternative proposals are suggested, through SolidWorks 3D, followed by a structural static analysis, to observe the behavior of the caissons when the maximum load is applied.

Finally, perform a feedback to determine if a change of geometry or material thickness is required.

Results

In order to have a controlled sequence in the development of the project, the results shown below are accomplished in stages.

Stage 1.

The first stage consists of the search for available information in the state of the art. Patents of conventional and non-conventional equipment for large-sized materials were also reviewed, with the purpose of characterizing the common parts among the machines and making a design proposal according to the required needs. Three patents are available (Table 1).

Patent	Description
CN102207436B	High performance direct shearing apparatus for large contact surfaces.
CN102607966B	Large contact surface characteristic direct shear apparatus with cycle loading function.
US10197483B2	Tester assembled by multiple sets of mechanisms for shear strength-scale effect of rock joint

Table 1 Active patents of direct cutting machines for large materials

None of these machines operate commercially, since they are designed for research purposes to study the mechanical properties of materials.

Subsequently, visits were made to the soil testing laboratory in order to observe the testing process, as well as to size the direct cutting machine and characterize the equipment (Figure 4 and Figure 5).

Figure 4 Current direct cutting machine, in the process of filling the test box (CCCI.S.C).

Figure 5 Characterization of the equipment (CCCI, S.C.)

ISSN-2531-2189

Stage 2.

The second stage is to perform the current modeling of the machine in order to have a digital visualization of the machine, also, to obtain the design of the drawer or insert for medium materials and to have an overview of the adaptation in the current machine (Figure 6).

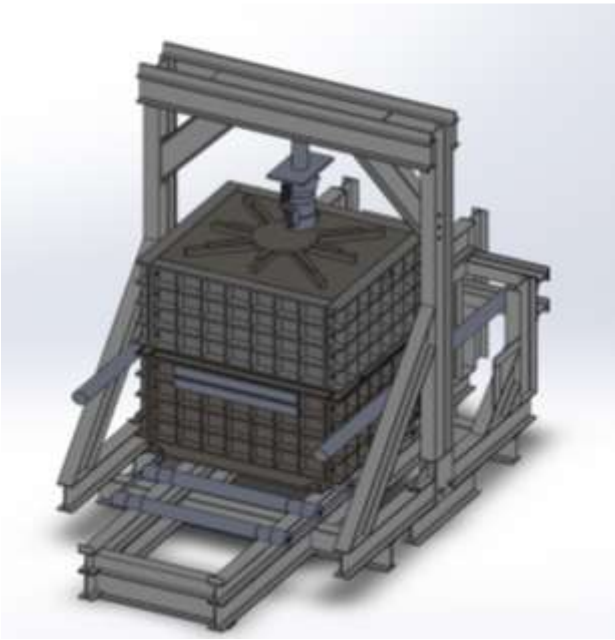

Figura 6 Modelado de la máquina actual de corte directo, CCCI.S.C. *(SolidWorks)*

Stage 3.

The third stage is based on proposing design alternatives to carry out test trials for medium-sized materials.

– Proposal 1.

It consists of a geometry similar to the test box of the direct cutting machine. The box dimensions are 1.1 m X 1.1 m. Additionally, with bolted joints of 1/2" hexagonal screw and welding (see Figure 7). Table 2 shows the main components of proposal 1.

Main components
17 mm thick steel plate ASTM A-6
1" x 2" steel sills

Table 2 Main components of proposal 1

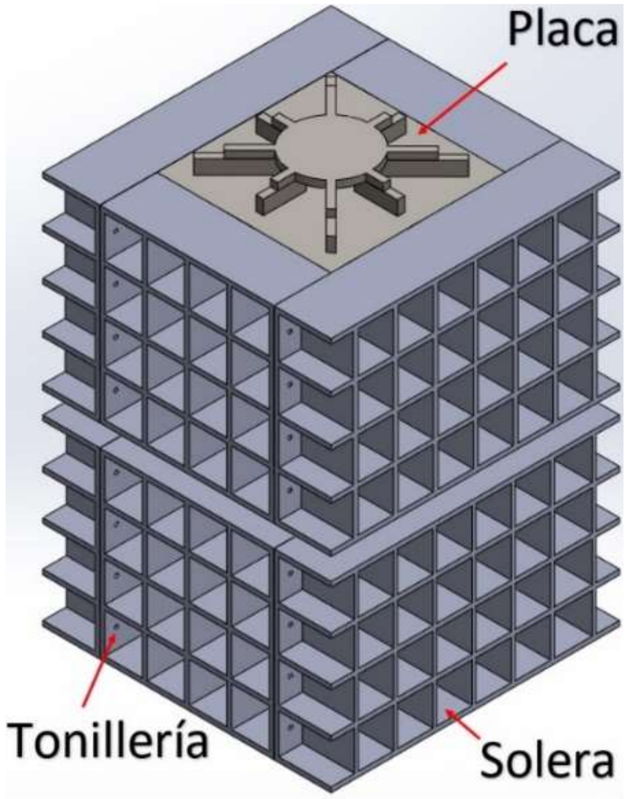

Figure 7. Proposal 1, modeling of the box to perform tests for medium-sized materials *(SolidWorks)*

Next, Proposal 1 is simulated inside the drawer of the current giant direct cutting machine of CCCI.S.C. (Figure 8).

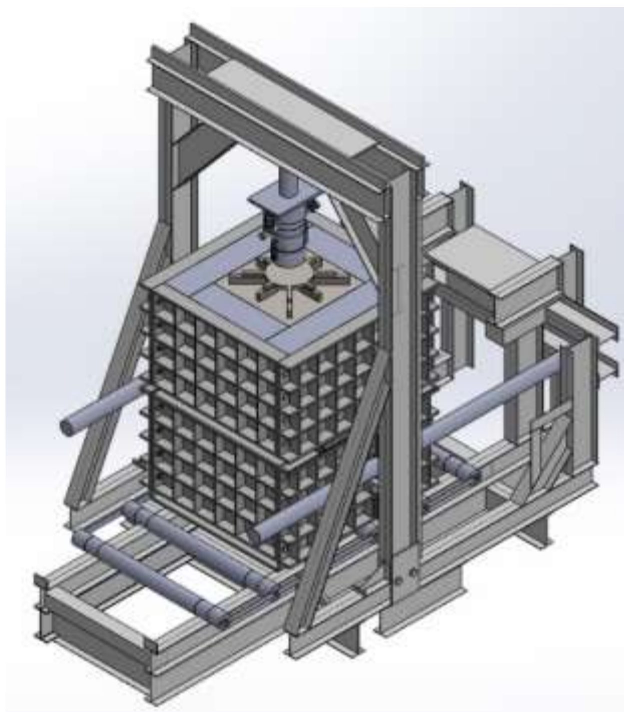

Figure 8 Isometric view of the simulation of the assembly of proposal 1 to the drawer of the current CCCI.S.C. machine *(SolidWorks)*

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– Proposal 2.

To reduce the size of the box for 3-inch materials, it is suggested to reduce the drawer space. In the upper part (Figure 9), an 8" x 10" rectangular profile lid is designed to reduce the drawer space. Taking as a reference the current lid of the external drawer, the upper part of the adaptation is made. Likewise, in the lateral parts, the 6" x 6" PTRs serve as posts for a 2" steel plate, with the same purpose of the profiles, to reduce the size of the external box. Table 3 shows the main components of proposal 1.

Main components
Structural PTR 8" X 10" CAL. 3/8
Structural PTR 6" x 6" CAL. 3/8
1" x 2" screed
1" and 2" steel plate

Table 3 Main components of the proposal 2

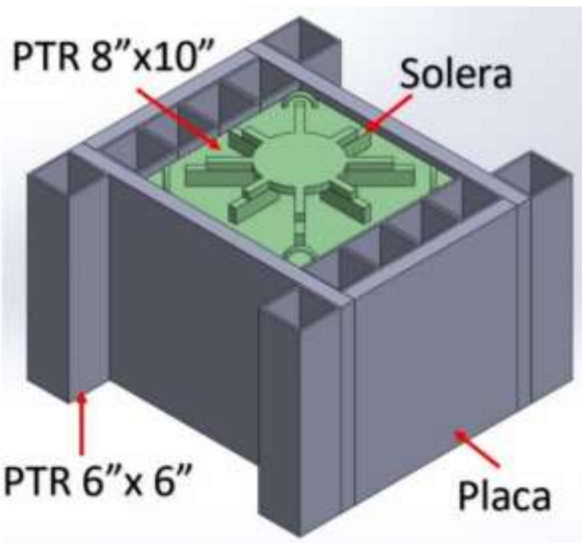

Figure 9 Drawer top design *(SolidWorks)*

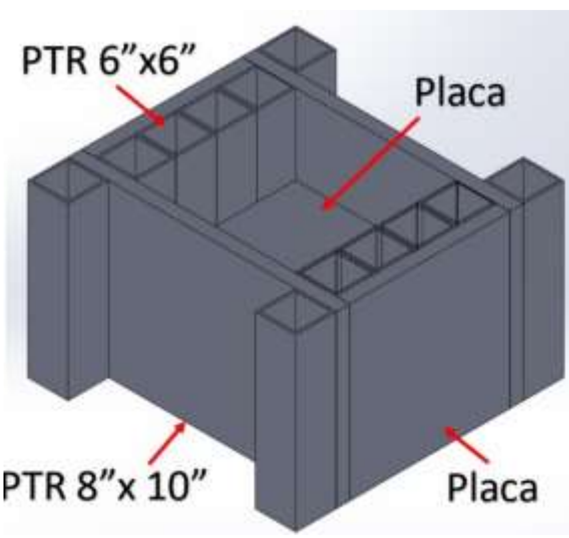

Figure 10 Drawer bottom design *(SolidWorks)*

In the bottom part (Figure 10), a base of 6 x 6-inch square profiles is made, plus square PTR used as posts to support a 2-inch steel plate, in order to reduce the lateral space of the box

Next, Proposal 2, in green color, is simulated inside the box of the current giant direct cutting machine of the company CCCI.S.C. (Figure 11).

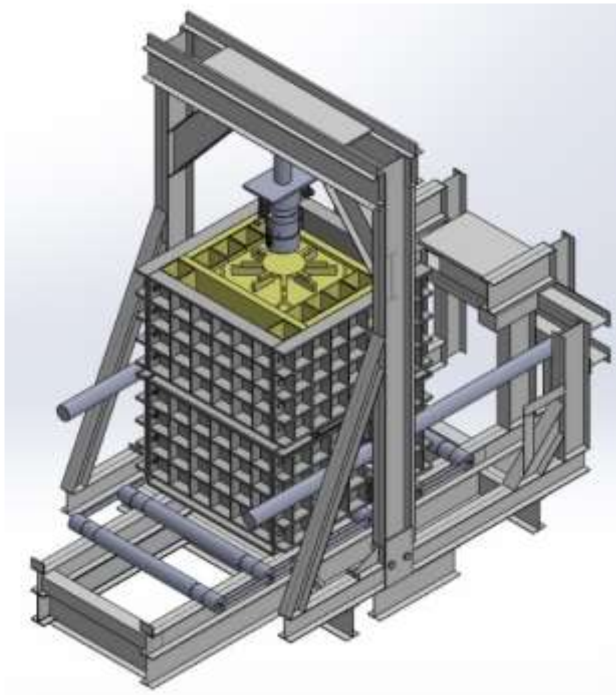

Figure 11 Isometric view of the simulation of proposal 2 to the drawer of the direct cutting machine of the company CCCI.S.C. (SolidWorks).

Stage 4.

In this stage, a simulation of the total deformation, the von-Mises stress and the safety factor of the proposals was carried out by means of CAE, in order to analyze the behavior of the structures when subjected to the maximum load applicable during the test and finally, to select the optimum design for the direct shear tests for 3-inch materials.

– Proposal 1.

Table 4 shows the results of the parameters previously mentioned in Proposal 1. The results indicate that the design has a maximum total deformation of 0.24202 mm (see Figure 12), a maximum tensile stress of 138.41 MPa (see Figure 13) and a minimum safety factor of 1.8062 (see Figure 14).

ISSN-2531-2189

Results	Mín.	Máx.	Units	Time (s)
Deformatio Total	0	0.24802	mm	1
Von-Mises stress	4.648 e-003	138.41	MPa	1
Safety factor	1.8062	15	-	1

Table 4 Summary of the results of the analysis of proposal 1

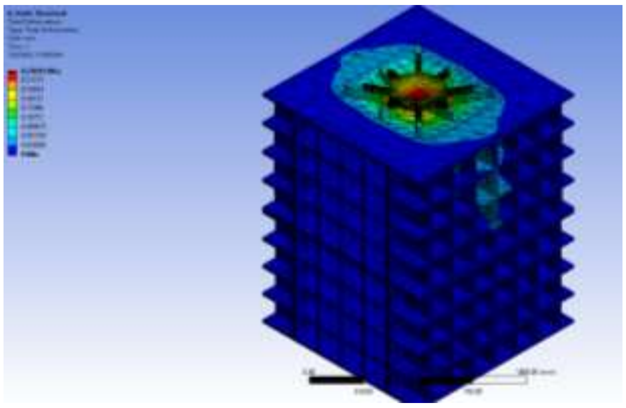

Figure 12. Total deformation of proposal 1 at a maximum load of 50 tn.

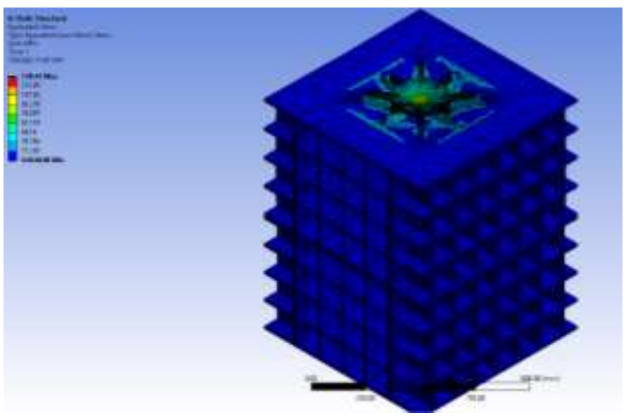

Figure 13. von -Mises stress of proposal 1.

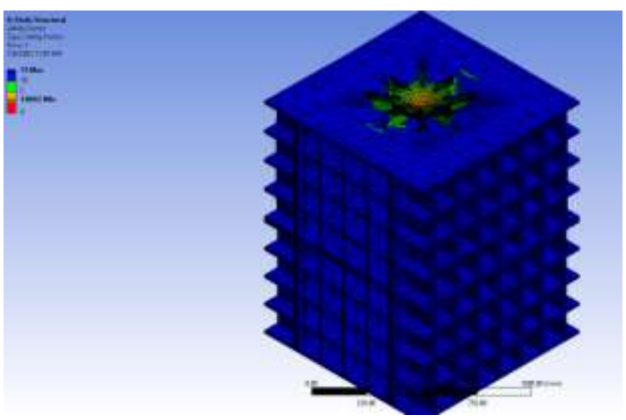

Figure 14. Factor of safety of proposal 1.

– **Proposal 2.**

Table 5 shows the results of the above mentioned variables of the insert. The matrix indicates that the structure in proposal 2 has a maximum total deformation of 0.1412 mm (see Figure 15), a maximum tensile stress of 192.23 MPa (Figure 16) and a minimum factor of safety of 1.3005 (Figure 17).

Results	Mín.	Máx.	Units	Time (s)
Deformatio Total	0	0.1412	Mm	1
Von-Mises stress	3.4904E-003	192.23	MPa	1
Safety factor	1.3005	15	-	1

Table 5 Summary of the results of the analysis of proposal 2

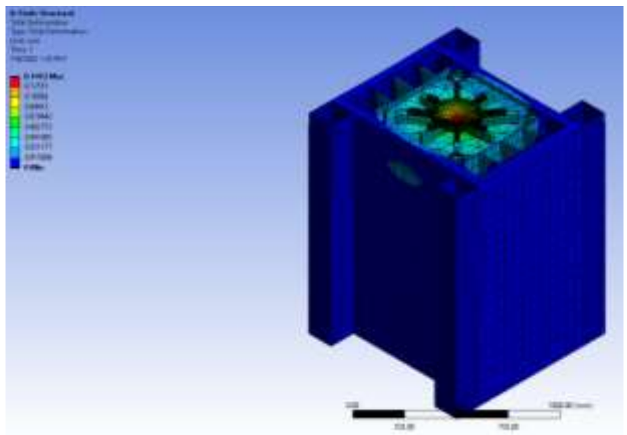

Figure 15. Total deformation of the insert in the 1.20 m x 1.20 m machine test box (ANSYS).

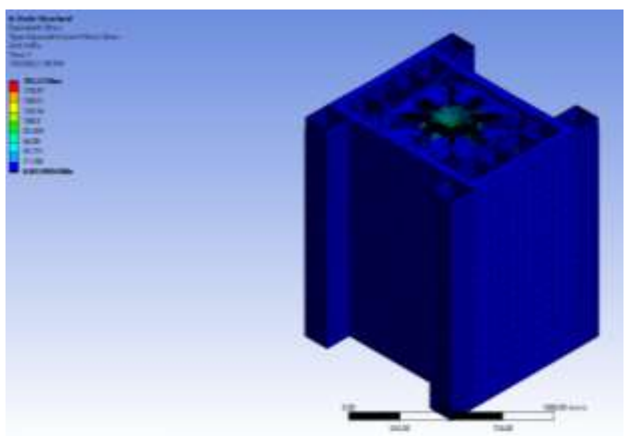

Figure 16. Von - Mises stress of the insert in the test box of the 1.20 m x 1.20 m machine.

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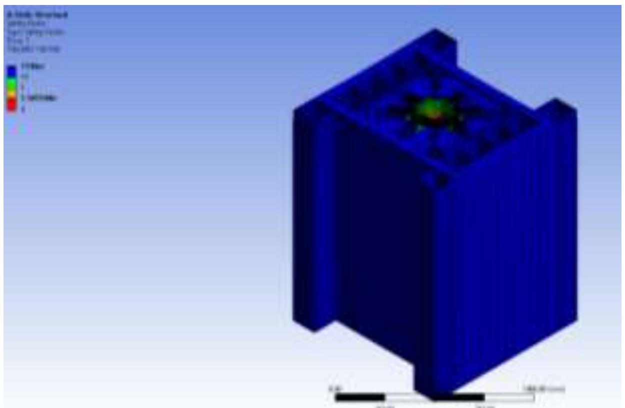

Figure 17. Factor of safety of the insert in the test box of the 1.20 m x 1.20 m machine

Acknowledgment

I would like to thank the engineer Rafael Leal Vaca, the engineer Román Landeros, and all their collaborators for their support during my stay at the CCCI, S.C. laboratory, as well as for providing me with the necessary tools to carry out this research.

Conclusions

According to the results obtained, two preliminary proposals are obtained: a test box with the same geometry as the test box for large-sized materials and a coupling composed of several assemblable parts.

According to the results of the simulations, both proposals are suitable for the practice of direct shear tests for medium-sized materials, since, in the maximum total deformation, in both, does not exceed 1 mm; in addition, the minimum safety factor is greater than 1, that is, they are designed to support exactly what is needed during their use, since, the more unnecessary safety factor, the more costly and less competitive they are. Additionally, it allows us to know if it is feasible to reduce the thickness of the material, therefore, it translates into manufacturing and manufacturing costs of the equipment.

However, it is contemplated the possibility of rethinking the design of the test box for three-inch materials, due to the progressive physical conditions to the machine, regarding the behavior of the test box of the giant machine during the testing process.

Therefore, it is suggested to consider current mechanical aspects of the machine, to make a punctual analysis, both the basic engineering and the regulations on which the giant machine is based, in order to determine the adequate design for the manufacturing, experimentation and approval of the box. Consequently, to reduce the time in the testing process for medium-sized materials, as well as for future research on soil mechanics.

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Article **Journal of Mechanical Engineering**
June, 2022 Vol.6 No.17 26-32

Evaluation of modal frequencies obtained with the impact hammer technique on an epoxy matrix composite material reinforced with glass fibers

Evaluación de las frecuencias modales obtenidas con la técnica del martillo de impacto en un material compuesto de matriz epóxica reforzado con fibras de vidrio

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DOI: 10.35429/JME.2022.17.6.26.32 Received: March 30, 2022; Accepted: June 20, 2022

Abstract

A numerical-experimental methodology is presented to obtain the modal frequencies of polymeric composite materials reinforced with unidirectional fibers (glass fiber and epoxy resin) for possible aeronautical applications. The objective of this study is to compare the behavior of an isotropic material with an orthotropic one. This comparison is to observe the influence of the material properties on its performance under dynamic conditions, where the modal frequencies of a material can directly affect the performance of each element of a structure. The first case describes the numerical and experimental identification of the modal frequencies of an isotropic material (6065 T5 aluminum). The second case study is presented to show how this methodology is adapted to the composite material. The experimental results are obtained by applying the impact hammer testing method. The comparison provides new insights into the modal behavior of vibrations in composite materials. A significant finding of this work is to provide a detailed analysis of the behavior of a unidirectional composite material in terms of the fiber's orientation. Then, this work would be established the fundamentals of the composite material performance for rotative elements applications.

Modal frequencies, Composite materials, Modal analysis

Resumen

Se presenta una metodología numérico-experimental para obtener las frecuencias modales de materiales compuestos poliméricos reforzados con fibras unidireccionales (fibra de vidrio y resina epoxi) y evaluar su comportamiento en componentes aeronáuticos. El objetivo de este estudio es comparar el comportamiento de un metálico con uno material compuesto. Esta se realiza para observar la influencia de las propiedades del material en su respuesta en condiciones dinámicas, donde las frecuencias modales de un material pueden afectar directamente el funcionamiento de un elemento en una estructura. El primer caso fue identificar numérica y experimentalmente las frecuencias modales de un material isotrópico (aluminio 6065 T5), mientras que el segundo caso de estudio fue replicar la metodología en el material compuesto. Los resultados se obtuvieron a partir del ensayo modal con martillo de impacto y un modelo de elementos finitos. La comparación proporciona nuevos conocimientos sobre el comportamiento modal de las vibraciones en materiales compuestos. Un hallazgo significativo de este trabajo es proporcionar un análisis detallado de los comportamientos de un material compuesto unidireccional en términos de la orientación. Por lo tanto, trabajo, fundamenta las bases del desempeño de los materiales compuestos en elementos rotativos.

Frecuencias modales, Materiales compuestos, Análisis modal

Citation: COCA-GONZALEZ, Juan Manuel, AVILA-HERNÁNDEZ, Sergio Albano, REYES-SOLÍS Alberto and TORRES-CEDILLO, Sergio Guillermo. Evaluation of modal frequencies obtained with the impact hammer technique on an epoxy matrix composite material reinforced with glass fibers. Journal of Mechanical Engineering. 2022. 6-17: 26-32

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Article
Journal of Mechanical Engineering
June, 2022 Vol.6 No.17 26-32

Introduction

The main advantage of substituting metals by composite materials is observed in the final weight of the elements, for example, Titanium alloys or Aluminum have densities of 4500 kg/m3 and 2700 kg/m^3 respectively, while glass or carbon fiber reinforced composites vary between 1400 and 2600 kg/m^3 [I]. As a consequence, air vehicles require less fuel for their operation and, in general, less economic resources to perform the take-off and landing cycles, which makes an aircraft whose constitution is mostly made of composite materials more efficient. An example is the latest models of Boeing and Airbus, the 787 and A350, where practically 50% of the total weight of the structure is made of carbon fiber reinforced plastic and various composite materials [II]. This is a tangible issue addressed with the implementation of new materials.

However, the analysis of the dynamic behavior of a composite material involves an increase in its complexity due to its orthotropic nature. In general, a composite material is composed of two elements, the matrix and the reinforcement, the former being continuous and the latter dispersed. If it is required to manufacture specific components with high structural performance, polymer matrix composites with long continuous fibers are used. For their manufacture, an arrangement of fibers is usually made on a resin layer whose thickness is less than one millimeter. From the above, we can define the matrix as an organic material that can be solid or glassy, where its main function is to keep the fibers together and in position, it also provides protection from the environment, provides a shape, gives ductility and transmits loads uniformly throughout the composite. On the other hand, the reinforcement mainly supports the loads applied to the material as a consequence of which it provides stiffness and strength [III] [IV] [V].

The study of dynamics in orthotropic materials is in development, since its implementation in industries such as aeronautics is becoming more and more common. Therefore, covering all possible combinations of operating and manufacturing conditions is essential to ensure the reliability of components made of composite materials.

The analysis can be approached with different techniques and theories: causing an external excitation to the element under study and finding its modal frequencies through the output signals [VI], considering the effect of the variation of in-plane loads and the stacking sequence in plates [VII], subjecting thin-walled beams to deformations and initial stresses [VIII], taking into account the effect of normal stresses in a laminate [IX] or the influence of factors such as the number of layers, thickness and orientation of the fibers of each one [X].

The focus of this work is to obtain the modal frequencies ω_i of two elements similar in dimensions, the first one made of Aluminum 6065 T5 and the second one of E-type fiberglass and epoxy resin.

The results are obtained theoretically, numerically and experimentally using the impact hammer method. The above in the metallic element to define the most appropriate methodology for each case.

From the results obtained in the metallic element, it is replicated in the finite element model, feeding it back with the mechanical properties of the composite material obtained from the physical characterization of this [XI]. The results of both cases are compared to describe the behavior and from the graphs obtained experimentally, hypotheses of the phenomenon of vibrations in composite materials are proposed.

The added value of the work is the experimental methodology and the analysis of the graphs of the vibratory response of the material.

Methodology

Since the behavior of composite materials under dynamic conditions is not consistent and predictable, it is necessary to have first a theoretical and experimental basis from which to start, in this case to analyze an isotropic material (Aluminum 6063 T5), there are a large number of studies, methodologies and formulations to predict the way in which materials of this nature react [XII].

ISSN-2531-2189

One of the first objectives is to obtain an experimental methodology applied to the metallic material where the experimental results coincide or do not present relevant discrepancies with the results obtained analytically and numerically. The analysis condition will be beam-type elements with free-free boundary conditions, *i.e.*, there is no restriction of movement in any direction along the beam.

A simple metallic geometry is chosen to start the analysis, a 12.7 x 1.58 x 300 millimeters 6063 T5 aluminum slab section is used. The first five natural frequencies of this element are obtained analytically using formulas 1 and 2. Two equations from different authors are used to compare the analytical results [XIII] [XIV].

$$\omega_i = \frac{\lambda_i^2}{2\pi L^2}\sqrt{\frac{EI}{m}} \quad (1)$$

$$\omega_i = \frac{K_n}{2\pi}\sqrt{\frac{EIg}{wL^4}} \quad (2)$$

Where:
ω_i is the natural frequency, i=1,2,3,5....
E is the Young's modulus of the material..
I is the moment of inertia of the cross-sectional area of the element.
g is the acceleration of gravity.
L is the length of the element.
m is the self-weight of the material with respect to the element, $m = density * area$
$w = L * density * area * g$
λ_i constant dependent on boundary conditions and mode of vibration..
K_n constant dependent on boundary conditions and mode of vibration.

Once the values of the natural frequencies are obtained, the finite element software Abaqus CAE is used as a second validation method. The way in which the solution is reached by this method is based on dividing the model into smaller elements, this by meshing, by increasing the number of elements that make up the original element, the results are usually better approximated to the real phenomenon, however, it is not always the appropriate technique to reach the solution, in addition to the fact that increasing the number of elements consumes more computational memory.

In the finite element method it is possible to control the degree of solution used by each element, this is known as shape function, which allows defining the element in a linear or quadratic way. This function tries to resemble the behavior of an object of study after being subjected to the indicated stresses. The consequence of increasing the degree of the shape function is greater accuracy in the final results [XV].

Subsequently, experimental tests are performed to the element, the experiments are carried out with an impact hammer that has a sensor capable of measuring the force with which the element is hit, the blow generates a measurable vibratory response from an accelerometer that is attached to a specific point of the element, the data generated are collected through an array created in the LabVIEW® software. Finally, with the help of Matlab software, Frequency-Response Function (FRF) graphs of the material are generated and from the analysis of these it is possible to obtain the natural frequencies [XVI].

In general, the experimental procedure consists of:

- Recreate the boundary conditions to be studied.
- Place an accelerometer in a specific point previously defined in the element.
- Hit the element with the hammer at specific points.
- Collect displacement and force data for each test.
- Perform data processing.
- Generate the Frequency-Response Function (FRF) of the element.
- Analyze the FRF plot to obtain the natural frequencies.

Figure 1 shows the general scheme of an impact hammer test, where two signals are used, an input signal which is the excitation caused by the impact hammer and an output signal, being the vibratory response of the element obtained through the displacement of the accelerometer. Both are measured with respect to the time elapsed in the test, collecting the data in real time. Impact force data are obtained in Newtons and displacement data in micrometers.

ISSN-2531-2189

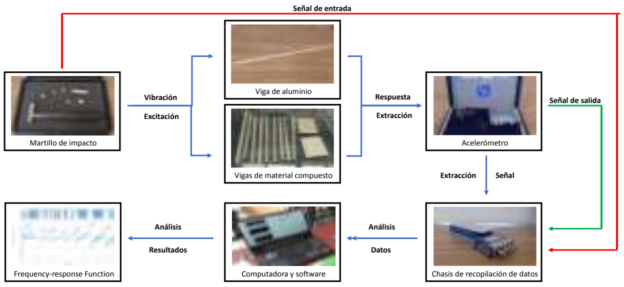


Figure 1 General scheme of the experimental impact hammer test

In the experimental tests, the boundary conditions are simulated by placing the element on a granite table to avoid as much as possible the influence of external elements on the accelerometer readings, at the same time, the beam is supported on sponges with the lowest possible density to avoid restrictions in its displacements in any direction. The accelerometer is attached to one end of the element (Figure 1). Subsequently, a vibratory response is generated by means of blows with the impact hammer at previously defined points of the element (Figure 2).

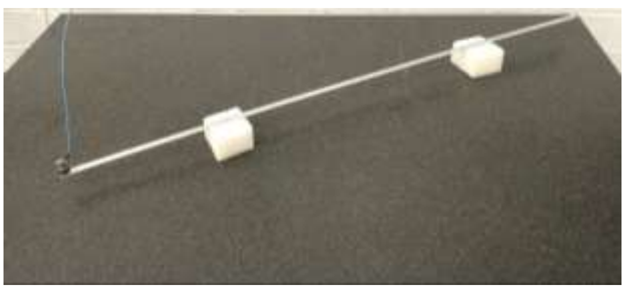


Figure 2 General configuration of the assay

Figure 3 Excitation of the element by impact hammer

Once the tests were completed, composite specimens (epoxy resin reinforced with E-type glass fibers) were fabricated, replicating as far as possible the dimensions of the aluminum element by means of the resin infusion process [XVII].

The properties of Aluminum 6065 T5, E-type glass fiber and EPOLAM 2015 epoxy resin used in this work are presented in Tables 1, 2 and 3 [XVIII].

Properties	
ρ (gr/cm^3)	2.7
E (MPa)	70000
G (MPa)	26300
ν	0.33
σ_C (MPa)	110

Table 1 Mechanical properties Aluminum 6065 T5

Properties	
ρ (gr/cm^3)	2.58
E_1 (MPa)	74000
G_{12} (MPa)	30000
ν_{12}	0.25

Table 2 Mechanical properties E-glass fiber

Properties	
ρ (gr/cm^3)	1.12
E_1 (MPa)	2420
ν	0.3
σ_{Fm} (MPa)	120
σ_{Tm} (MPa)	70

Table 3 Mechanical properties Epoxy resin EPOLAM 2015

Results

As first results we compared the values of the analytical frequencies using formulas 1 and 2 with those obtained in the Abaqus software in its linear and quadratic form functions. The data are presented in Table 4.

Modal frequency	Analytical value 1	Analytical value 2	Linear Abaqus	Quadratic Abaqus
1	92.09	91.99	86.79	91.99
2	253.68	253.57	239.28	253.61
3	497.50	497.11	469.19	497.28
4	822.32	821.74	775.83	822.25
5	1229.36	1227.54	1159.40	1228.70

Table 4 Analytical and numerical modal frequencies in Hertz

Analyzing the obtained values, it becomes evident that the quadratic analysis in the finite element software practically eliminates the margin of error with respect to the analytical results.

Figure 2 compares the modal frequency 5 using the quadratic shape function. A total of 45000 elements were generated for the model, replicating the geometry of the Aluminum sill section.

ISSN-2531-2189

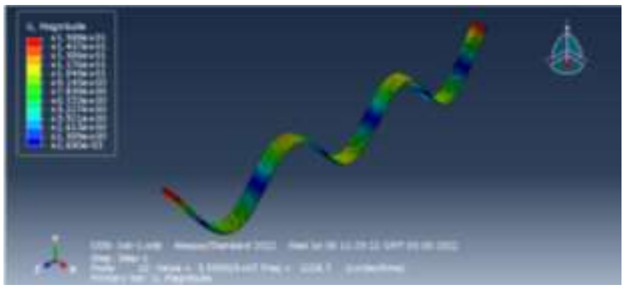


Figure 4 Fifth modal frequency with value 1228.7 Hz.

Seventy experiments were carried out, 42 of which were successful. From the total number of samples, the values of the first five natural frequencies were obtained, making an average to compare with the analytical and numerical results, which are presented in Table 5.

Modal frequency	Experimental values
1	75.21
2	201.65
3	538.11
4	890.2
5	1231.32

Table 5 Experimental modal frequencies in Hertz.

After the analysis of the aluminum element and the fabrication of the composite specimens, a physical characterization of the material is carried out in order to calculate its mechanical properties from the rule of mixtures [XI]. The properties shown in Table 6 are obtained.

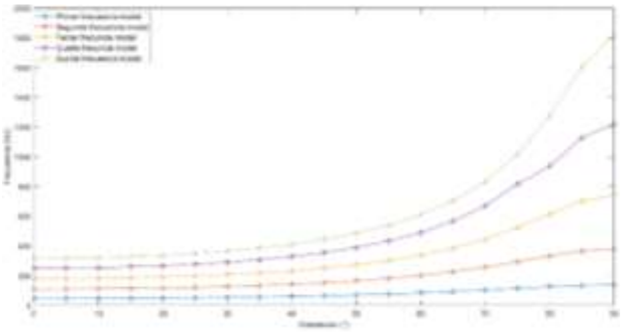
Properties	
ρ (gr/cm^3)	2.009
E_1 (MPa)	47501.74
E_2 (MPa)	6684.54
ν_{12}	0.26
ν_{21}	0.036
G_{12} (MPa)	6169.02
G_{23} (MPa)	3.22

Table 6 Mechanical properties of composite material

Due to the complexity of the manufacturing process and the lack of control of dimensional parameters such as thickness during the manufacturing process, fiberglass specimens with dimensions of 14.54x2.48x300 millimeters were obtained. With an average layer thickness of 0.031 millimeters.

With the values in Table 6 and the dimensions of the specimens, the finite element model is fed back.

It is considered that the composite beams have 8 stacked layers whose orientations are equal in each one and of constant thickness, in order to analyze the most fundamental behavior of the material. Simulations are performed at different orientations to observe the behavior of the modal frequencies with respect to the orientation of the unidirectional fibers in a composite.



Graph 1 Values of the first five modal frequencies in a composite material with respect to the orientation of its unidirectional fibers

From graph 1, it is evident that, as the orientation of the fibers increases, so do the values of the modal frequencies, in the first instance this is due to the increase in the stiffness of the material when the fibers are aligned to its longitudinal axis.

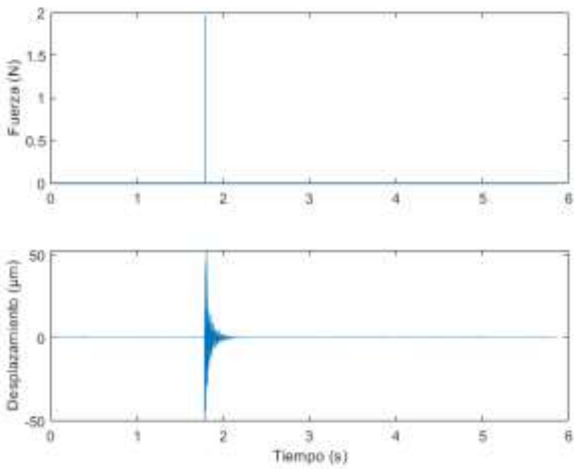
However, since there were discrepancies in the dimensions, it is relevant for the study to perform simulations of the elements of each material with the same dimensions. We will use those obtained in the specimens (14.54x2.48x300 millimeters) to perform the analyses. The results are presented in Table 7.

Modal frequency	Aluminum 6065 T5	90° composite material
1	143.71	137.65
2	396.13	379.04
3	776.58	741.91
4	1283.73	1223.8
5	1917.67	1823.02

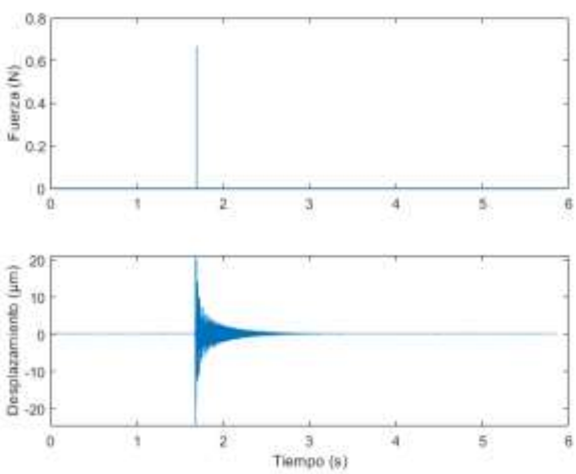
Table 7 Values of modal frequencies in elements of equal dimensions of aluminum and composite material in Hertz

During data processing, two graphs are obtained, the force of the hammer blow and the displacement of the accelerometer, both with respect to time. In metallic elements the displacement is constant and homogeneous, while in the composite material another phenomenon occurs (graphs 2 and 3).

When observing graphs 2 and 3 it is noticeable that, in spite of being the same material, the vibratory response differs greatly in one case with respect to the other. While the specimens with fibers oriented at 45° show a very fast energy dissipation, the 90° orientation requires more time to return to the resting state. This may be due to the individual damping ratio of the matrix and the reinforcement, as well as the significant influence of the material stiffness on the response, which is more relevant than the element dimensions.



Graph 2 Excitation force and element displacements with respect to time in glass fiber specimens with their fibers oriented at 45°



Graph 3 Excitation force and element displacements with respect to time in glass fiber specimens with their fibers oriented at 90°

Conclusions

Since the experimental, theoretical and numerical results of the metallic material tests do not present very high error ranges, it is possible to assure that the methodology can be replicated in composite materials.

The use of the quadratic shape function for the solution of the finite element model of the composite material component is recommended, since it provides results that are closer to the real phenomenon.

It was observed that, in unidirectional composite beams, the change in fiber orientation has a direct impact on the modal frequencies of the beams. Therefore, it is feasible to expand the study to other elements such as plates or shafts.

It is also observed that in aluminum and FRP components with the same dimensions, the first 5 modal frequencies are considerably close. This is of interest since theoretically two elements identical in dimension would have the same dynamic behavior.

A particular phenomenon was detected during the experimentation. This suggests further analysis of the vibrational responses of the composite material. Therefore, it is proposed to calculate the damping ratio present in the composite material, which is mainly attributed to the resin, since it has a more ductile behavior than the reinforcement.

Acknowledgments

We are grateful to the UNAM-PAPIIT projects IN113921 and IN118820 for the computing resources used for this article. We also thank CONACYT for the support through the national scholarship program and the researcher system (SNI).

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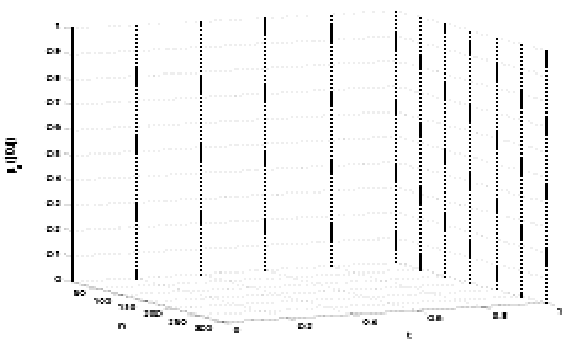

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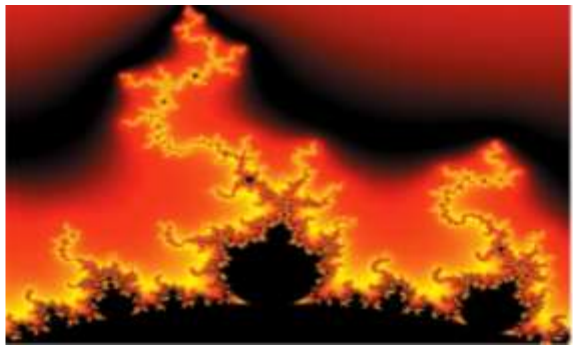

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