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VIBRATION CONTROL OF A HELICOPTER ROTOR BLADE BY EMPLOYING A SUPERELASTIC NITI SMA PITCH LINK

Andersson Guimarães Oliveira

JOÃO PESSOA – PARAIBA

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ANDERSSON GUIMARÃES OLIVEIRA

CONTROLE DE VIBRAÇÃO EM PÁS DE ROTOR DE HELICÓPTERO COM *PITCH LINK* SUPERELÁSTICO DE LMF NiTi

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Advisor: Prof. Dr. Marcelo Cavalcanti Rodrigues

Co-advisor: Professor Dr. Antonio Almeida Silva

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por



ANDERSSON GUIMARÃES OLIVEIRA

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Prof. Dr. MARCELO CAVALCANTI RODRIGUES

Orientador – UFPB



Prof. Dr. ABEL CAVALCANTE LIMA FILHO

Examinador Interno - UFPB



Prof. Dr. RODINEI MEDEIROS GOMES

Examinador Interno - UFPB



Prof. Dr. ANTONIO ALMEIDA SILVA

Examinador Externo – UFCG



Prof. Dr. CARLOS JOSÉ DE ARAUJO

Examinador Externo



Prof. Dr. RÔMULO PIERRE BATISTA DOS REIS

Examinador Externo – UFERSA

Assinado de forma digital por ROMULO
PIERRE BATISTA DOS REIS:02351281470
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CANDIDATE BIOGRAPHY

Andersson Guimarães Oliveira was born in Campina Grande, Paraíba, Brazil, in 1982. Electronic and Telecommunication technician (integrated into high school) in 2000 from Redentorista Technical School. Developed professional experience in electronic and instrumentation. Received the bachelor's degree in Mechanical Engineering in 2008 from Federal University of Campina Grande (UFCG), Campina Grande, Brazil. Received Master's degree in Mechanical Engineer in 2018 from Federal University of Campina Grande. He is Assistant Professor at Federal University of Alagoas – UFAL.



Research fields: Helicopter Dynamics, Shape Memory Alloys; Instrumentation

Author' ID's

ORCID: <https://orcid.org/0000-0002-6472-7941>

Lattes ID: <http://lattes.cnpq.br/0488725741611424>

LinkedIn: <https://www.linkedin.com/in/andersson-oliveira-99489bb8/>

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“Work as if you were never die. Pray as if you were to die today”

Priest Cicero (religious leader that lived in Brazil)

CONTROLE DE VIBRAÇÃO EM PÁS DE ROTOR DE HELICÓPTERO COM PITCH LINK SUPERELÁSTICO DE LMF NiTi

RESUMO

Problemas de vibração em helicópteros são uma preocupação de fabricantes e usuários da indústria aeronáutica. Esta questão representa um desafio, principalmente no controle de vibração e caracterização dinâmica, pois as medições de vibração são difíceis de serem realizadas, principalmente em partes rotativas de helicópteros. Com materiais inteligentes, a indústria busca soluções que contribuam para a redução da vibração sem gerar impacto negativo no peso e na complexidade do sistema. Este estudo projetou, produziu e testou um dispositivo de Liga com Memória de Forma (LMF) que substitui a haste rígida destinada à variação do ângulo de passo da hélice, denominada *pitch link*, em um protótipo de rotor de helicóptero para reduzir a vibração na frequência crítica do rotor empregando amortecimento passivo histérico, sem consumo de energia da máquina. O estudo emprega um protótipo de helicóptero de pequeno porte para uso em laboratório, com duas pás para realizar a identificação da frequência de operação crítica no mecanismo original para estabelecer uma condição de referência. Em sequência o dispositivo de LMF substituiu o dispositivo rígido para realizar uma análise comparativa da resposta dinâmica. Após uma bateria de testes, com ensaios de longa duração usando mini molas helicoidais LMF abertas e fechadas, os resultados demonstraram que o dispositivo apresentou o melhor desempenho com uso de molas helicoidais do tipo fechada. Os resultados com o dispositivo de LMF demonstraram redução nas amplitudes de vibração em cerca de 2dB, chegando a 4dB em condições de teste severas. Em condições de operação essa redução representaria aumento de conforto para tripulação, melhoria de performance da máquina e redução dos desgastes dos sistemas mecânicos, decorrente da vibração.

Palavras chave: controle de vibrações, ligas com memória de forma, dinâmica de helicópteros, dinâmica de rotores, *pitch link*

VIBRATION CONTROL OF A HELICOPTER ROTOR BLADE BY EMPLOYING A SUPERELASTIC NITI SMA PITCH LINK

ABSTRACT

Vibration problems in helicopters are a concern of manufacturers and users in the aeronautical industry. This issue represents a challenge, mainly in the vibration control and dynamic characterization because the vibration measurements are difficult to perform, especially on helicopter rotating parts. With smart materials, the industry seeks solutions that contribute to vibration reduction without generate a negative impact on weight and system complexity. This study design, produce and test a Shape Memory Alloy (SMA) device that replace the original pitch link of a helicopter rotor prototype in order to vibration reducing in the critical frequency employing hysteretic passive damping, without energy machine consumption. The study employs an indoor two-bladed helicopter prototype to perform the critical operation frequency identification in the original mechanism to stablish a reference condition. In sequence, the designed SMA pitch link device replaces the original pitch link in order to perform a comparative analysis of the dynamic response. After a battery of tests, in long-term experiments with closed and distalizing springs, the results demonstrate that the device exhibited the best response when employing closed springs. The results with the SMA device demonstrate a reduction in amplitudes vibration of about 2dB, reaching about 4dB in severe test conditions. In operational circumstances, this attenuation represents crew comfort increasing, performance improvement, and wear reduction of mechanical systems due to vibration.

Keywords: vibration control, shape memory alloys, helicopter dynamics, rotor dynamics, pitch link.

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List of symbols

V_i	Induced velocity
P_∞	Pressure far from the rotor
V_c	Velocity of climb
T	Thrust
p	Air pressure
A	Area covered by the rotor
D	Drag
ψ	reference azimuth of a blade
Ω	rotor spin
N	Number of blades
A_s	austenite start temperature
A_f	austenite final temperature
M_s	martensite start temperature
M_f	martensite final temperature
ρ	air density
α_D	angle of incidence in forwarding flight
V'	total velocity of the rotor in forwarding flight
J	Energy unit (Jooule)
L	lift force
a	lift slope constant
b	number of the blades in a rotor
c	chord of the blade profile
α	angle of attack

Acronyms

CAD – Computer Aided Design

SMA – Shape Memory Alloy

SMA-SE – Shape Memory Alloy Superelastic Effect

SMA-ME – Shape Memory Alloy Memory Effect

TMD – Tuned mass damper

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1 Introduction

1.1 Motivation

Helicopters are subject to mechanical vibrations, inherent to the nature of their mechanisms and due to aerodynamic interactions. Considering routines wise, helicopter vibration is inconvenient, due to noising in operations near urban areas, for instance. Regarding performance, helicopter vibration increases maintenance costs, limits speed, and affects directly crew comfort parameters (KESSLER, 2011a). Classical solutions for helicopters vibration attenuating are well successful, yet all of the engineer solutions applied for aircraft have to commit with low weight, looking for a good lifespan, fuel economy, and good performance. The aerodynamic loads on a helicopter rotor induce vibrations, which in steady-state are periodic, so that cause fuselage vibration, transmitted from the rotor by means shaft, bearings, and linkages to the rest of the aircraft (BRAMWELL *et al.*, 2001).

The state-of-the-art in helicopter vibration control shows two main strategies: Higher Harmonic Control (HHC) and Individual Blade Control (IBC), each one exhibiting a different approach. The swashplate is a device placed in the frontier between rotating and non-rotating parts of a helicopter, (which will be detailed hereafter) that establishes a frontier aiming HHC or IBC definition. The swashplate transmits movements from the non-rotating framework to rotating parts so that, devices located above the swashplate are classified as IBC control, whereas devices located below the swashplate is classified as HHC (KESSLER, 2011a). Thus, HHC concerns vibration control in non-rotating frame and IBC leads to rotating frame control system. Solutions in IBC concerns to work with the blades, among other strategies. Within this framework, morphing technologies is employed in the blades adopting functional materials as Shape Memory Alloys (BARBARINO, S. *et al.*, 2014; BARBARINO, Silvestro *et al.*, 2011).

Shape Memory Alloys (SMA) are employed in a variety of engineering applications, among these, vibrations control. The solid-state phase transformation makes possible both the modal shapes changing and energy dissipation as a damper. Stress (Superelastic Effect), and Heat (Shape Memory Effect), induces the phase transformation of SMA by stress (SMA-SE) or heat (SMA-ME). In both circumstances, the energy, which excites SMA, take place from an external source, and the vibration control take advantage of this physic phenomenon (LAGOUDAS, 2008). Design SMA dampers is not an easy task, once specific design practices are necessary to reach successfully application (DOS REIS *et al.*, 2019).

Current helicopter vibration control employs base isolators, hydraulic actuators, and tuned mass dampers (TMD), among other technics. Concerning smart materials to helicopters vibration, in general, literature reports piezoelectricity solution and Shape Memory Alloys as SMA-ME in

morphing systems (LYNCH *et al.*, 2016; MADSEN *et al.*, 2013; PRAHLAD *et al.*, 2001). Based on vibration control with SMA-ME, vibration control technics employ active systems to induce phase transformation, in most the examples by joule effect. Based on SMA-SE is necessary enough stress to induce phase transformation and blend the proper geometry with the displacement to reach efficient energy dissipation.

HHC represents fewer technical challenges in vibration control when compared with IBC (due to non-rotating parts). In this sense, SMA may be employed as a frame structural element, gearboxes support, and similar in order to resonance avoidance (stiffness shifting) or working as a passive damper. Yet, Kessler (2011b) reports HHC limitations related to the number of the blades. This limitation avoids HHC effectiveness in all the critical harmonics. The IBC overcomes these limitations, which motivate IBC application instead of HHC. The use of SMA-ME in IBC faces the challenge of active systems (power supply) in rotating mechanisms suggesting SMA-SE application as a passive device.

Thus, in this research, it's possible to match SMA properties with helicopter problems towards reducing vibrations and their undesirable effects. Taking into account the wide of this field, this research intends to a particular case study: replace a stiff pitch link by a SMA-SE pitch link, named smart pitch link, in a helicopter prototype analyzing its dynamical response to mimic vertical flight. Thus it is possible to evaluate the performance of SMA to vibration control. Despite the literature reports about the drawbacks in applying SMA at high frequency this project demonstrates the success of the solution due to its particular configuration to specific application.

1.2 General objective

This project aims to design, build, and test a smart pitch link for helicopter vibration reduction as passive IBC to replace a stiff pitch link of a helicopter prototype. In this sense, this project performs a comparative analysis in vibration with and without SMA pitch link in the prototype as a condition reference.

1.3 Specific objectives

- Design and build a whirl tower (an experimental test bench) comprising the most common parts present in real helicopters;
- perform vibrations measurements and dynamic characterization with spectral response similar to a real machine in vertical flight;

- design and test an SMA device, named smart pitch link, to replace the original pitch link of the prototype;
- compare the dynamic response with and without the smart pitch link in vibration wise.

1.4 Scientific contribution

The application of smart materials in aeronautics is not a newness (BARBARINO, S. *et al.*, 2014; JANI *et al.*, 2014; LECCE *et al.*, 2015; EMILIAVACA *et al.*, 2019). Yet the use of Shape Memory Alloys in rotating wings, like helicopters and helix propeller, occurs in some limited circumstances, often as morphing technologies with SMA-ME (CHANDRA, 2001; EPPS *et al.*, 2004a; LECCE *et al.*, 2015; LINGHUA *et al.*, 2020; PRAHLAD *et al.*, 2001; ROGLIN *et al.*, 1996). In this sense, Lecce *et al.* (2015) classifies four strategies in helicopter blades employing shape memory alloys: variable twist, variable chord, camber and twist control. All of them concerning to morphing changing on the blades. The issues and opportunities in helicopter vibration control, aiming improvements about comfort, maintainability and performance, stimulates another functional materials application. In this subjects, piezoelectric and shape memory alloy polymers are employed in aeronautical application (blades and structures), including helicopters (NITZSCHE *et al.*, 2013; SUN *et al.*, 2016). There are a particular application of smart material, named Smart Spring employing piezoelectric ceramic in a device that changes de equivalent stiffness of a helicopter pitch link working as an active system to vibrating control (YONG *et al.*, 2004).

Despite the literature background, the use of Shape Memory Alloy adopting a passive method, like superelastic damping, is not explored and corresponds to wide field to be studied. The response time limitation when SMA-ME works as actuator may suggest drawbacks in damping applications. So far, experimental researches with SMA-SE, as DOS REIS *et al.* (2019) and MORAES *et al.* (2018) demonstrates the effectiveness of SMA-SE for structural damping, witch suggest an employing in rotating blades in an experimental approach. Thus, the possible application of SMA will not restricted to morphing blades, but also, the use of SMA-SE in damping would be used to current aircraft improvements and for future helicopters models. Considering the proposal to develop a device used in rotating wings, the proposed device may expand the field of this technology to helix propeller with variable pitch, and drones.

The characteristic of this research is theoretical-experimental. Regarding the institutional contributions, this research opens a new field for UFPB/UFCG to SMA uses for vibration control, which would become proper to industrial applications. Taking into account the potential of this technology and the strategic importance of the aeronautical industry, the patentable reduced scale

SMA device prototype is one of the research outputs. In addition, this research can supply useful information that aims the technological advance of the aerospace field. The legacy of this study trespasses the limits of this thesis, once the prototype opens an opportunity for wide further improvements.

1.5 Thesis structure

The following sections organize this thesis:

Chapter 1 presents the organization of the work in order to prepare the reader to study, motivations, justifications, and theme introduction. The introduction presents to the reader the overall perspective about this study stablishing a logic linkage between helicopter vibrations and Shape Memory Alloy technology towards vibrations suppressing. The general and specific objectives delimit the research to stablish the milestones necessities to reach the final thesis propose.

Chapter 2 presents the theoretical background about Helicopters, principles of flying, the rotor mechanism, and its vibration issues guiding the reader to the state of the art in vibration control.

In **Chapter 3**, dealing with the theoretical background about Shape Memory Alloys. Thus the chapter deals with SMA technology since it physical principles to explain SMA-SE and SMA-ME effects toward to SMA applications in vibration control.

In **Chapter 4** is presented the methodology detailing the project's main milestones: helicopter whirl tower design, and smart pitch link design. In the first phase the methodology shows the chief requirements and the prototype subsystems. The second phase show the strategy employed to design the smart pitch link and the desired performance.

In **Chapter 5** the results of whirl tower dynamic behavior are presented and the dynamic performance of the prototype with the smart pitch link installed. In addition, the fatigue tests aim to evaluate the coil spring geometry influence in long-term performance.

In **Chapter 6** express the discussions about the most relevant results concerning to device (regarding to performance) and the perspectives in context with its application were suggests future researches.

2 Helicopters

2.1 Generalities

The working principle of helicopter lifting is similar to an airplane either consider the airflow around the wings. Yet, in helicopters, the wings, known as blades, rotate around a self-static point, the main shaft, so that lift is due to the air pushing in a downward movement when the blades spin. Because of this, a helicopter is known as a rotating-wings aircraft (BRAMWELL *et al.*, 2001). In addition, the aerodynamic behavior of helicopter motivate and grounded other industry that operates according to rotating wings: wind turbines sharing the basic literature and some fail modes (HANSEN, 2008).

The application of rotary-wing aircraft is wide, from civilian to military. The versatility in maneuvers and the ability to establish hovered flight highlight its importance, mainly on difficult access places. There is amount of examples of well-success missions, executed by helicopters, considering moreover, the popularity rising of other rotating wing aircraft, for instance, drones.

Helicopters power supply are from Internal combustion or Turbofans engines, and in some models, electrical. The power transmission from the engine to shafts depends on gearboxes, corresponding to mechanical elements of power.

Helicopter rotors can be set on several ways, according illustrated on Figure 1. However, the most common configuration, found in large-scale applications, compounds a single rotor, with vertical shaft and a tail rotor with a horizontal shaft excited by the same motor. The tail rotor provides a counter-torque of the main rotor and holds the helicopter straight. Other rotor configurations like tandem-rotor and axial rotors are less common.

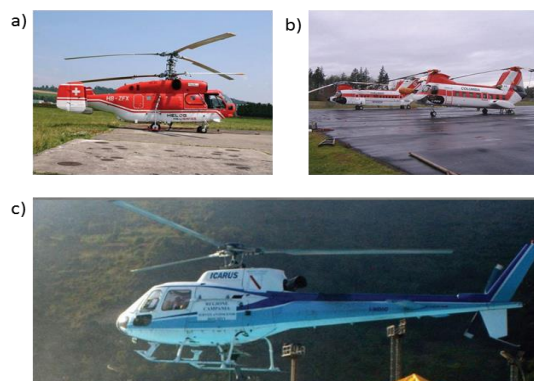


Figure 1 Helicopter rotor configuration: a) coaxial b) tandem c) single.

Adapted from FAA (2012)

For simplification, Figure 2 represents axial flight. Far upstream of the rotor, the velocity of climb and pressure are V_i and P_∞ , respectively. After flowing through the rotor, the air speed corresponds to $V_c + V_i$, where V_i corresponds to the induced velocity by the blades. Similarly, pressure below rotor is $p + \Delta p$. Thrust (T), provided by the rotor, is given by the product where A, which represents the circular area covered by the rotor and its difference of pressure (Δp), thus ($T = \Delta p A$). According to the mathematical demonstration, detailed in appendix A, is established the general expression for thrust (T) in rover and it respective induced velocity (v_i), known as Momentum Formula, respectively on Equation (1) and Equation (2).

$$T = 2\rho A(V_c + v_i)v_i \quad (1)$$

$$v_i = v_0 = \sqrt{\frac{T}{2\rho A}} \quad (2)$$

Equations (1) and (2) consider the ambient air density (ρ) and induced thrust velocity (v_0), which in hover flight is equal to v_i . Momentum formulas have importance for helicopter projects once have been associated with power, rotor dimensioning, and ground effect.

The Momentum Formula represents one particular flight condition: flight in the axial direction, with the rotor plane parallel to the ground. In the axial direction, the induced velocity vector is perpendicular to the rotor plane, because of this $v_i = v_0$. Yet, there is a particular condition, which stems from the inclined flight. This inclined flight, in steady-state, due to straightforward movement of the aircraft, is named as forwarding flight (BRAMWELL *et al.*, 2001; JOHNSON, 1994).

Forward flight is more critical than axial flight, once induces asymmetrical and periodic loads on the blades. The dynamic of forward flight affects the aircraft performance and vibrations. On Figure 3 is represented a helicopter in forward flight, stablishing the angle of incidence(α_D), the relative velocity V far upstream of the rotor, the induced velocity v_{io} and the total velocity at the rotor V' .

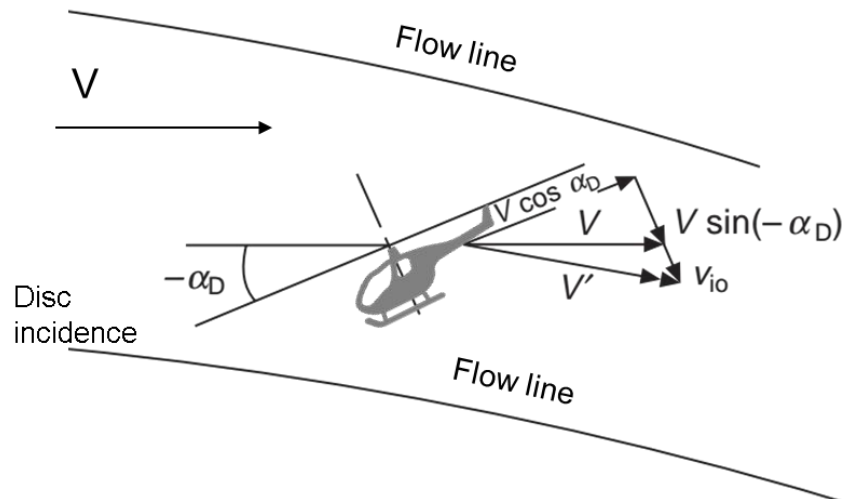


Figure 3 - Flow scheme of velocity vector in forward flight.

Adapted from Bramwell et al. (2001)

Equation (3) represents the total velocity at the rotor V' .

$$V' = \sqrt{\{V^2 \cos^2 \alpha_D + (V \sin \alpha_D - v_{i0})^2\}} \quad (3)$$

The analysis that supports Momentum Formula, assumes the induced velocity is uniform in all of disk area. Yet, in forward flight, there are two combined movements: vertical lift and horizontal displacement, which vector is represented by V' (Figure 3) (JOHNSON, 1994). The scheme in Figure 4 represents the relative behavior of rotor disk and its blades during forward flight.

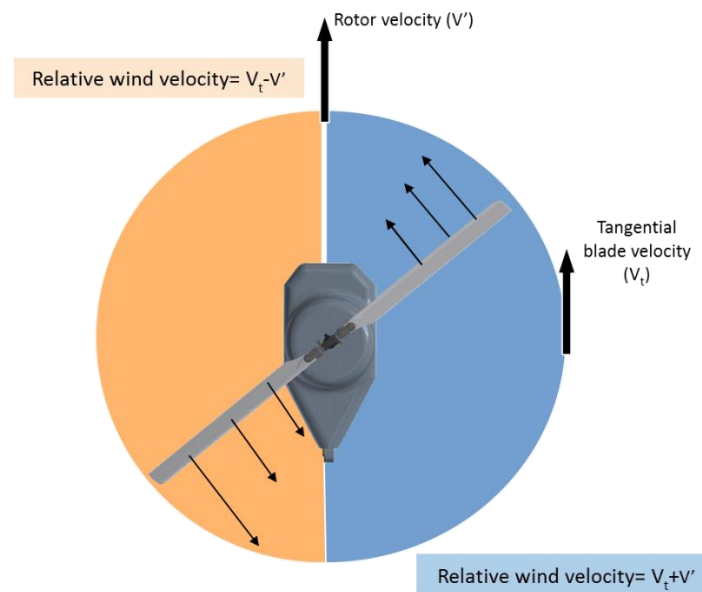


Figure 4 - Aerodynamic behavior for rotor in forward flight.

Adapted from FAA (2012)

In forward flight, in the advancing blade, the relative wind speed raise ($V_t + V'$), comparing to the retreating blade. Thus, take place differences in lifting between advancing and retreating blades. This phenomenon is named dissymmetry of lift, been one of the induced vibrations root causes (BRAMWELL *et al.*, 2001; FAA, 2012). In this scenario is possible to establish a mathematical expression, based on Equation (2), proper to forward flight.

In the course of forward flight, the disk theory can be adapted: Johnson(1994) says that in this condition, the rotor disk might be viewed as a circular wing, that advances with velocity V' . Assuming this advancing, takes it account the induced drag (D_i), which may mathematically represented as Equation (4), for a planar wing.

$$D_i = \frac{T^2}{2\rho AV'} \quad (4)$$

The induced velocity can be represented in power wise, $v_{i0} = (P_i/T)$, where P_i is the power derived from rotor. Therefore, the Equation (2) can be adapted to forward flight, as follows on Equation (5).

$$v_{i0} = \frac{P_i}{T} = \frac{VD_i}{T} = \frac{T}{2\rho AV'} \quad (5)$$

Equation (4) adopted to forward flight situations consider that, at high forward speed, the induced velocity is small when compared to the other velocity components at the rotor blade. Johnson(1994) says that at high forward speeds the rotor wake vorticity is swept back in the plane of the disk. The downwash air through the rotor impinges the tail, causing tail shake, a vibration problem, among others relates to flow field fuselage interaction (DE GREGORIO, 2012).

The vicinity of ground during take-off, for example, generates aerodynamic interactions between rotor disc and the ground, so that affects lifting, power and reduces the induced velocity of the rotor. One of the most affected parameters is the thrust, so that the relation between disc rotor radius and the height from the ground changes the thrust coefficient (C_T) which can be represented as a value of the thrust at constant power $C_T/C_{T\infty}$. In Figure 5, Johnson (1994) illustrates the behavior of thrust coefficient according to relation z/R , where z is the rotor height and R is radius of the rotor. Comparing low and high it realized the thrust increasing at constant power due to relation z/R changing.

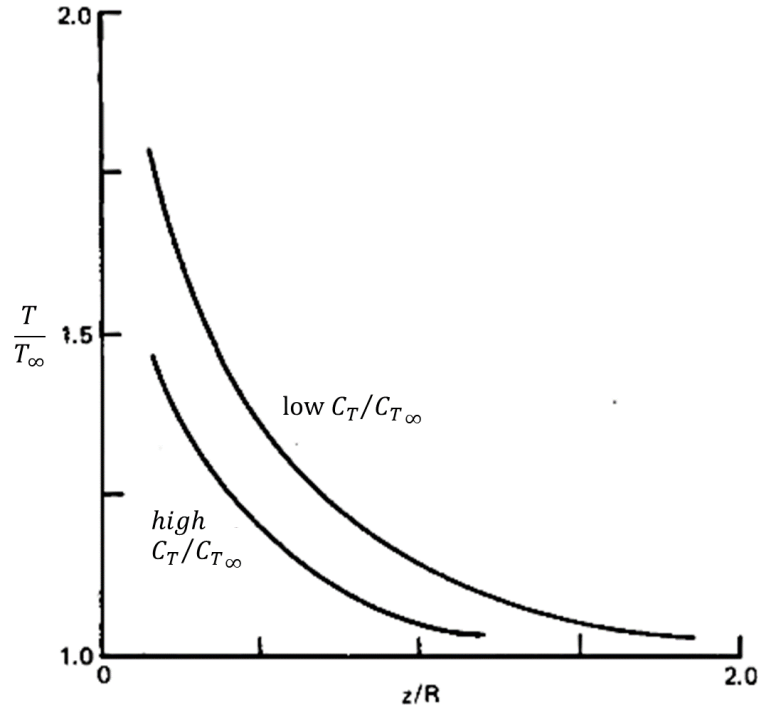


Figure 5 - Ground effect: a thrust increase at constant power. Adapted from Johnson (1994).

Cheeseman and Bennet's equation calculate the ground effect applying the ground coefficient (K) with ratio T/T_∞ in Equation (6).

$$\frac{T}{T_\infty} = \frac{K}{1 - \left(\frac{R}{4z}\right)^2} \quad (6)$$

In Equation (6) K corresponds to the ground coefficient, T is the thrust, R is the radius of the rotor and z is the rotor height relates to the ground reference.

2.3 Blade element theory in vertical flight

The relationship between thrust and velocity of airflow through the rotor disk area is established in Equation (1). In addition, it's feasible the relationship between thrust and lifting in a blade element, which forces and velocities are depicted in Figure 6.

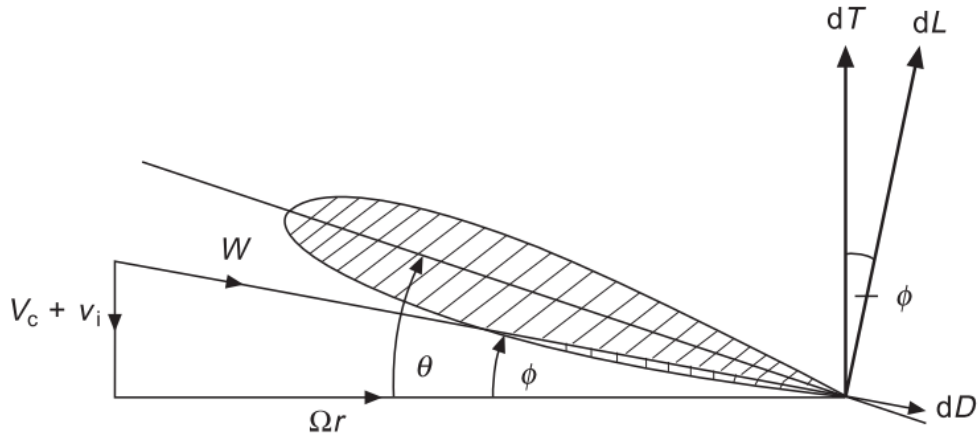


Figure 6 – Force components in blade element. (BRAMWELL *et al.*, 2001)

Figure 6 depicts a symmetric blade element profile with the forces thrust (dT) and lift (dL), in addition, climbing velocity (V_c) and local induced velocity (V_i). The pitch angle of the blade element, relative to rotor plane, named as tip path plane is θ . The direction vector of the airflow, which impinges the lead edge of the blade, is W . The angle ϕ is named inflow angle. For small ϕ , $dT \approx dL$ and $W \approx \Omega r$, where Ω the angular speed of the rotor. The variable r represents the distance between the blade root (linkage near the rotor shaft) and the blade element analyzed, so that maximum r corresponds to rotor radius (R). In Figure is expressed dD that corresponds to drag force. According to literature, based on Bramwell(2001) analysis, the elementary lift (dL) is given in Equation (7).

$$T = \frac{1}{2} \rho a b \Omega^2 \int_0^R c(\theta - \phi) r^2 dr \quad (7)$$

In Equation (7) the variables ρ, a, b and c represents: ρ - density of the air; a -lift slope, b - number of the blades, c – chord of the blade profile. The literature suggest $a=5.7$ (lift slope) based on empirical data (BRAMWELL *et al.*, 2001; JOHNSON, 1994).

The angle ϕ may be represented according to the air speeds V_c and V_i and Ωr in Equation (8).

$$\phi = \frac{V_c + V_i}{\Omega r} \quad (8)$$

In the same analysis is established the attack angle(α), been $\alpha = (\theta - \phi)$. Thus, for hovering flight ($V_c = 0$) the angle of attack is calculated according to Equation (9).

$$\alpha = \theta - \frac{v_i}{\Omega r} \quad (9)$$

In addition, considering lift per unit span (L') and drag per unit span (D'), as depicted in detail in Figure 7, the concept of aerodynamic moment per unit span establishes ($M'_{c/4}$) as a moment virtually applied on center of pressure x_{cp} .

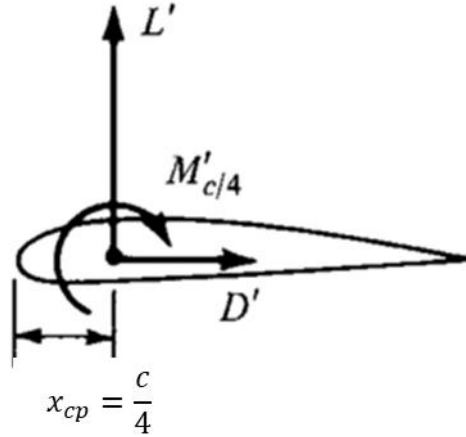


Figure 7 – Equivalent force at quarter-chord point. Adapted from *Anderson(2013)*

As depicted in Figure 7, x_{cp} is placed at $c/4$ from leading edge of the profile, for symmetric airfoils (Equation (10)). Thus, the aerodynamic forces generate a moment in x_{cp} , so that for rotating machines may transmitted to blade mechanism.

$$M'_{\frac{c}{4}} = -x_{cp}L' \quad (10)$$

Assuming $dT \approx dL$ and the thrust vary according to r (Equation (7)), the most severe condition ($r=R$), in the blade tip, matched to aerodynamic moment (Equation (10)), centered in the center of pressure, Equation (11) determine the maximum moment in the root of one blade ($b=1$).

$$M'_{\frac{c}{4}} = -\frac{1}{6}\rho a\Omega^2 c(\theta - \phi)R^3 x_{cp} \quad (11)$$

2.4 Maneuvers mechanisms

Helicopters has mobility to maneuver in virtually all directions. Unlike an airplane, a helicopter is capable of performing three basic movements: stationary mode (hover), forward flight, backward flight (reverse flight) and rolling. Maneuvers are possible due to individual blade mechanisms, were the rotor hub connects the blades by hinges, which scheme is depicted in Figure 8.

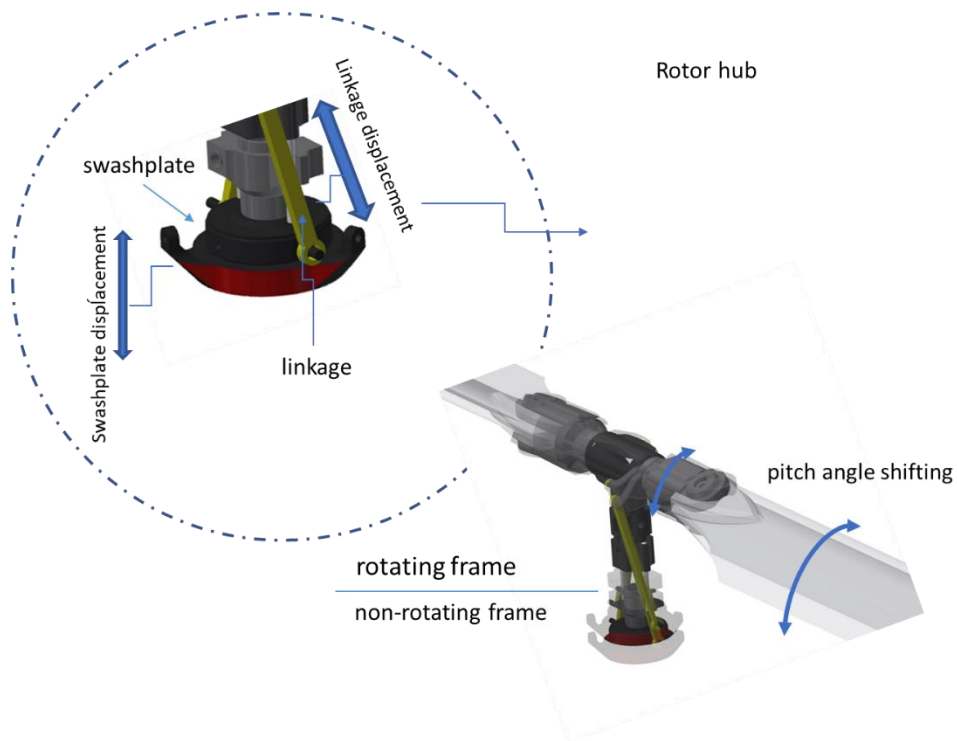


Figure 8 - Rotor hub basic mechanism. The author.

The swashplate is the chief component, which detailing of a real scale part is represented in Figure 9. The stationary part of the swashplate is connected to the non-rotating frame of the aircraft so that receives commands from the pilot by servos. A rotating swashplate is a part of the rotating components and connects blades by linkages. Spherical bearing contacts the main shaft and spin freely. In brief, the swashplate conveys the command from a stationary frame to the rotating frame.



Figure 9 - Swashplate of a Sikorsky S-61N helicopter.

Adapted from AAIB (2020)

The servo controls the swashplate in three points, placed around its circumference with 120° span. These points may have commanded individual or simultaneously. When control displace the swashplate in all the points simultaneously, these movements results from a Collective Command. When occurs individual displacement of one or more of these points, the command is named Cyclic Command. The Figure 10 shows the mechanism of cyclic/collective. Points A, B and C represents the linkage connections.

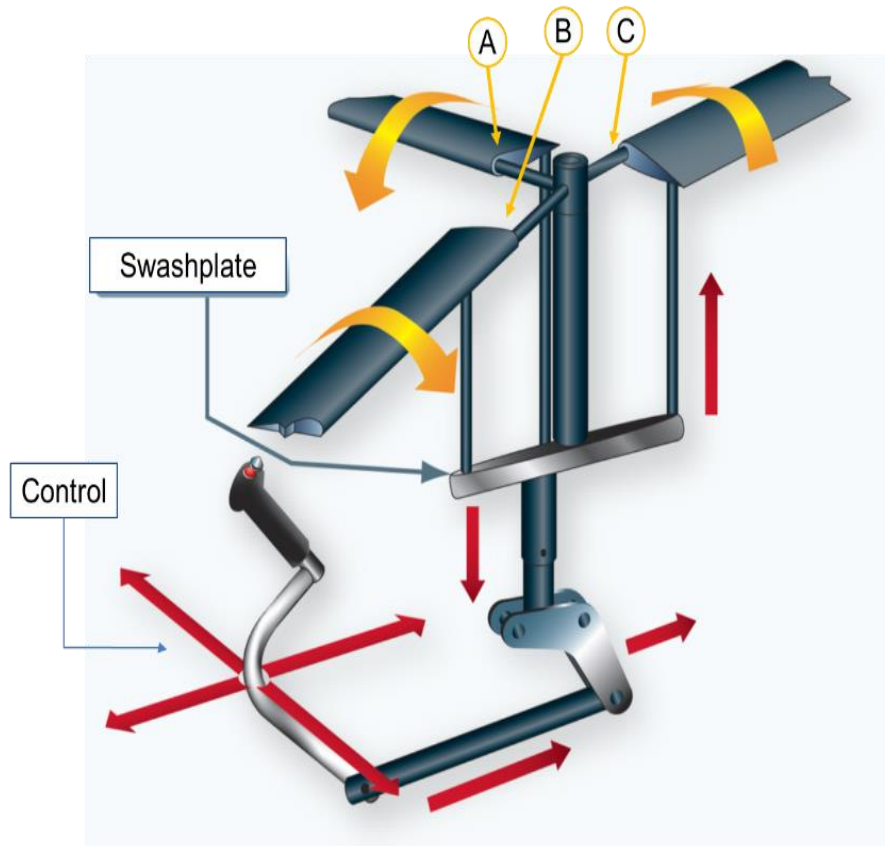


Figure 10 - Cyclic/collective mechanism.

Adapted from FAA (2012)

Collective command performs the displacement of linkage points A, B and C simultaneously, up or down. Collective activation occurs for hover, on take-off and landing operations, due to variation of vertical thrust, keeping the rotor parallel to the ground. The changing of thrust is allowed by pitch angle shifting by collective command. Otherwise, cyclic command allows lateral maneuvers, like rolling, or towards forward flight due to asymmetrical displacement of points A, B, C according to the maneuver. In cycling, each blade can perform a different pitch angle for each angular position of the rotor.

For Bramwell et al (2001), the development of helicopter owes much to the introduction of hinges (allowing relative movement of the blades on rotor hub) about which the blades are free to move. The typical movements of the blade are labeled in Figure 11.

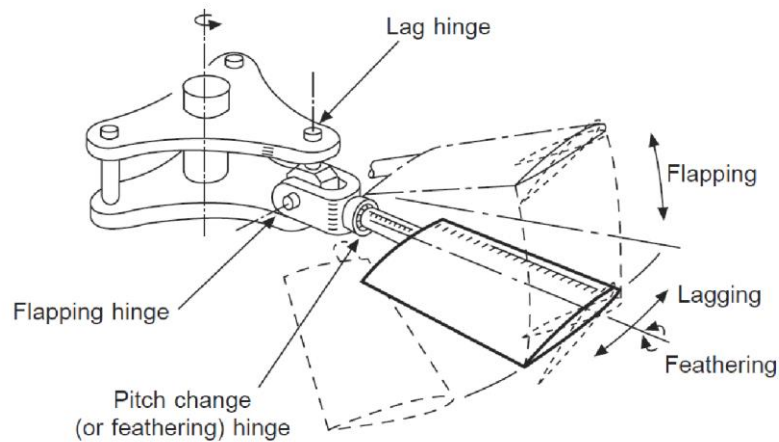


Figure 11 - Hinge arrangement and blade movement's label. From Bramwell *et al.* (2001)

2.5 Helicopter vibrations

The vibration phenomena are present in helicopters and observed in various aircraft models, regardless of scale. Welsh(2018) demonstrates in a review paper, supported by Bramwell *et al.* (2001) and technical papers in the American Helicopter Society (AHS), among others, that the rotor corresponds to the main source of vibration.

The intrinsic instability of helicopters results in dynamic effects, like vibrations, which represent a limiting of performance, comfort, safety, and increasing maintenance costs. Thus, is important to find out the root causes of these instabilities in order to eliminate or mitigate their effects.

Vibrations are a very common phenomenon in helicopters and are not only intrinsic to the running of its motor and gearboxes but also associated with temporary instabilities brought about by maneuvers(CAILLET *et al.*, 2012). Vibrations increasingly occur during forwarding flight, reverse flight, and rolling, so that is important to itemized each root causes concerning to it proper approach.

Vibration levels have to follow international standard limits, regarded to crew comfort and safety. In addition, the monitoring of such vibrations is an indicator of the machine structural integrity (EASA - EUROPEAN UNION AVIATION SAFETY AGENCY, 2020; EUROPEAN AVIATION SAFETY AGENCY, 2021; PEGADO, 2010). In Figure 12, Kessler(2011) shows helicopter vibration levels decreasing since 1955 to 1995, highlighting some strategic targets like MIL-H-8501A (Military Specification: Helicopter Flying and Ground Handling Qualities, General Requirements For), the specifications of AAH/UTTAS (Utility Tactical Transport Aircraft System) and, minimum vibration levels recommended by NASA, a huge target to be achieved on 0.02g.

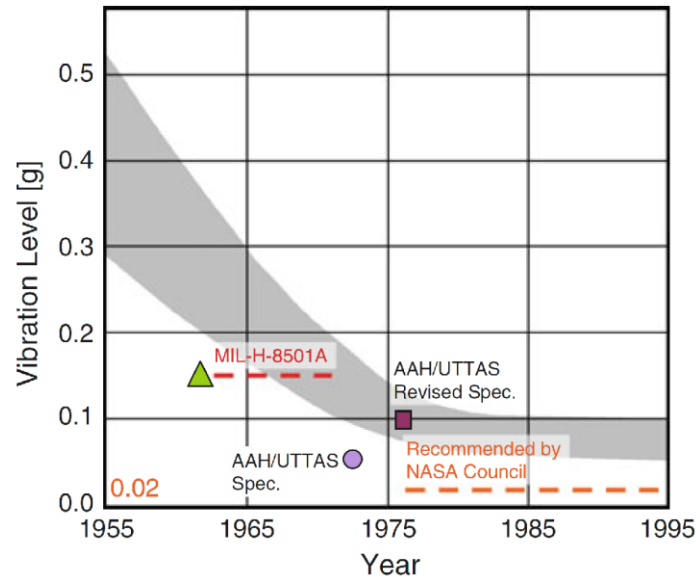


Figure 12 - Helicopter vibrations levels decreasing. Adapted from Kessler (2011a)

Helicopter vibration affects maintenance costs and flying safety. The European Union Aviation Safety Agency, in its Annual Safety Review, addresses failures in categories and, indicating that system failures, technical failures, and aircraft maintenance represent a considerable relevant risk (EASA - EUROPEAN UNION AVIATION SAFETY AGENCY, 2020). The Center of Investigation and Prevention for Aeronautical Accidents (CENIPA), statistically reported that system failures, system malfunction, or lost control in flight represent about one third of accident causes (CENTRO DE INVESTIGAÇÃO E PREVENÇÃO DE ACIDENTES AERONÁUTICOS - CENIPA, 2017). Thus, taking into account the importance of vibration, engineer works towards studies its root causes.

Vibration origins due forces interaction in the structure. During flight, many external forces take place on helicopter body and, internal forces in its blades and peripherals. In Figure 13a are depicted the hub forces in direction x, y, and z, respectively labeled as X, Y and Z. In addition, the moment components, where L correspond to rolling movements, M, tilt moments and, N yaw movements. Maneuvers mechanism, depicted in Figure 8, and Figure 10, performs moments M and L.

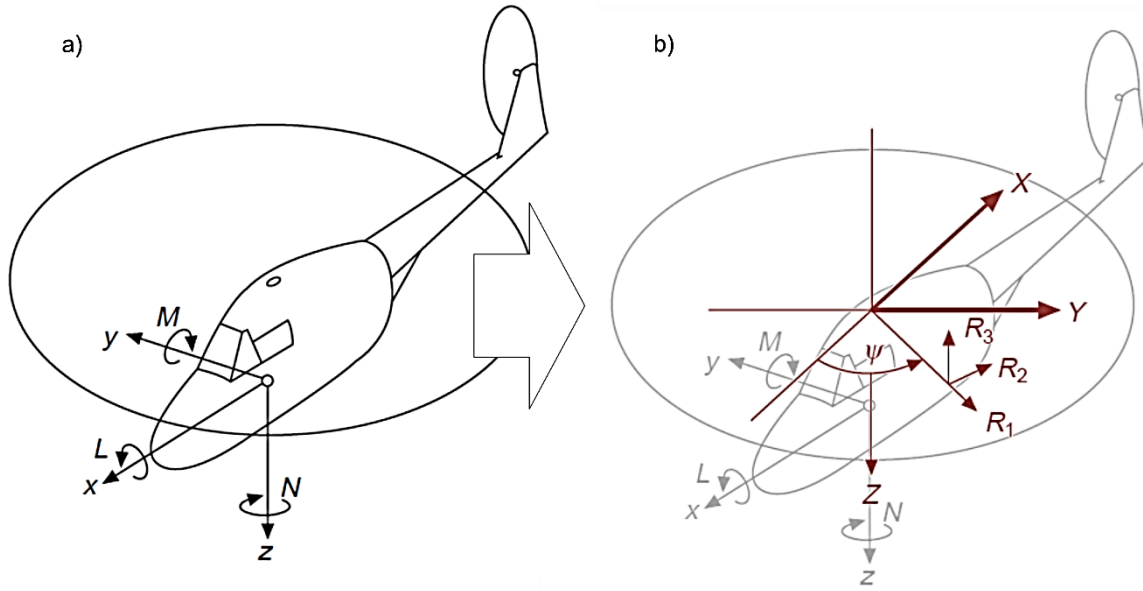


Figure 13 - Force and moments components along helicopter body and blade: a) moments in helicopter (L,M,N) b) force components on the blade. Adapted from Bramwell *et al.* (2001)

Figure 13b shows the force components of reactions relating of the blades hinge: R_1 along the blade span, on xy plane; R_2 along the blade chord, on plane xy and, the component R_3 on z direction. The angle Ψ represents the reference azimuth of a blade. For k_{th} blade of a rotor, $\Psi_k = \frac{2\pi k}{b}$ where b represents the number of blades. The mathematical background developed in Bramwell *et al.* (2001) and Johnson (1994) concludes that the correspond components of forces in x , y , z direction, are modeled as Equations (6-8).

$$X_k = -R_{1k} \cos \Psi_k + R_{2k} \sin \Psi \quad (6)$$

$$Y_k = -R_{1k} \sin \Psi_k + R_{2k} \cos \Psi \quad (7)$$

$$Z_k = -R_{3k} \quad (8)$$

In steady flight, forces R_{1k}, R_{2k}, R_{3k} are periodic. Taking into account the rotor spin at frequency Ω , and if the blades are perfectly matched, the internal forces from all the blades cancel itself at the hub, except for those harmonics at multiples of $N\Omega$. This harmonics are transmitted to the aircraft where N corresponds to the number of the blades. Thus, rotor hub works as a filter, transmitting to the helicopter body only frequencies multiples of $N\Omega$. Johnson (1994) summarizes in

Table 1, the helicopter vibration through the rotor hub with N blades, where “rev” represents each rotor revolution, and t represents each harmonic of the source signal.

Table 1 - Transmission of helicopter vibration through the rotor hub.

Adapted from (JOHNSON, 1994)

Rotating frame (vibration source)	Nonrotating frame (transmitted to)
Vertical shear at $\Omega N/rev$	thrust at $t \Omega N/rev$
Lagwise moment at $\Omega N/rev$	torque at $t \Omega N/rev$
In-plane shears at $\Omega N \pm 1/rev$	rotor drag and side forces at $t \Omega N/rev$
Flapwise moment at $\Omega N \pm 1/rev$	pitch and roll moments at $t \Omega N/rev$
Feathering moments at $\Omega N/rev$	collective control system forces at $t \Omega N/rev$
Feathering moments at $\Omega N \pm 1/rev$	cyclic control system forces at $t \Omega N/rev$

Based on Table 1, vibrations not exclusively transmitted through the shaft but also through swashplate and links mechanisms, associated with collective and cyclic systems. The individual blade movements (lagwise and flapwise), as depicted in Figure 11, transmit vibratory loads to collective and cyclic systems, while vertical forces are transmitted through the shaft.

The rotor's vibrations tend to increase in forwarding flight, as the speed increases. Otherwise, these vibrations tend to fall as the number of the blade increases. Thus, a helicopter that employs N blades exhibits fewer vibration levels than a model with $N-1$ blades, for instance.

Another source of vibration, which directly affects safety, is named ground effect: the dynamic instability involving a harmful interaction between the blade lag-mode with the fuselage motion during aircraft landing (CIAVARELLA *et al.*, 2018; DA SILVA *et al.*, 2020; SANCHES, 2011; SOUZA *et al.*, 2014).

Finally, general aerodynamic effects can induce vibration on helicopters as downwash movement of the air, that corresponds to changing of air flows through the main rotor reaching the aircraft tail, inducing vibrations on the structure or fluid-structure interactions when the aircraft is near obstacles (SHAHDIN *et al.*, 2011; ZAGAGLIA; ZANOTTI; GIBERTINI, 2018).

2.6 Vibration control systems

Vibration control systems for helicopters comprise active and/or passive strategies, in most of the cases, adopting classical engineer solutions to vibrating control. Yet, is important to define two

wide fields on vibrations control for helicopters: the solution classified as High Harmonic Control (HHC), solutions grounded on Individual Blade Control (IBC), and Active Control for Structural Response (ACSR). Kessler(2011), in a review paper, addressing this subject, where cite the first proposal of HHC, which principle dates back to 1952. In the same paper, the author presents the drawbacks of HHC, and IBC principles, as a modern solution, although more complex. In addition, the literature reports the use of ACSR, which doesn't deal with the origins of vibration but minimizes its consequences (AUSTRUY, 2011; KRYSINSKI *et al.*, 1998). In order to help the definition between HHC, IBC or ACSR, the swashplate represents a border on this classification.

The swashplate (Figure 8 and Figure 9) is a device located in the frontier between rotating and non-rotating parts of helicopter, where transmits movements from non-rotating structure to rotating parts. Devices involved in vibration control, located above the swashplate are classified as IBC control, while devices in this wise located below the swashplate are classified as HHC. Due to the difficulty of implementing mechanical devices in rotating parts, IBC faces more challenges to implementation than HHC. ACSR corresponds to the solution implemented in the fuselage to react in opposition to vibration from the rotor, so that, the state-of-the-art focused in HHC and IBC.

Considering the nature of solution, Welsh (2018) classifies vibration methods reduction in three categories: isolation, fuselage suppression and, rotor suppression. In other words, independently of its position in rotorcraft, the strategies may concern to:

- a) isolate de origin of vibration, not focused on solve the vibration source, but avoid this effects for cabin and crew;
- b) suppress the effect of vibration in fuselage, employing actuators to induce anti-resonance, for instance, classified as Fuselage Suppression;
- c) suppress the effect of vibration in its origin, rotor and blades, classified as Rotor Suppression.

The methods employed to vibrating control in helicopters make use of passive or active technologies. These solutions might be incorporated into the pre-existent structure or requires several redesigns. A group of passive methods currently applied to vibration reduction employs devices for base isolation such as elastomeric isolators, installed between the fuselage and the block compounded of gearbox, engine, and rotor.

Another passive method, as an IBC system, may employ dynamic vibration absorbers, such example the Dynamic Anti-Resonant Vibration Isolator (DAVI) and all similar solution, based on extra mass attaching an elastic connection on the body (BRAMWELL *et al.*, 2001; FLANNELLY, 1967). In Figure 14 the scheme shows DAVI with a bobweight mass, spring K and two masses, M_1 (rotor hub) and M_2 (fuselage).

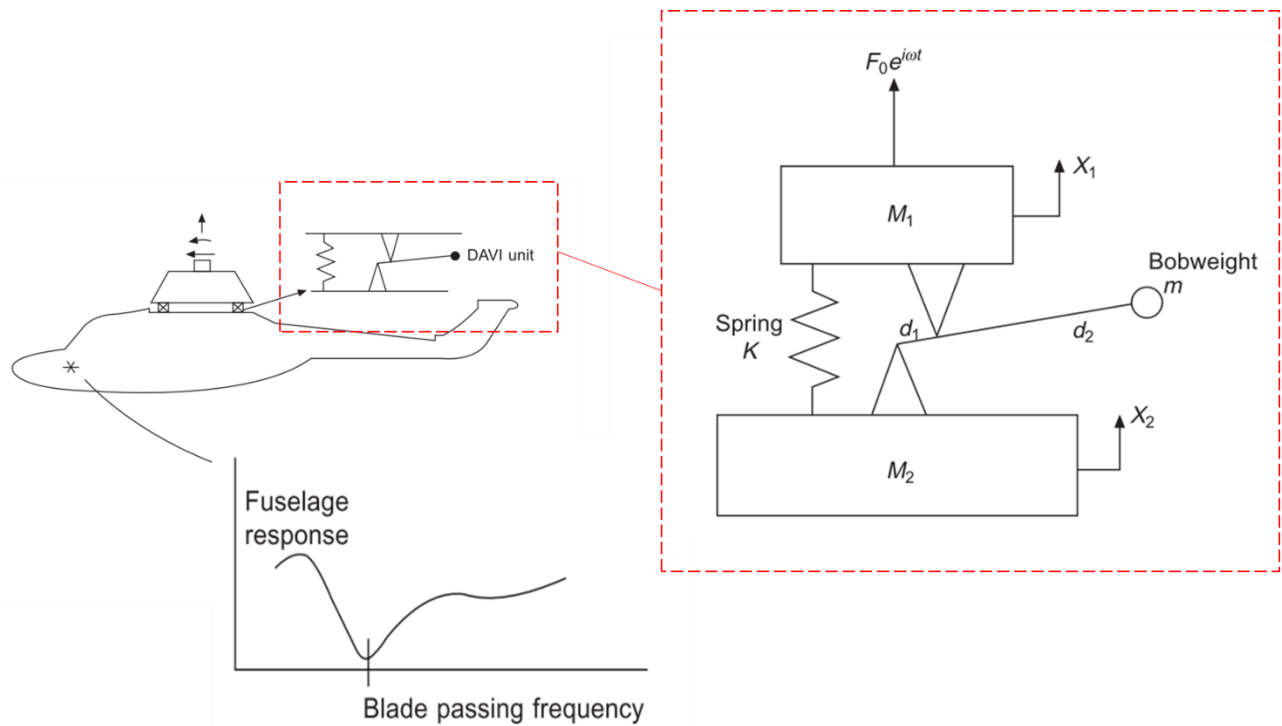


Figure 14 – Scheme of DAVI system.

Adapted from (BRAMWELL *et al.*, 2001)

Taking as an example of vibration absorbers, incorporated as a part of an IBC system, such pendulum absorber, is installed on Bell 412 Helicopter, depicted on Figure 15 (NAGASAKA *et al.*, 2010).

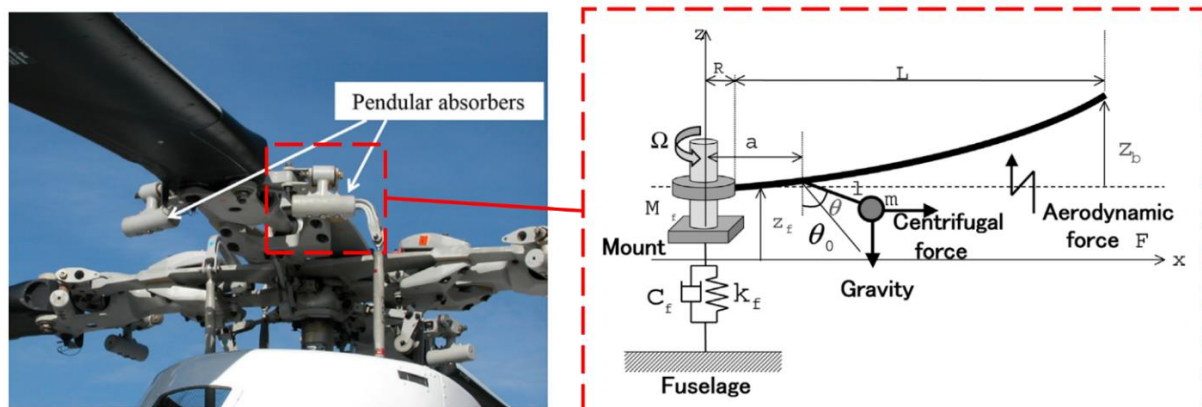


Figure 15 - Example of pendulum absorber and respective physical model.

Adapted from (WELSH, 2018; YANG *et al.*, 2009) and (NAGASAKA *et al.*, 2010)

The pendulum absorber takes advantage of the centrifugal force. The system implements a self-tuning, once the centrifugal force depends on the rotor speed. At low angular speed Ω , the centrifugal force on mass m is neglected, compared with gravity. The rod where mass m is attached is placed at a distance a from the main shaft and is articulated. According to the rotational speed raising,

the centrifugal force increasing to yield a system that equalizes the effect of periodic aerodynamic forces and other blade internal forces.

All devices, which applies TMD (Tuned Mass Damper), in these circumstances, are named Frahm, in memory of Herman Frahm, who invented vibration absorber in 1890, using TMD concept.

For Welsh (2018), a good vibration level should be less than 0.03-0.05g, so that helicopter reliability and cabin comfort would be significantly improved.

The author aware the technologies listed on Table 2 do not represent all of the strategies currently adopted to helicopter vibration control, so that this table do not exhaust this content.

Table 2 – Strategies to vibrating control in helicopters

Designation	Approach	Where is applied	Type
DAVI (Dynamic Ant-resonant Vibration Isolator) (FLANNELLY, 1967)	Pendular absorbers take advantage of centrifugal force	Blade root	Passive
LIVE (Liquid Inertia Vibration Eliminator) (DENNIS R. HALWES <i>et al.</i> , 1980)	A liquid flows between internal chambers so that works as a DAVI without mechanisms.	Between main transmission, gearboxes and fuselage	Passive
Magnetorheological Fluid damper (MARATHE <i>et al.</i> , 1998)	Damper employs magnetorheological fluid activated by On-off control scheme, or Feedback Linearization scheme.	Blade	Active
Trailing edge SMA actuator (EPPS <i>et al.</i> , 2004b)	Trailing edge tab shifting by SMA actuator.	Blade	Active
Smart Spring (YONG <i>et al.</i> , 2004b)	Stiffness shifting on pitch link rod by piezoelectric mechanism, changing the dynamic response of the blade.	Blade	Active
SARIB (Système Antivibratoire à Résonateur Intégré à Barres) (KONSTANZER <i>et al.</i> , 2008)	Leaf spring and an arm supporting a flapping mass.	Supports the gearbox and rotor head.	Passive
Smart spring piezoelectric actuators (NITZSCHE <i>et al.</i> , 2015)	Varying blade linkage stiffness with an active device using piezoelectric	Blade	Active

ACSR (Active Control of structural response) (GANGULI <i>et al.</i> , 2016).	Varying forces by actuators located between the rotor and the fuselage.	Fuselage	Active
ATR (Active Twist Rotor) (GANGULI <i>et al.</i> , 2016)	Composite with piezoelectric on the blade. The blade twist, morphing according to demand.	Blade	Active
Algorithm to suppress limit cycle oscillations (DA SILVA <i>et al.</i> , 2020)	Strategy employing Fuzzy to control shock absorbers on the structure to avoid the effects of ground resonance.	Fuselage	Active
Periodic mono-material for noise reduction (WANG <i>et al.</i> , 2020)	Absorbers employing supports with cylinders and spiral structures to generate large impedance mismatch through the geometrical discontinuity	Fuselage	Passive
NiTi SMA superelastic pitch link (OLIVEIRA, 2022)	Passive NiTi SMA superelastic pitch link prototype employing helical mini coil spring for vibration attenuation.	DEVICE PROPOSED IN THIS RESEACH	

2.7 Experimental vibrations benches

The more accurate method to measure helicopter vibration perform experiments in a real machine, as reported in many examples on the literature (DAMY, 2006; GEÇBURA *et al.*, 2017; STUPAR *et al.*, 2012). However, studies and experiments with real helicopters may expensive and dangerous. An alternative to this challenge is to use numerical simulation (ANUSONTI-INTHRA *et al.*, 2001; LIU *et al.*, 2017; POMIN *et al.*, 2001; RAGHAV *et al.*, 2013; SINEM *et al.*, 2018). When real prototypes are affordable, it is feasible to compare numerical simulations with experimental results so that to validated models. The facilities properly designed to carry on these experiments are named whirl towers.

The whirl towers are physical prototypes, with parts mimic real helicopters, that support experiments and supply substantial and reliable results. These facilities correspond to a physical prototype, generally compounded of the main rotor hub with its blades, which may take into account gearboxes and part of the fuselage. Whirl tower sizes vary significantly and their dimensions depends on the experimental purpose. Figure 16 shows examples of real scale prototypes. Yücekayali *et al* (2013) (Figure 16a) employs an outdoor whirl tower, similar to an unmanned helicopter in order to study the performance of the rotor in hover flight, previously simulated by numerical method.

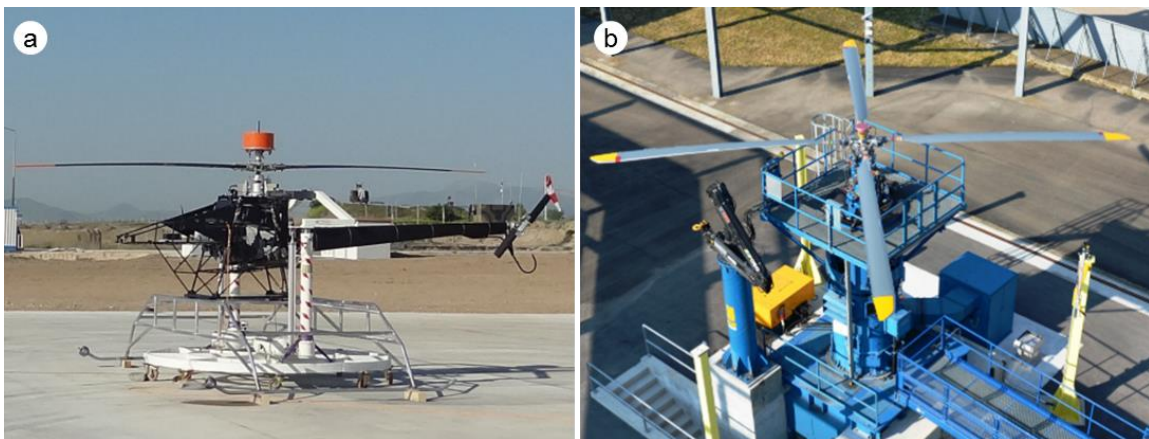


Figure 16 - Outdoor whirl tower examples: a) studied rotor installed on an unmanned helicopter.

Adapted from Yücekayali *et al.*(2013) b) BK117-type main rotor installed on the whirl tower. Adapted from Richter *et al.* (2016)

Similarly, Richter *et al.*, (2016) (Figure 16b) applied an outdoor whirl tower of Airbus Helicopter Deutschland to evaluate the hover performance of a BK117 helicopter rotor. The rotor radius $R=5.5\text{m}$, where measurements comprised the main shaft and control path forces and moments of the rotor.

Figure 17 depict examples of real scale facilities. Ciavarella *et al.*, (2018) (Figure 17a-b) characterize the structural dynamic interaction due to ground effect, where a real machine, supported through a test rig, is excited by a modal shaker in order to capture vibration signals in about 250 acceleration sensors. In (Figure 17c-d) Straub *et al.* (2018) submits a prototype in a wind tunnel in order to test a smart material active flap rotor.

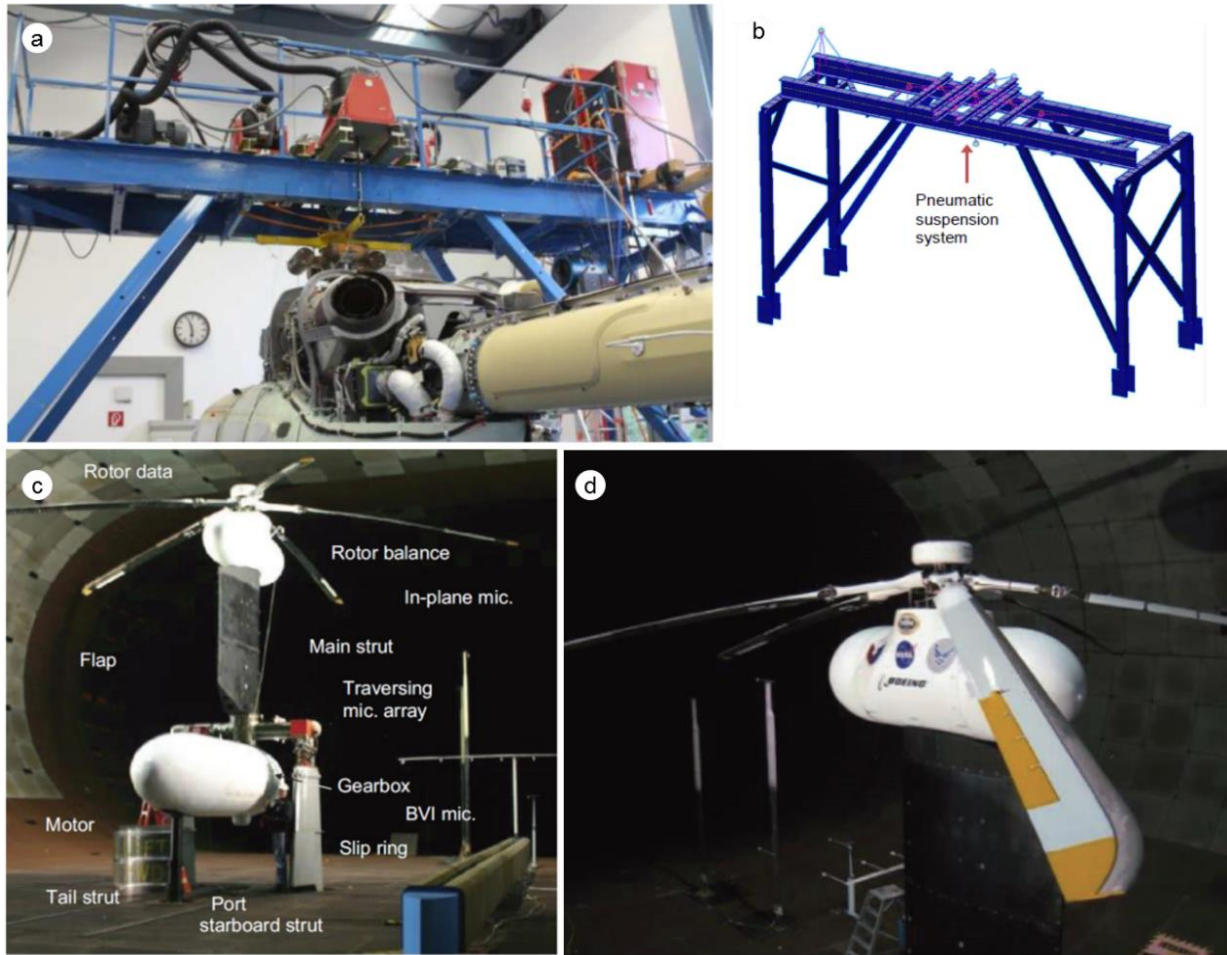


Figure 17 - Indoor test facilities: a) real aircraft suspended, with rotor excited by a modal shaker b) overall view of helicopter test rig applied in "a", modeled by FEM. c) overview of helicopter prototype under test on wind tunnel d) detailed view of rotor hub displayed in "c". Adapted from Ciavarella *et al.* (2018) and Straub *et al.* (2018)

Other examples of indoor scale prototypes are presented in Cameron *et al.* (2015) and Dong *et al.*, (2020) employing dynamic characterization and digital image correlation of blades three-dimensional deformations. Shape similar to a real aircraft fuselage is applied in De Gregorio (2012) concerning study fluid-structure interference and, in Zagaglia *et al.* (2018) to study aerodynamic interaction with obstacles.

Reductions in the number of parts may employed to delimit the object of study, as in Nitzsche *et al.*, (2013) (Figure 18), in which a simple rotor made of a motor and one blade was used to validate an adaptive pitch link for rotor vibration reduction.

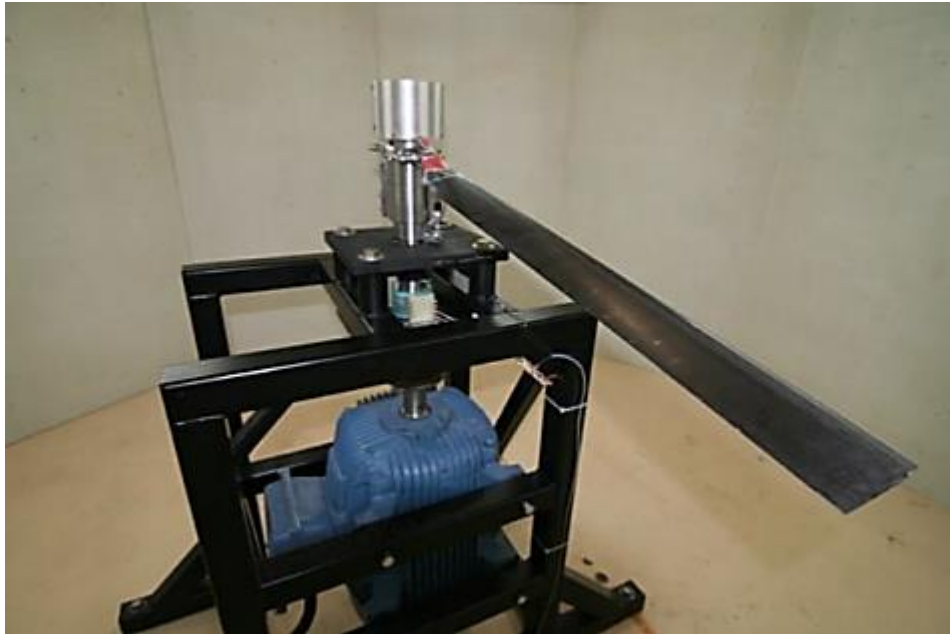


Figure 18- Whirl tower facility to validate pitch novel link. From Nitzsche *et al.* (2013)

3 Shape Memory Alloys - application of helicopter vibration control

The application of SMA in vibration is not the exclusive use for this material. Jani *et al.*, (2014), in a complete review, show the state of the art in SMA application in a large variety of areas, among them, aeronautics. In this chapter, after a brief SMA theoretical explanation, the lines concern to SMA application in helicopter vibration.

The distinction between Shape Memory Alloys (SMA) and other alloys is, by far the capacity of restoring the original shape after accentuated deformations. The study devoted to industrial and medical application is widely covered by the literature and no limited to shape memory effects (LECCE *et al.*, 2015). The fundamental reason for the unique behavior of the SMA is due to the transformation of the martensitic phase, a term originally used in the Fe-C structure, also adopted in SMA to designate a type of phase transformation into a solid-state of the non-diffusional type (ELAHINIA, 2015).

3.1 Theoretical framework

Generally, SMA exhibit two behaviors, in which it takes advantage in order to vibrations reduction: Memory Effect (ME) or SMA-ME (thermally induced), and Superelastic Effect (SE) or SMA-SE (mechanically induced). The literature deals with SMA supplying substantial information about microstructural phenomena of this material, its applications, and potential. Figure 19 summarizes SMA-ME, and SMA-SE.

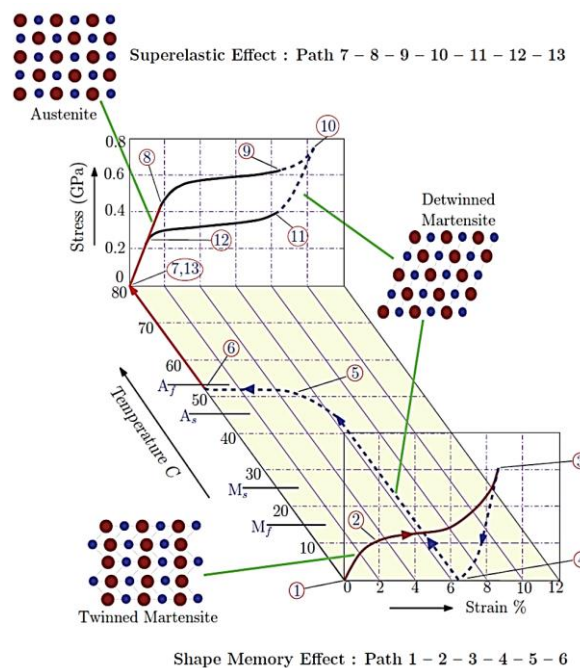


Figure 19- A three-dimensional stress–strain–temperature plot describing the thermomechanical behavior of SMA. SMA-ME (path 1 to 6). SMA-SE (path 7 to 13). Adapted from Rao *et al.* (2015)

The plot depicts the microstructure illustration (Martensite and Austenite), the phase transformation temperatures A_s (austenite start temperature), A_f (austenite final temperature), M_s (martensite start temperature), and M_f (martensite final temperature).

In the origin of the axis system (point 1), the material in the martensitic phase has a classic elastic behavior of metals when subjected to small loads. In the section that starts at the point identified by 2 until point 3, under high loads, the martensite undergoes deformation, leading to the shape of the self-accommodated martensitic variants. Removing the load, section (3 $\rightarrow$ 4), the material remains as twinned martensite. In point 4, it is observed the material exhibit deformation even after unloading. When heated, on the way (4 $\rightarrow$ 5 $\rightarrow$ 6 $\rightarrow$ 7), reaching the temperature above A_f , the material recovers its original dimensions, without loading, remaining in the austenitic phase. This complete recovery describes the memory effect in a simple way. The phase shift to stable austenite is only complete when the temperature exceeds A_f (RAO, Ashwin *et al.*, 2015).

Path 7 to 13 describes the Superelastic effect (SE). In SE, the phase transformation is induced by stress, in a region which stable austenite under temperature above A_f . Due to mechanical loading, take place the solid-state transformation from austenite to martensite phase, after unloading, there is a reverse phase transformation to martensite (RAO, Ashwin *et al.*, 2015).

When mechanical loading (7 $\rightarrow$ 8) starts, austenite exhibits elastic behavior and under continuous deformation (8 $\rightarrow$ 9), it is thermodynamically stable for austenite to start transforming to a single variant martensite. The deformation continues (9 $\rightarrow$ 10), without phase transformation and only with elastic behavior, when the loading stops (point 10). During unloading (10 $\rightarrow$ 11), starting from point 11, the reverse phase transformation from austenite to martensite begins. This transformation will be complete at point 13.

The phase transformation in the superelastic effect presents mechanical hysteresis, represented by the closed region, demarcated by the line 7-8-9-10-11-12-13. This region corresponds to the energy dissipated in each cycle, and the size of this region will depend on the composition of the alloy and the loading conditions (LAGOUDAS, 2008).

Grounded in applications wisely, the variety of SMA solutions covers areas from medical to aerospace. The level of technology maturity motivates the scientific research, technological applications, demonstrated in the number of patents involving this material. This results are illustrated in Figure 20. Although the data collection took place in 2014, the projection depicted in the graph indicates a high number of papers and patents expect until 2019.

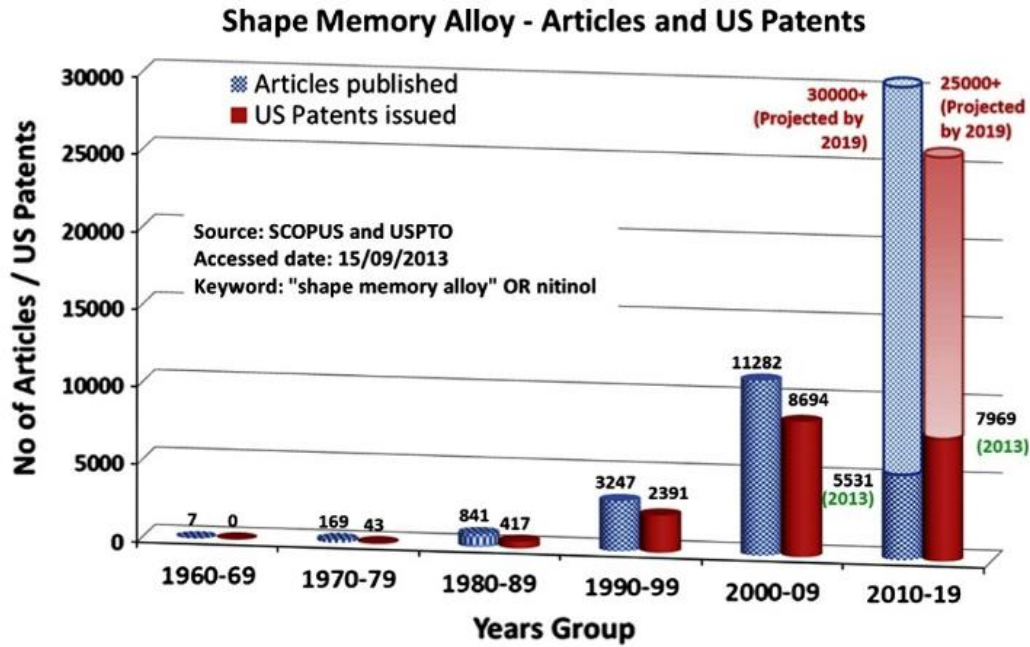


Figure 20 - Number of “Shape Memory Alloy” articles and patents by years-group applications.

Source Jani *et al.* (2013)

The motivation to SMA employments can be explained due to its advantages when compared with “classical” technologies as hydraulic, pneumatic, and electric actuators. The reduced number of components when use SMA and the higher energy density taking into account the SMA device weight corresponds to the main advantages (JANI *et al.*, 2014; LAGOUDAS, 2008).

Taking account SMA-ME and SMA-SE, is possible applying this material in vibration control. SMA-ME promotes stiffness changing allowing dynamical natural frequencies, shifting in order to avoid resonance as reported by a few examples from literature (AGUIAR *et al.*, 2013; BORGES *et al.*, 2013; HE *et al.*, 2007; OLIVEIRA *et al.*, 2020; UTTER, 2019). SMA-SE mechanical hysteresis corresponds of energy dissipation due to phase transformation, which allows reducing vibration levels, as reported by literature.

Thus, SMA exhibit wide potential for vibration control, in all types of machines. In addition, considering helicopter application, it takes advantage of the SMA effectiveness, aligned with few payload increasing, an important requirement for aeronautics.

3.2 Helicopter vibration control with Shape Memory Alloy

Shape Memory Alloys in helicopters may adopted, both passive and active methods. In both circumstances, the chief gain reached with SMA is the low net load increasing in the aircraft due to its employment when compared with the previous solution. In this sense, SMA may employed as HHC solution and in IBC in variety of forms. According to Lecce and Concilio(2015) a drawback of SMA applications in helicopters is regarded to its low frequency response. Yet this limitation regards to SMA-ME activated by heat, instead of SMA-SE. In this framework Reis (2018) employs SMA in wind turbine blades to attenuate frequencies above 18Hz, for instance. Thus, avoiding generalizations, the applications toward strategies to pursue solutions below maximum frequency limits.

Considering low-frequency application, SMA may be employed as an actuator to perform morphing in helicopter blades. The patents research, using the database of Google Patents, Spacenet, and INPI, with a combination of the keywords “Shape Memory Alloy” and “Helicopter Vibrations” exhibits solutions toward SMA working as an actuator. For INPI, due to the few results, its used the term “Helicopter Vibrations”. Table 3 exhibit the summary of patents search results, highlighting the latest result submission.

Table 3 – Patents search summary

Database	Number of results	Latest result submission
Google Patents	4	2019
Spacenet	12	2021
INPI	4	2012

The SMA actuators perform morphing wing strategy or its variations, similar to exhibit since Rogling and Hanagud (1991), to Chopra and Epps (2004) that employs SMA in way of wires or thin plates to change the camber in a blade or wing profile to shifting trailing-edge tab, as depicted Figure 21. SMA are disposed of in wire or slender plate (as can see in the red line, highlighted in the Figure) so that, activated by electrical current (joule effect) shift mechanisms to change the blade profile or rotate a tab. These mechanisms do not require high frequencies due to low shape speed requirement because to not directly relate with the rotor's excitation frequency.

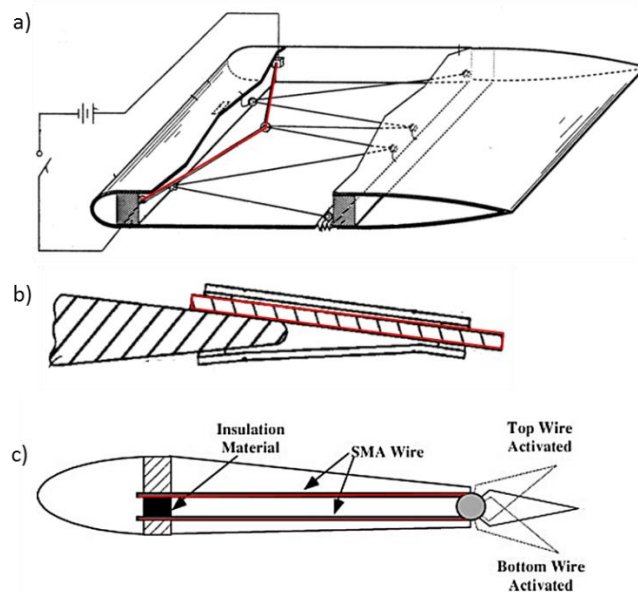


Figure 21 – SMA employment in blade profiles to shift trailing edge: a) SMA wire (red line) contracts pulling the mechanism. Adapted from Rogling and Hanagud (1991); b) SMA actuator in trailing edge (red line) changing the shape when activated. Adapted from Mckillip and NJ (1997); c) Tab actuator embedded within an airfoil section. Adapted from Epps and Chopra (2004).

Other solutions employs SMA in spanwise to change blade twist or perform stiffness changing as depicted in Figure 22. Yet, these examples exhibit SMA as an actuator. The superelastic effect for IBC control in helicopter is not observed. SMA as damper is applicable in diverse circumstances as mentioned so that suggest its use in IBC system to suppress vibration.

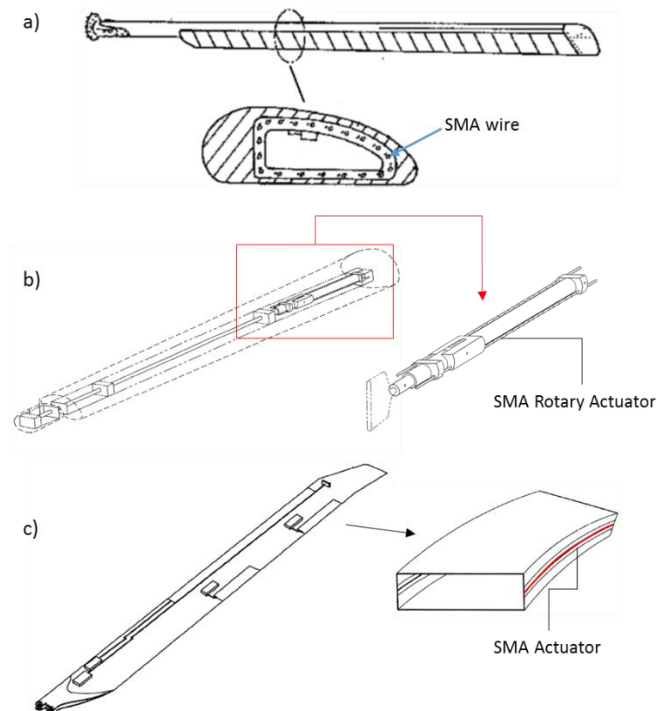


Figure 22 – SMA actuator proposals for blade application: a) Vibration damping device by rigidity changing. Adapted from Yujiro Shirai (1994); b) blade twist system. Adapted from Kent, Kirkland and Auburn (2006); c) blade twist and morphing shape. Adapted from (MADSEN *et al.*, 2013)

3.3 Hysteretic damping analysis

A mechanical system, undergo free or forced vibration (with external forces) is modeled as a mechanical arrangement compounded by a mass, spring, and a damper. The system of a helicopter blade and its supports is modeled as a cantilever beam with stiffness (where k the stiffness constant) and damping (where c is the damping constant) subjected to external force F .

Damper is the element that dissipate mechanical energy, so that damping systems movements die out due damping effect. The damper should be classified in subcategories according to the energy dissipation. Viscous damping force is velocity dependent and can be expressed by $F = -c\dot{x}$, where $\dot{x}$ corresponds to the velocity. Coulomb damping is another damping subcategory, independent of displacement and velocity, only dependent by friction between surfaces. In other systems, is found neither viscous damping nor Coulomb damping, but is observed that the energy loss is independent of the velocity and frequency. In those systems, the energy dissipation is approximately proportional to the square of the amplitude. In this case, the damping is called hysteretic damping (RAO, 2011).

The hysteretic damping modeling, based on original viscous damping modeling, establishes similarities that allow comparing both damping. Figure 23 shown similarities between viscous and hysteretic damping with it respective force versus displacement response.

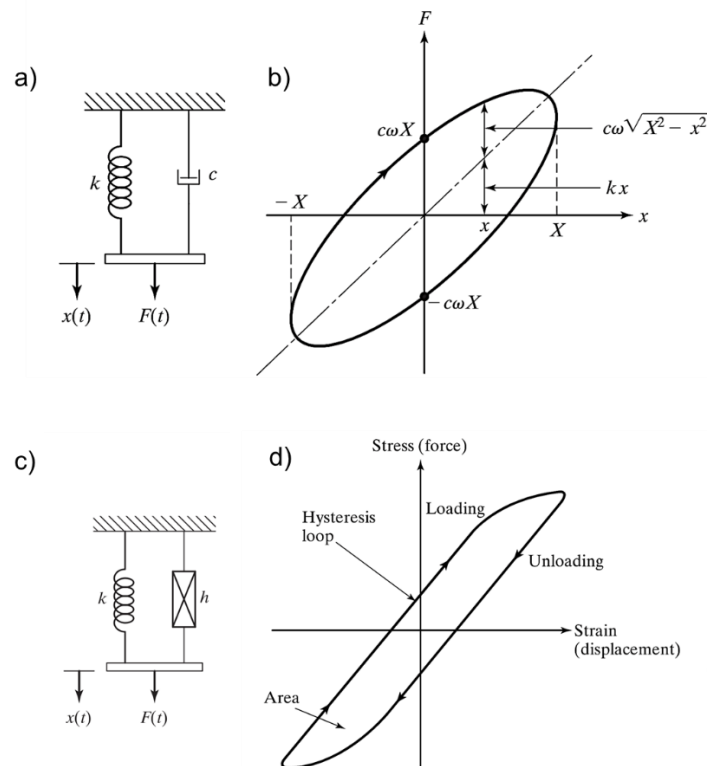


Figure 23- Similarities between viscous and hysteretic damping: a) viscous damping model; b) viscous damping force versus displacement response c) hysteretic damping model; d) hysteretic damping force versus displacement response. Adapted from Rao(2011).

Parameters plotted in Stress versus Strain (Figure 23d) corresponds to stress force according to the displacement, so that to viscous damping, the energy dissipated by a cycle of motion ΔW corresponds to the area inside the loop, given in Equation (12), where ω corresponds to the angular velocity, c the viscous damping constant and X the maximum displacement amplitude.

$$\Delta W = \pi \omega c X^2 \quad (12)$$

Figure 23d illustrates the hysteric damping behavior, where path due to loading is different to path due to unloading, named as hysteretic loop. Thus, the analysis induces a comparison to viscous damping. In addition, the literature reports the energy loss per cycle, due to internal friction in hysteretic damping. This loss is independent of the frequency but approximately proportional to the square of the amplitude, so that the energy dissipated by a cycle of motion in hysteretic damping is given by Equation (13) where constant h is named as hysteresis damping constant (MCCONNELL, 1995; RAO, Singiresu S., 2011; REIS, 2018).

$$\Delta W = \pi h X^2 \quad (13)$$

The relation between viscous and hysteretic damping in steady state is threatred in Mcconnnell (1995), summarized in Reis(2018) according to Table 4.

Table 4 – Relation between viscous and hysteretic damping in steady state. *Reis (2018)*

Damping	Dissipated energy by cycle	$c_{equivalent}$	$h_{equivalent}$
Viscous	$\pi c \omega X^2$	c	$c \omega$
Hysteretic	$\pi h X^2$	h/ω	h

Receptance, mobility and accelerance can be rewrite considering hysteretic damping instead of viscous damping, according to Table 5.

Table 5 - Mathematical representation of receptance, mobility and accelerance.

FRF parameter	Mathematical relations
Receptance	$\alpha(\omega) = \frac{X(\omega)}{F(\omega)}$
Mobility	$Y(\omega) = \frac{\dot{X}(\omega)}{F(\omega)}$
Accelerance	$A(\omega) = \frac{\ddot{X}(\omega)}{F(\omega)}$

The comparison is not limited for damping constants: considering the maximum amplitude of a dynamic response, depicted as a bode diagram (Figure 24), the loss factor and damping ratio for viscous and hysteretic damping, respectively, is summarized on Table 6. Thus, by bode diagram is possible to calculate damping ratio due to hysteretic damping.

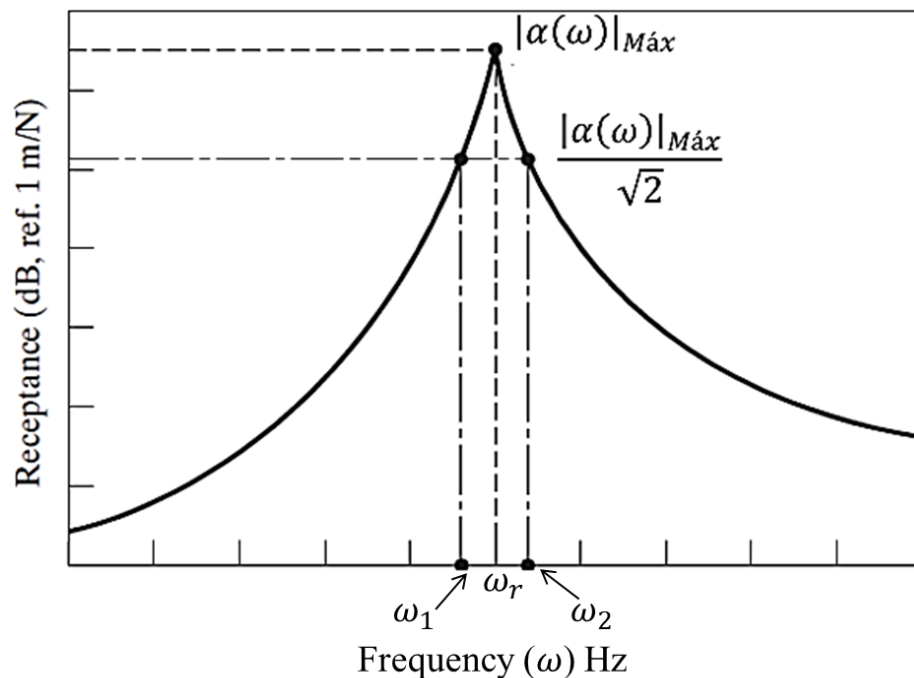


Figure 24 – Bode diagram showing resonance region: example of peak observing due to FRF analysis. Adapted from Rao(2011).

Table 6 – Damping and Loss factor formulas

Damping ratio	Loss factor
$\zeta = \frac{\omega_2 - \omega_1}{2\omega_r}$	$\eta = \frac{\omega_2^2 - \omega_1^2}{2\omega_r^2}$

In addition, the modal analysis can be conducted by a Nyquist diagram or Nyquist circle, a one-dimensional diagram plots real and imaginary parts of the FRF of a single-degree-of-freedom system (EWINS, 2000; RAO, 2011; REIS, 2018). Reis(2018) apply a Nyquist circle, depicted in Figure 25 to estimate the damping of Shape Memory Alloys micro springs.

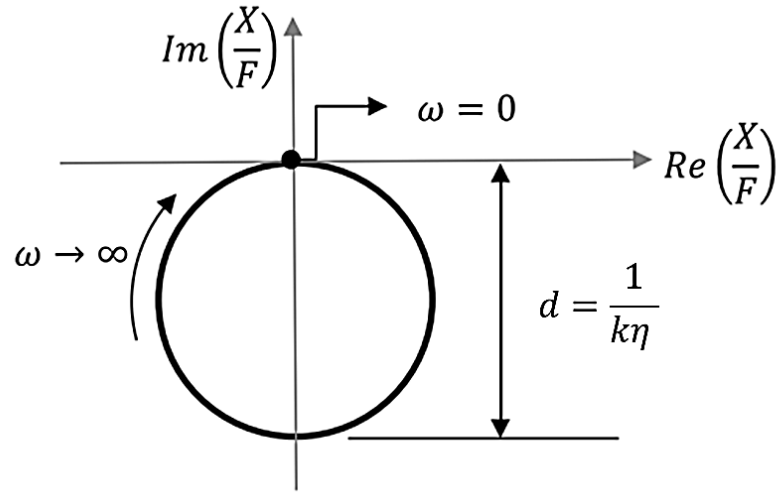


Figure 25- Nyquist circle circle for receptance. Adapted from Reis(2018)

The half-power points in this graph take place at two end of horizontal diameter of the circle, so that damping is directed related to Nyquist circle diameter. In Nyquist circle of Figure 25 the diameter corresponds to the inverse of hysteretic damping.

4 Methodology

In order to study the vibrations phenomena in helicopters rotor, after theoretical research and inspired in previews facilities, it is possible to build a customized whirl tower to vibrations measurements and dynamic characterization, composed of the most common parts available in real helicopters. Thus was designed a prototype with dimensions and features compatible with the Laboratory of Vibration and Instrumentation's of Federal University of Campina Grande (LVI-UFCG). The flowchart is depicted in Figure 26.

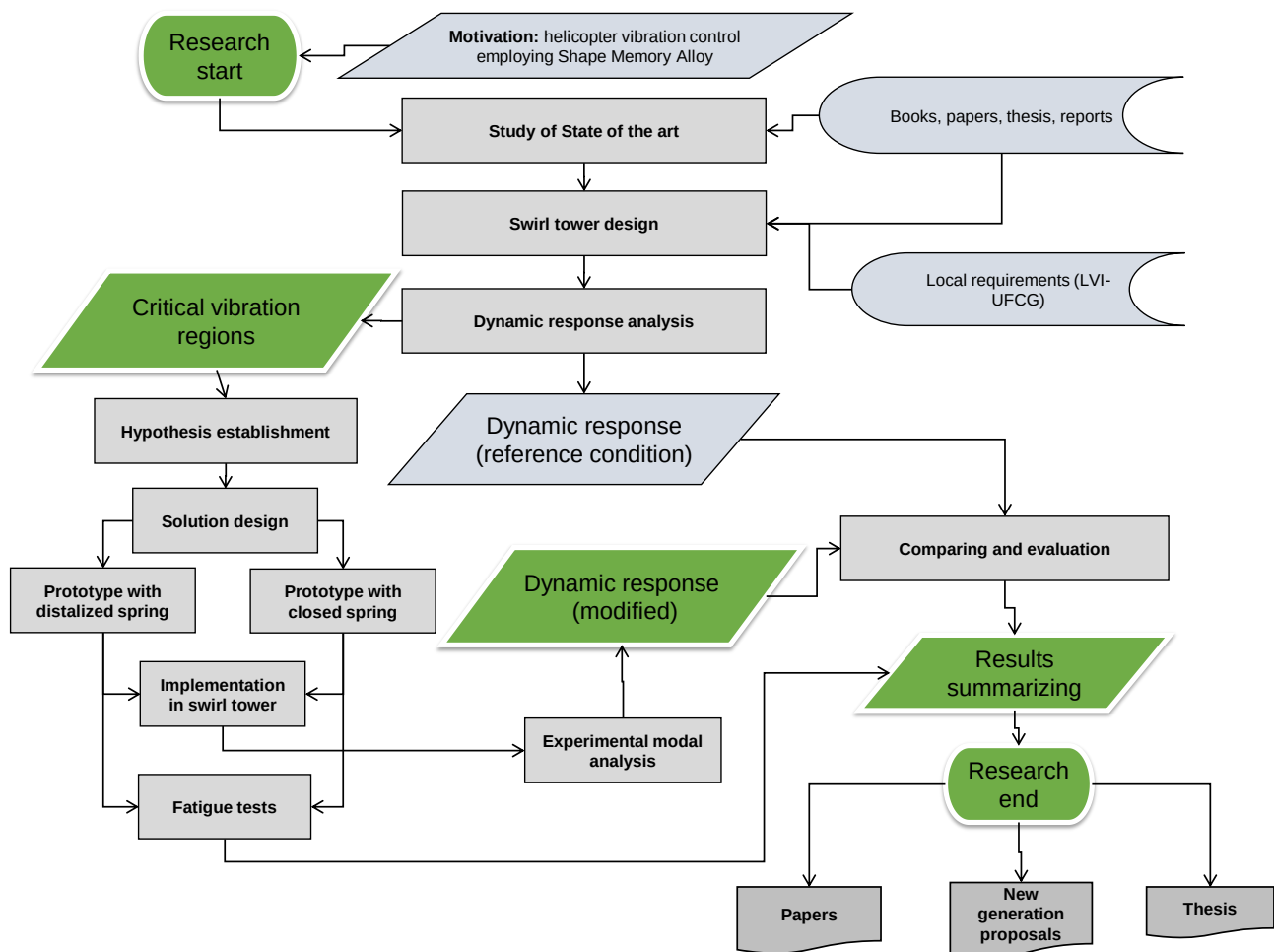


Figure 26 - Flowchart research

4.1 Whirl tower design

Whirl tower design follow the project requirements concerning viable installation at LVI. Figure 27 detail the whirl tower flowchart design, where the project guidelines represent the first step witch source the basic requirements for each prototype subsystem.

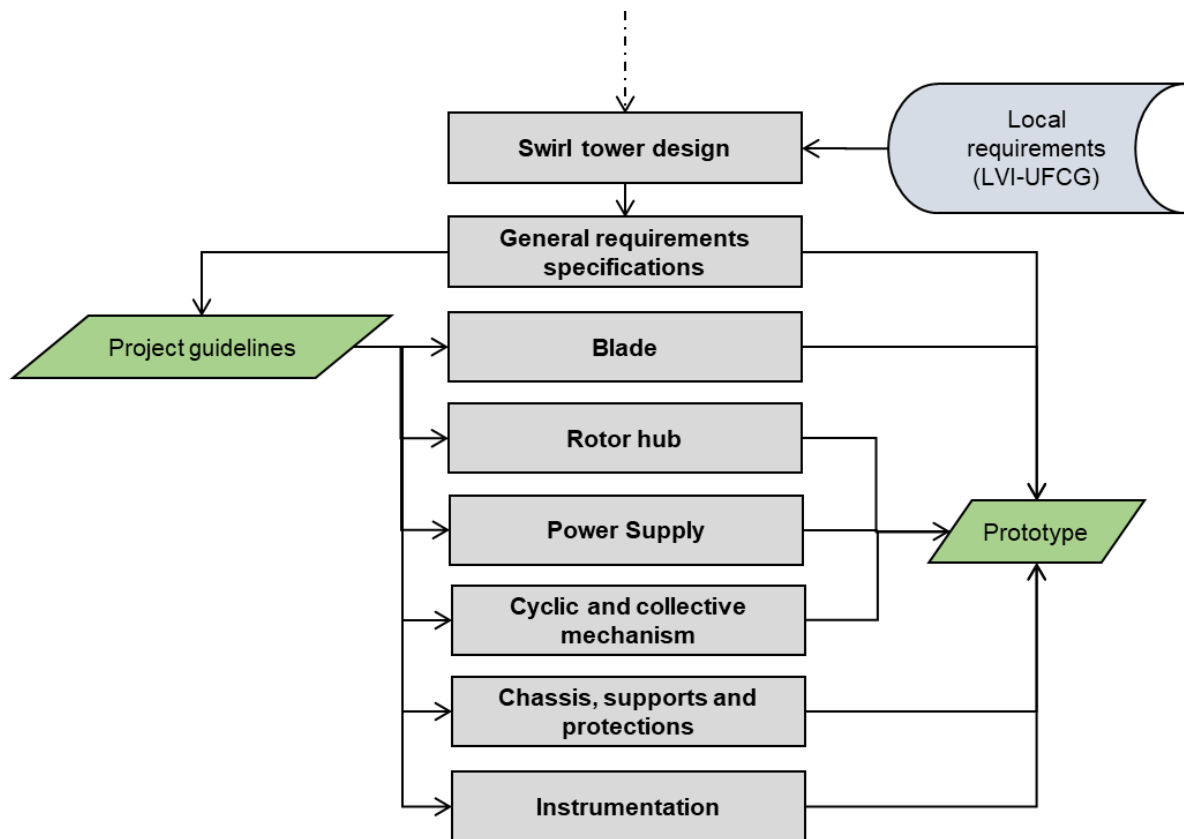


Figure 27 – Prototype design flowchart

The whirl tower project output is the prototype commissioned to dynamic response analysis, according to flowchart in Figure 26.

4.1.1 General requirements

The design of whirl tower carrying on under general requirements, bordered for LVI resources and parameters, focused on dynamic characterization, as follow:

- Compatible scale the laboratory available space which allows installation indoor;
- Capable to represent basic and generic rotor hub mechanisms;
- Easy assembly
- Easy transportation;
- Size compatible with laboratory instrumentation;
- Influence of ground effect has to be negligible.

Among of these characteristics, prototype size is a concern, because the size of prototype should match feasible instrumentation and maintainability. Based on indoor prototypes it is possible to estimate the general components and available physical space for whirl tower, as depicted in Figure 28.

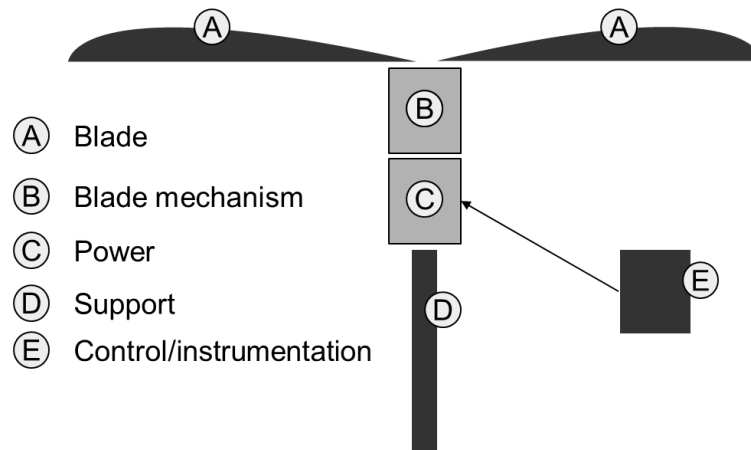


Figure 28- LVI whirl rotor scheme

The prototype exhibits the main rotor with blades (A), hub head, and swashplate mechanism (B). Power supply (C) without gearboxes, directly coupled with an electrical motor on the shaft. At least, no gearboxes avoid high frequencies components. The framework (D) should positioning rotor at a height from the ground enough to minimize the ground effect, and obstacle interactions. For this purpose, the rotor is positioned 1 meter above the ground reference. The ground reference is an inertial table, where the support has to be crimped.

4.1.2 Blade

The frequency harmonics are directly related to the number of the blades times rotational speed. Besides, the number of the blades affects directly the dynamic response of the machine (BRAMWELL; DONE; BALMFORD, 2001). In order to induce the worst vibration scenario, focused on highlighting vibrations effects, is adopted two-bladed rotor.

Once defined the number of the blades was specified a commercial shape to reduce the demand of building parts. The blades were built by additive manufacturing, in PLA(Polyacetic acid) as depicted in Figure 29, with 49grams, 442mm length, 42 chord and 6mm thickness, each one. Airfoil profile is NACA0012, according dimensions obtained in Rosata (2015). Although the geometrical characteristics of the blade are based on a commercial model, due to PLA, the prototype blade exhibits particular dynamic behavior.

The rapid prototyping was feasible by the Digital Manufacturing Lab of Federal University of Paraiba (FabLab-UFPB), which produced the components in four separated parts, for each blade, due to maximum size admitted by the printer. The total work spent amount of four hours. The parts were bonded applying glue based on cyanoacrylate. The surface finishing, based on Lima (2015), was made with acetone to reduce surface roughness.

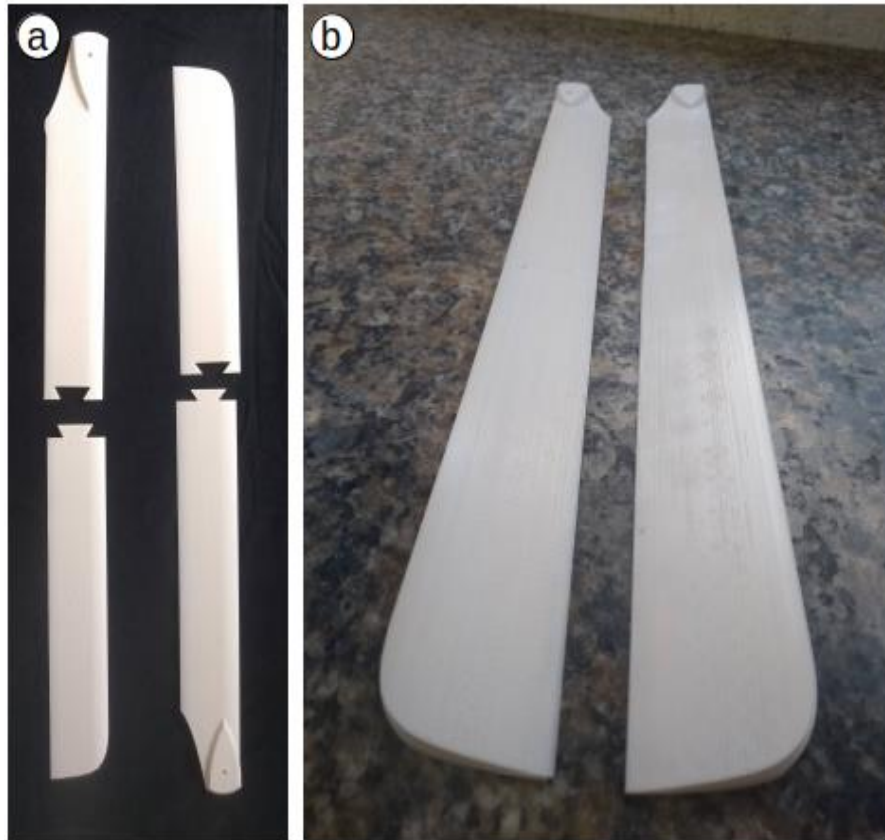


Figure 29 - Blades: a) segments after additive manufacturing b) available to assembly in the rotor.

4.1.3 Rotor Hub

The rotor hub has to represent a common helicopter mechanism with blade hinge and mechanisms to feasible pitch angle shifting. It is capable of assembly with a swashplate mechanism to enable basic blade movements. Rotor hubs with these characteristics are composed of a large number of parts. Thus, this project employs a commercial model compatible with the small dimensions of the prototype. The applications adopts a kit of helicopter hobby T-Rex 500 of Align® manufacturer. The swashplate is from same manufacture, originally used in T-Rex 450. Figure 30 shows the rotor hub and swashplate.

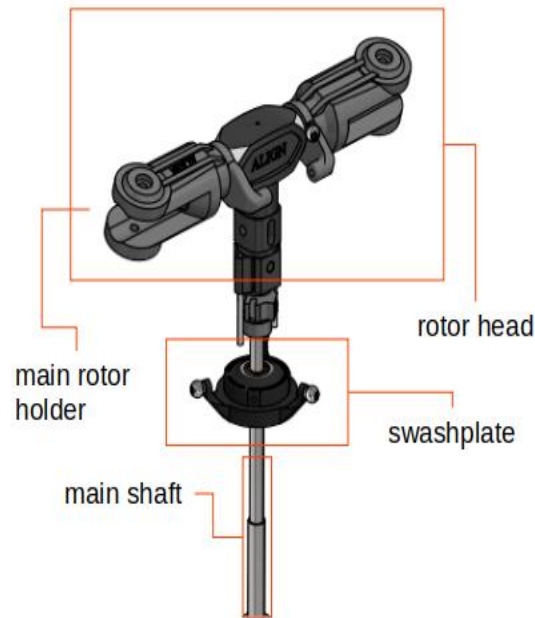


Figure 30- Rotor hub

4.1.4 Power supply

Power supply adopts an electrical AC motor WEG® model 63, 380Vac, 60Hz, 1/2cv, controlled by a Frequency Inverter WEG®, model CFW-10 to allows speed rotating control. The maximum rotation motor frequency is 60Hz, yet, the maximum main shaft rotation frequency is limited by the general structure of the whirl tower, links, mechanisms, and blade. Experimental tests demonstrate the maximum frequency of 12Hz to ensure tests with safety.

4.1.5 Cyclic and collective mechanism

To allow rotor maneuvering, due to blade feathering, the main rotor counts with the swashplate and electrical servos, controlled by-wire. Servos transmit the movements from static frame to rotating components through the swashplate. A simplified representation of this mechanism is depicted in Figure 31. Each blade can change its pitch angle (α) from zero to 12 degrees for T-Rex Main Rotor model.

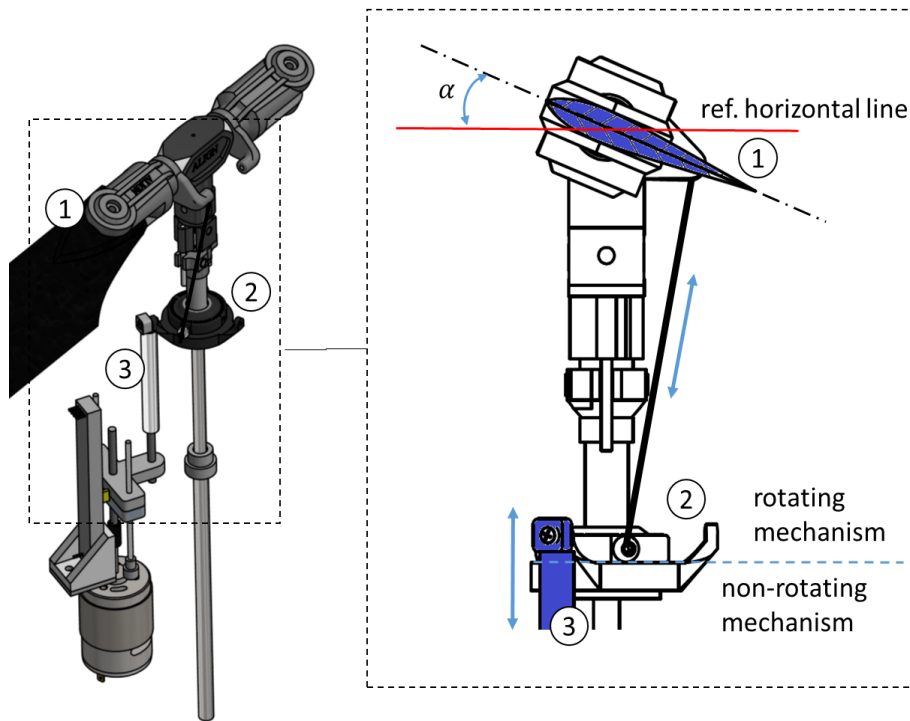


Figure 31- Blade mechanism: 1 – blade; 2 – swashplate; 3 – servo rod.

The rotor hub of prototype belongs to commercial helimodel T-Rex 500. This blade mechanism is characterized and detailed in Rosata(2015), which kinematic analysis grounded the scheme of the prototype depicted in Figure 32b.

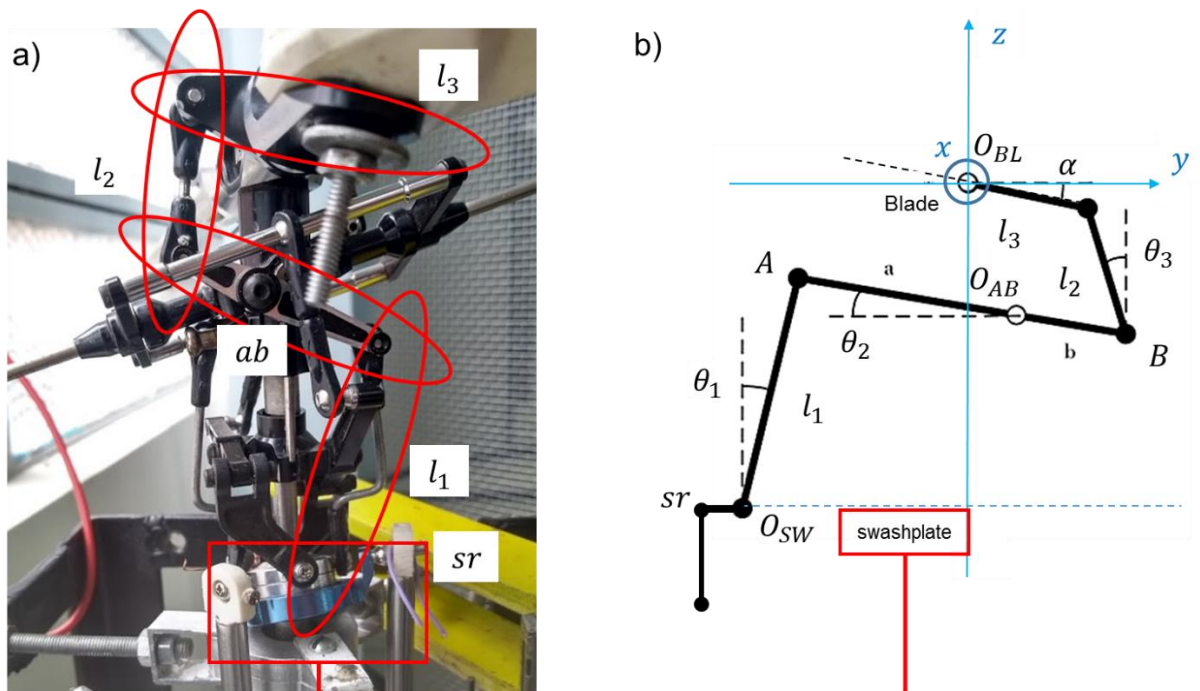


Figure 32 – Blade mechanism detail and cinematic diagram: a) mechanism links tags of prototype b) kinematic diagram, adapted from Rosata (2015).

There are three center in the mechanism: O_{SW} (swashplate linkage ball), O_{AB} (flybar mechanism movement) and O_{BL} (blade root). The xyz coordinates of O_{BL} are constant because O_{BL} is the rotational pin attached on the main shaft end in the rotor rub, with one degree of freedom, which corresponds to blade pitch angle (α) shifting. O_{BL} is the origin of xyz. The positioning of l_1 is different of other linkages because the attachment in linkage ball of the swashplate is 90° positioned opposite connection in B, as detailed in Figure 33.

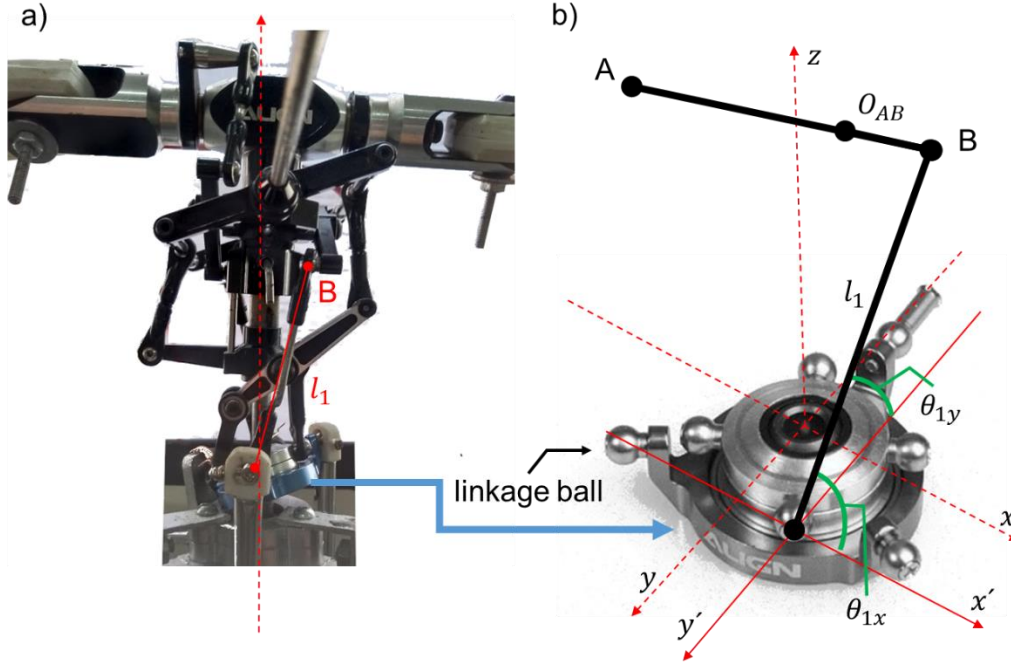


Figure 33 – Linkage l_1 positioning: a) linkage highlighted in rotor hub prototype; b) swashplate model HS1111T-84 applied in the prototype with l_1 attachment scheme.

Based in the kinematic analysis performed by Rosata (2015), considering each link as a rigid body, the kinematic relations of blade mechanisms is modeled as Equation (14).

$$y_B = l_3 \cos \alpha + l_2 \sin \theta_3 = y_{O_{AB}} + b \cos \alpha \quad (14a)$$

$$z_B = z_{O_{BL}} - l_3 \sin \alpha - l_2 \cos \theta_3 = z_{O_{AB}} - b \sin \alpha \quad (14b)$$

$$y_A = y_{O_{AB}} - a \cos \theta_2 = y_{SW} + l_1 \cos \theta_{1x} \sin \theta_{1y} \quad (14c)$$

$$z_A = z_{O_{AB}} + a \sin \theta_2 = z_{SW} + l_1 \cos \theta_{1x} \sin \theta_{1y} \quad (14d)$$

$$x_A = x_{SW} - l_1 \sin \theta_{1x} \quad (14e)$$

$$z_{sr} = z_{SW} (y_{sr} / y_{SW}) \quad (14f)$$

The vertical position (z) of O_{AB} depends on flybar mechanism (Figure 34), which kinematics approach is detailed in Rosata (2015). Yet, for elementary analysis, assuming flybar angle constant and parallel to the ground in hovering flight. Thus, the position of O_{AB} is constant.

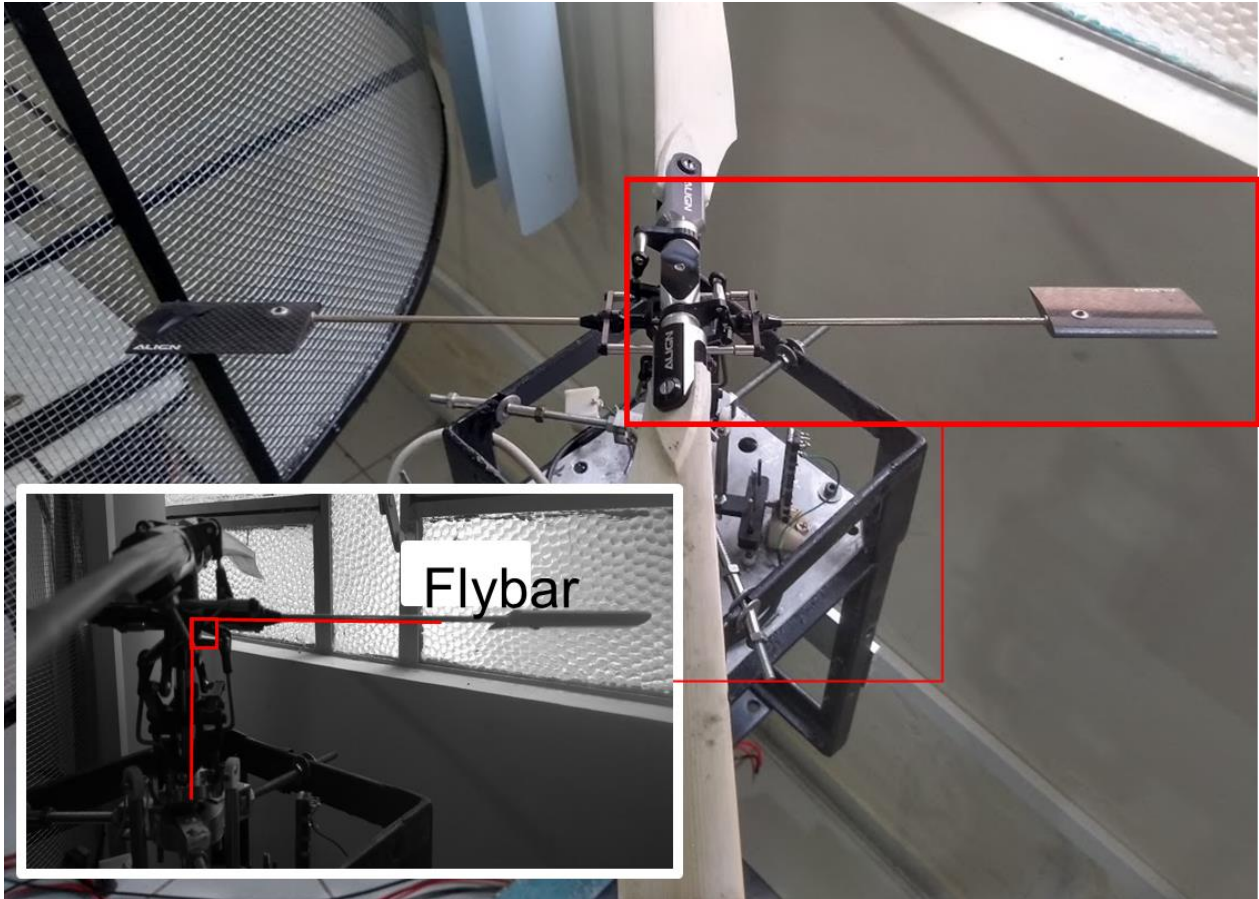


Figure 34 – Flybar – detail and general view of flybar position in hovering

Table 7 contains the dimensions and positions of elements depicted in Figure 32 and Equation (14).

Table 7 – List of parameters of blade mechanism according to the tag in Figure 32

Parameter	Value	Description
l_1	49.50mm	Length of rod l_1
l_2	25.30mm	Length of rod l_2
l_3	17.00mm	Length of rod l_3
a	25.20mm	Length of arm a
b	13.80mm	Length of arm b
y_{sr}	-21.00mm	Position of segment srO_{sw} (servo rod and swashplate linkage ball)
α	$0 - 12^\circ$	Blade angle of attack
θ_1		Angle of rod l_1
θ_2		Angle of segment AB
θ_3		Angle of rod l_2
O_{BL}	0	Position of O_{BL} in x, y, z
$z_{O_{AB}}$	-22.80mm	Position of O_{AB} in z axis
$y_{O_{AB}}$	7.00mm	Position of O_{AB} in axis
z_{SW}	-82.20mm	Position of SW in z axis
y_{SW}	-13.90mm	Position of SW in y axis

The vertical position of servo rod (z_{sr}) stem from the servo command which execute the vertical movement of swashplate (Figure 31), responsible to angle (α) shifting. In Equation (15) the position z_{SW} is calculated with α angle assuming $z_{O_{AB}}$, $y_{O_{AB}}$ and y_{SW} as a constant. The mathematical demonstration to Equation (15) is detailed in Appendix C.

$$z_{SW} = z_{O_{AB}} + a \sin \alpha - y_{O_{AB}} + a \cos \alpha + y_{SW} \quad (15)$$

The servomechanism control carried out by a control system compounded of drive board and control board (Figure 35a), from a National Control Board DAQ6210, 16 bits, 250Ks/s (Figure 35b). The drive board allows to control servo motors speed (by a Pulse With Modulation - PWM) and its rotating direction. The Labview interface commands each of three servos, independently (cyclic command) or simultaneously (collective command). The schematic of the drive board is depicted in Appendix A.

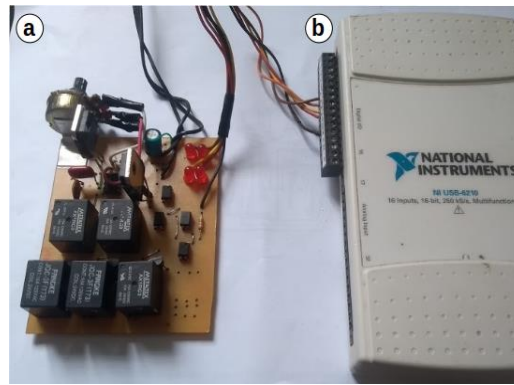


Figure 35- a) drive board b) Control Board DAQ6210

The interface allows to setup up the swashplate position and establishes a set point for each experiment, so that the previous position may recover to the next experiment. Figure 36a shows the scheme of servo control.

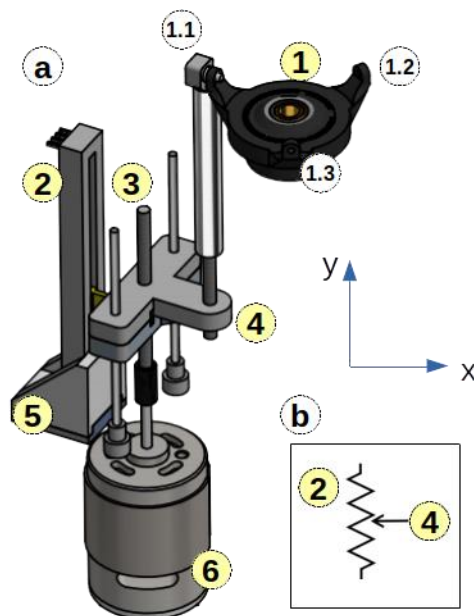


Figure 36- Servo Mechanism with its parts tagged

The tags of each listed parts, follow:

- 1 Swashplate;
- 1.1 to 1.3 Swashplate linkage point with cursor;
- 2 Potentiometer;
- 3 Screw bar;
- 4 Cursor;
- 5 Potentiometer support;
- 6 Electrical 24V DC motor.

The cursor movement towards y-direction. The displacement of each motor is measured by a linear potentiometer (Figure 36a, item 2), which communicates to DAQ6210 board the current vertical position of the cursor (Figure 36a, item 4). The correspondent electrical scheme of the potentiometer and cursor, as a transducer between electrical resistance and position, is depicted in (Figure 36b).

4.1.6 Chassis, supports and protections

Chassis has to support all the mechanisms, composed of two parts: tower and frame. It is made of iron with welding joints, except in the flange between frame and tower. A metallic net is positioned around the prototype to protect the environment against losing parts during rotation, which causes damages to humans or equipment. The ground effect is a concern, studied in Chereseman *et al.* (1957), so that the minimum height of tower has considered. In Figure 37 Rosata(2015) shows the Cheeseman and Bennett curve for the rotor of T-Rex 500.

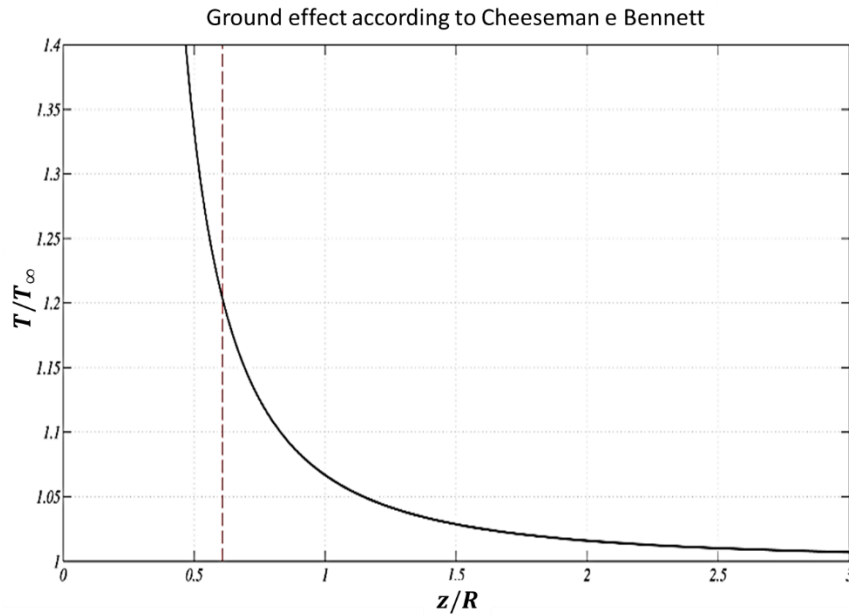


Figure 37 - Cheeseman and Bennett analytic result for ground effect; red dashed line represents the minimum ratio z/R for the T-REX 500. (ROSATA, 2015)

The curve consists of a ground constant $K=1$ where the dashed line indicates the minimum z/R possible for this rotorcraft. Taking account, the height limit in order to avoid ground effect, considering z as height of the whirl tower and R radius of the rotor, both in millimeters, was adopted 1080mm height. The result of Equation (16) demonstrates that the ground effect (T/T_∞) is negligible when analyzed in Figure 37.

$$\frac{z}{R} = \frac{1080 \text{ mm}}{485 \text{ mm}} = 2.22 \quad (16)$$

4.1.7 Instrumentation

Figure 38 represents the instrumentation diagram. The instrumentation allows real time measurements of rotating speed, vibrations signal and swashplate control.

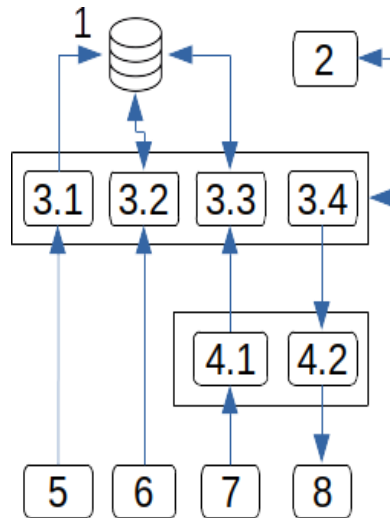


Figure 38- Instrumentation Diagram

The following parts compounds the diagram:

- 1 Database;
- 2 Control interface;
- 3 DAQ6210 input/outputs;
- 4 Driver board;
- 5 Accelerometer;
- 6 Rotation sensor;
- 7 Potentiometer
- 8 PWM motor supply.

Database is composed of a text file, which record in real time, the numerical position of swashplate set points, rotational speed and acceleration. Accelerations measurements are post-processed using Matlab aiming vibration analysis

Labview 2013 64-bit package supports the Control Interface, which graphical appearance is showed in Figure 39. The diagram of Labview application is depicted in Appendix B. Control interface allows to controlling each swashplate actuator independently or simultaneously, similar to cyclic and collective controls inspired on a real helicopter. The setpoint of each actuator can be adjusted as a set point. In addition, the acceleration values are showed on the screen, in graphic mode. During execution, the software records all data in the database file.

DAQ6210 (Figure 38, item 3) is capable to connect the accelerometer (Figure 38, item 5) directly in it analog input (Figure 38, item 3.1). To ensure the right value of acceleration, the package (accelerometer + interface) was calibrated by the portable modal shaker calibrator model PCB394C06.

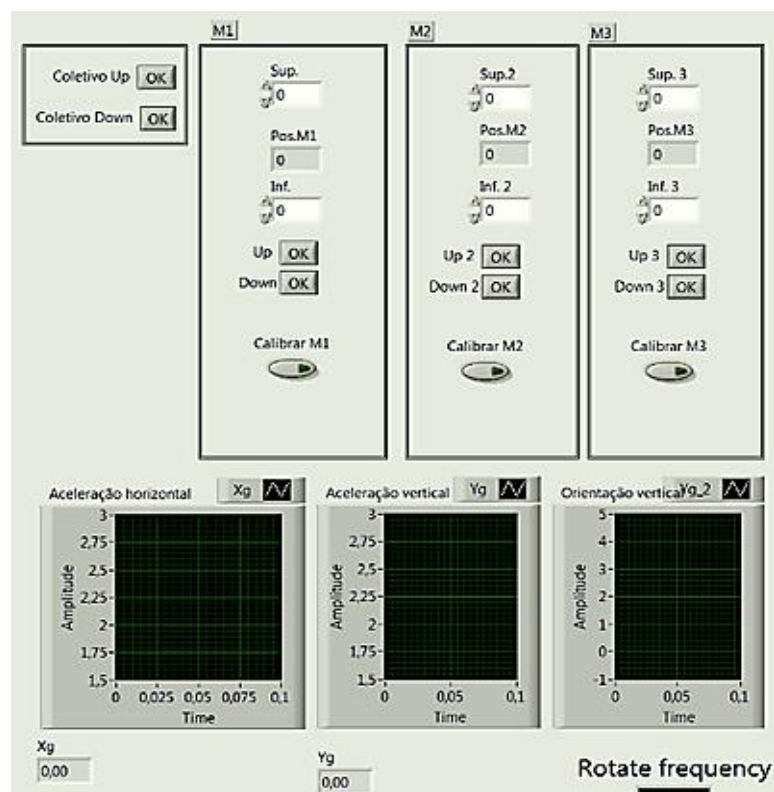


Figure 39- Labview Control Interface

The acceleration is measured by IC ADXL202 (Figure 38, item 5), which provides two-axis acceleration measurements (DEVICES, 1999). External 5V DC supply is provided for IC and the configuration pins is according IC datasheet default diagram. ADXL202 is installed directly on the measuring place. Figure 40 depicts the accelerometer, installed in the structure, which details shown the correspondent axis X and Y, labeled as components in direction X (A_x) and Y (A_y).

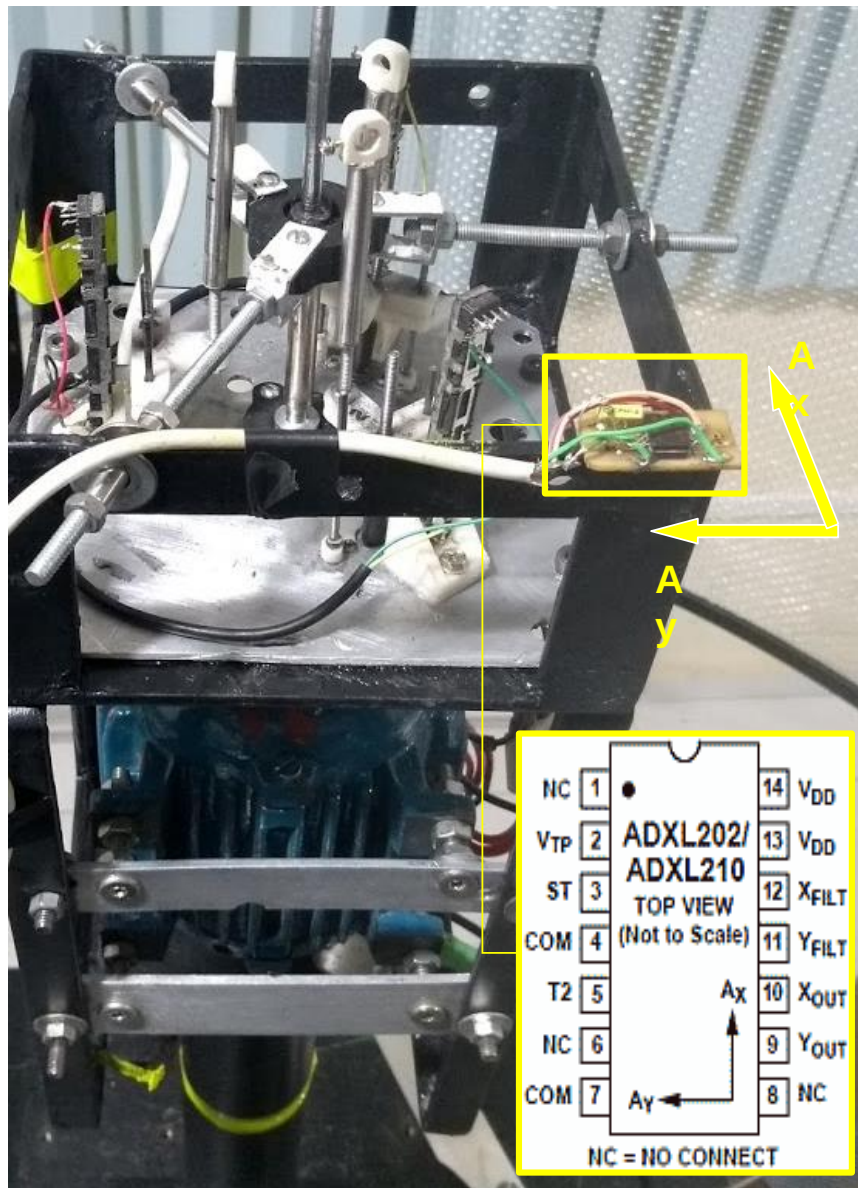


Figure 40- Accelerometer setup - axis x and y

The digital outputs of DAQ6210 board control each DC motor actuators (Figure 38, item 8), through the driver board (Figure 38, item 4). DAQ6210 is not capable to excite 24V DC motor directly due to power limitation. Thus, the driver board boosts the commands signal from DAQ6210 towards exciting the DC actuators motors. Another function of the driver board is to control the speed of DC motors by a PWM (Pulse Width Modulation) (Figure 38, item 4.2) power supply so that the rotational speed of each motor depends on the pulse modulation provided by the PWM. The adjustment speed is manual, by the setting on the control board.

The driver board is also used as a signal conditioner of cursor (Figure 21b) to allow the measurement of the difference of electrical potential throughout the potentiometer, which corresponds to current cursor position (Figure 38, item 7).

The first phase of the project depends on the modal analysis of the overall helicopter prototype (with prefabricated parts, already available). The modal analysis helps to identify the vibration-wise critical regions, predominantly by the experimental method, supported by numerical simulation.

The results of the modal analysis will collaborate with the second research step concerning the definition of the technology of smart material that would be applied to the final prototype. Finally, the effectiveness of smart actuation solutions will be compared in systems with and without a smart material device.

4.2 Analysis of dynamic response

There are a number of experimental technics to describe the dynamic behavior of a system leads modal analysis. The dynamic response, experimentally measured is capable to generate a FRF (Frequency Response Function) which represents the ratio between input and output signals. Considering $F(\omega)$, $X(\omega)$, $\dot{X}(\omega)$ and $\ddot{X}(\omega)$ as input force, displacement, velocity and acceleration, respectively, the FRF is classified according to the Table 5.

4.2.1 Modal Analysis of assembled prototype

The experimental modal analysis of whirl tower determines the natural frequency of the structure, in order to help the functional tests and characterize the expected behavior of prototype and the individual contribution of main parts in dynamic response. The test was performed without the blades, which corresponds to the reduction of approximately 0.62% of whirl tower total mass. The experimental measurements of general structure are performed with an impact hammer, model PCB086C03L, one accelerometer PCB352B10 and one dynamic signal analyzer Agilent 35670A, according to the scheme of Figure 41. It assumed the modal frequencies relates to displacements perpendicular to axis z , the direction of the maximum tower length, thus the impacts intend to excite the structure in direction parallel to xy plane. The input force, from impact hammer was applied in 4 points for each measurement, in order to excite the longitudinal and torsional modes. For each point was performed five impacts with hammer, so that the results of each point correspond to the average of the five impacts. Due to geometric symmetry of whirl tower, it assumed the modal shape in the direction of x and y are equivalent. In Figure 41 illustrates the points of accelerometer positioning. The impact of the hammer took place in the position and direction of each imaginary dashed line, depicted in the scheme of Figure 41. Thus, the force was applied aligned with the accelerometer

positioning, excepted in point 3, due to the necessity of torsional mode inducing, promoted by the misalignment between the application force point (dashed line 3) and the accelerometer position (point 3 placed in top of the tower).

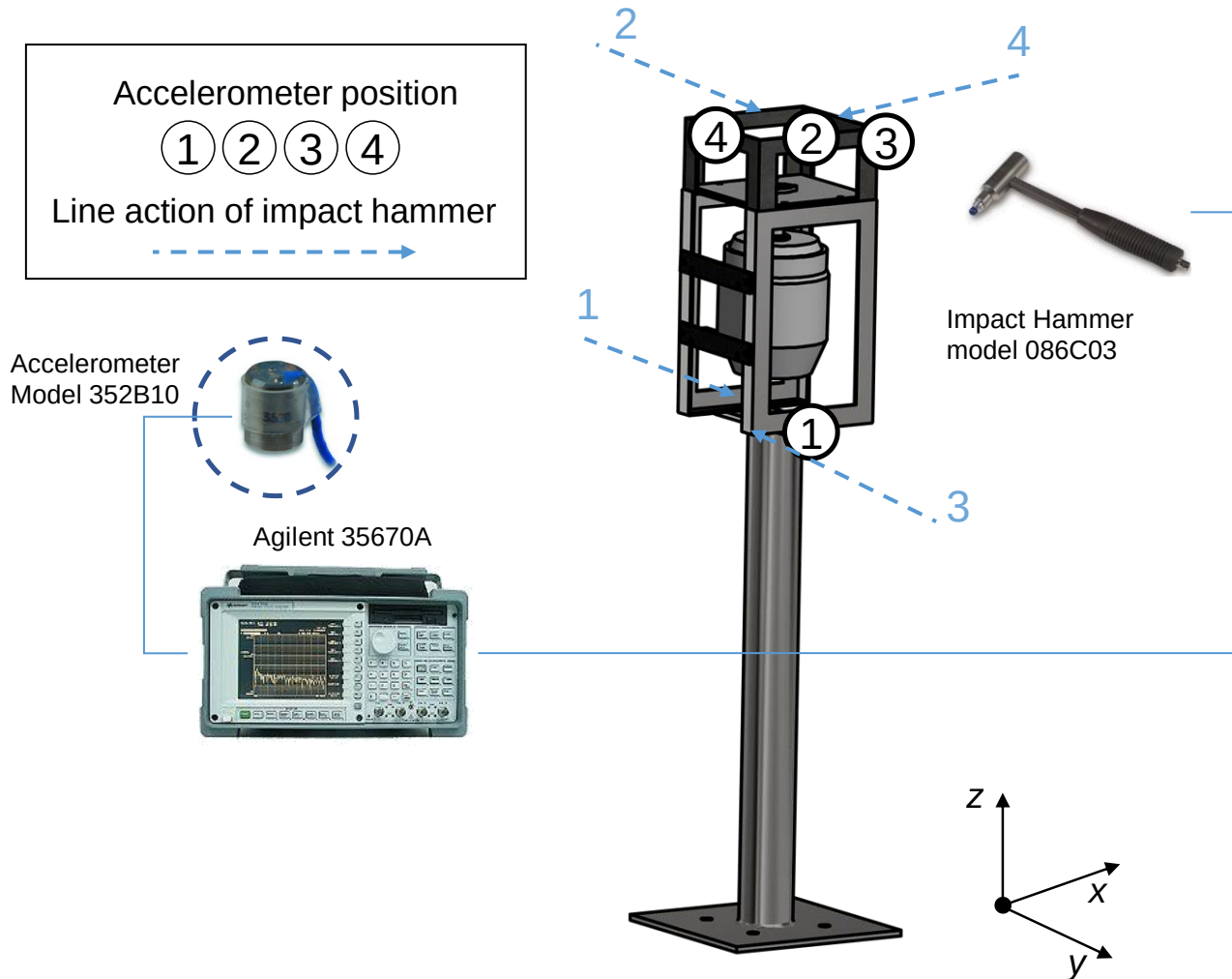


Figure 41 – Experimental modal analysis scheme: the bullets corresponds to the accelerometer installation point and hammer impact region, respectively.

The numerical modal analysis employs the finite element modeling Hyper Mesh® software, dedicated to model analysis simulation. For simulation, the 3D model designed in Autodesk Inventor, including tower, motor and rotor hub was imported to the platform. The Geometry file imported to Hyper Mesh meshed and constrained in the bottom springs of the Inertial Table. The numerical analysis was performed using a finite element model built in Hyper Mesh® and solve by Optstruct Altair software. The model has a total of 199.926 nodes and 723.602 elements, according to the element detailing of Table 8.

Table 8 – Mesh elements detailing of Hyper Mesh simulation

Element Type Information	Quantity	Mesh element description
CBUSH	4	1D spring
CPYRA	21	3D pyramid
CTRIA3	50	2D triangle
CBEAM	77	beam
CPENTA	706	3D pentagonal
CQUAD4	950	2D square
CHEXA	22468	3D hexagonal
CTETRA	699326	3D tetragonal

The material and proprieties of all parts: inertial table, main structure and attachment parts (brackets, bolts and welds) and total prototype mass (16.4kg) was adjusted in order to represent the prototype. The electrical motor geometry, rotor and bearings, was simplified and the material properties adjusted to represent the mass weighted (6.4kg). The simulation model was gradually improved adjusting the structural damping by the FRF curves performed at four different points of structure. As material, except the motor, was considered structural steel and the aluminum.

4.2.2 Blade Modal Analysis

An important parameter to overall spectral response comprehension is the blade natural frequency knowledge. The modal shape of the blades aims the critical rotor frequencies identification, performed in order to induce critical vibration on dynamic tests. The methodology of the blade modal analysis consists of performing numerical simulation supported by experimental validation in an individual blade, according to the milestones of Figure 42.

In order to run numerical simulation, CAD model, previously created by Autodesk Inventor, was imported into Ansys Workbench package to conduct the finite element analysis. It was considered blade mass, obtained experimentally, which crimped with cylindrical support, simulating the hub head hinge. For simulation was considered a mesh with 1.376.810 nodes and 827.755 elements.

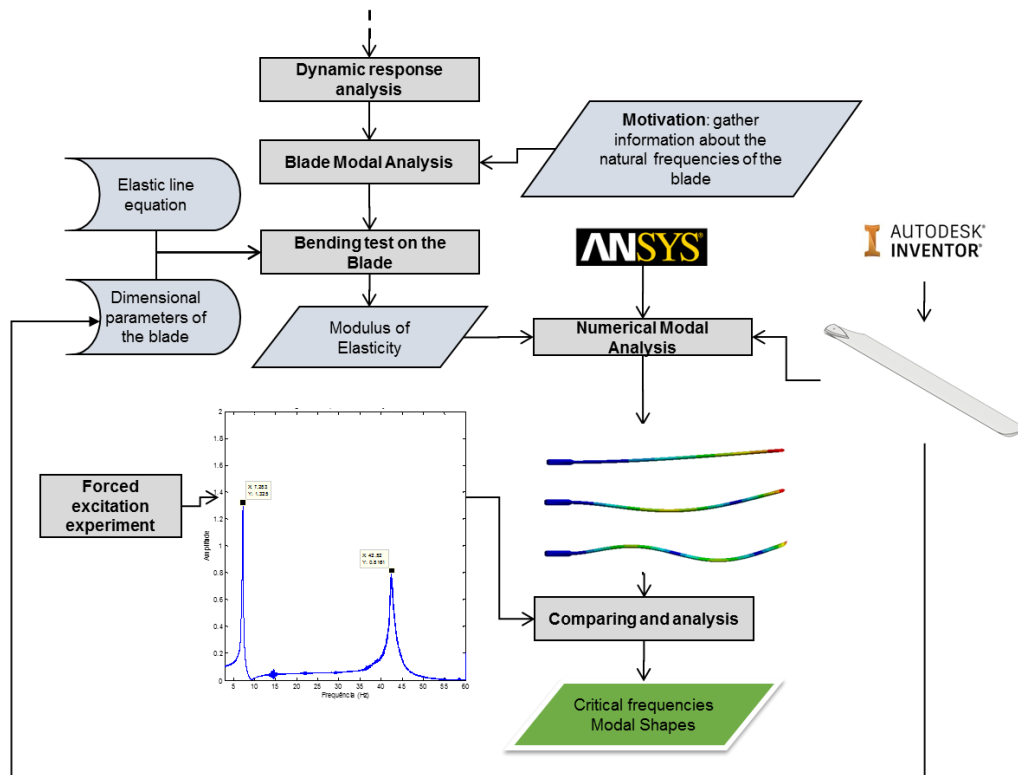


Figure 42 – Blade modal analysis methodology

Considering the Young modulus, to supply mechanical properties for simulation, initially intended to apply PLA literature mechanical properties. Yet there are a lot of variable manufacturing process that affects the mechanical properties of a PLA printed part, as described in (FERNANDES, 2016). Thus, carried out the methodology followed by Reis (2018), performing an experiment to obtain the approximated Young modulus of the blade, which scheme is depicted in Figure 43. The experiment consists in root blade clamping in a rigid support and applying a known force on the blade end to measure its displacement under flexion.

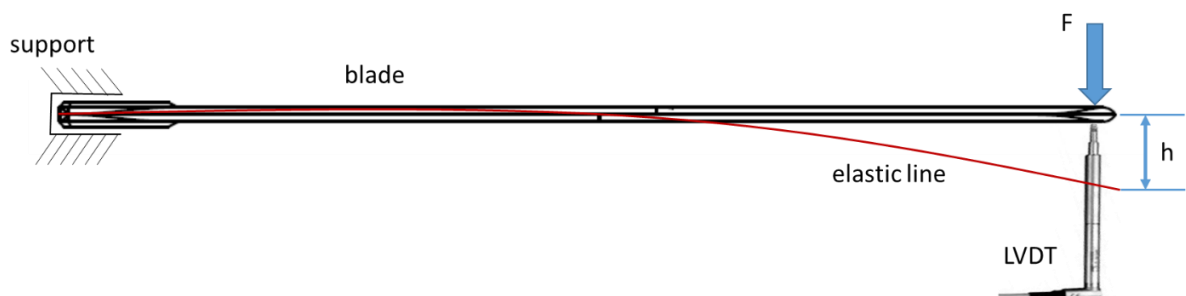


Figure 43 - Experiment for estimation of Young modulus of the blade

In order to measure the modal frequencies in blade span direction (flapping mode) was conducted an experiment depicted in Figure 44. The experiment consists of forced vibration with input force and output displacement using contactless sensor to avoid influence of probe mass. The

displacement sensor was installed near blade root, less 1/4 of total length, due to maximum displacement limit of sensor.

The acquisition data carried out with HBM MS840 module, coupled by voltage input with contactless sensor SKF CMSS665. Input force measurement was performed with load cell HBM U9C. The apparatus was attached in an inertial table. Post-processing data was executed in Matlab to perform Fast Fourier Transform analysis.

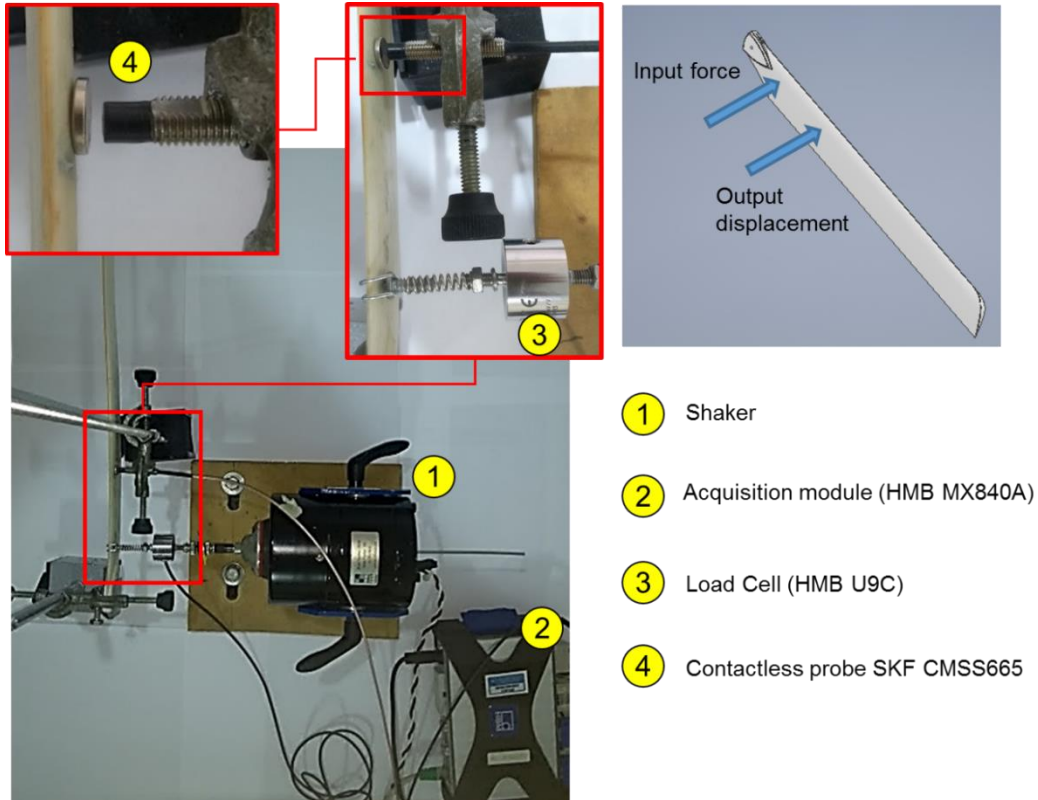


Figure 44 - Forced vibration experiment scheme

4.2.3 Analytical linkage displacement of blade mechanism

The Equation (17) defines the vertical displacement of servo in the linkage ball (Figure 33) taking into account the parameters from Table 7 and Equation (15).

$$z_{sr} = [38(\sin\alpha + \cos\alpha) - 65.98]mm \quad (17)$$

The vertical position of linkage ball induces pitch angle α shifting. Thus, due to cyclic movement, dz vary for each revolution of the blade, as a simplified scheme in Figure 45.

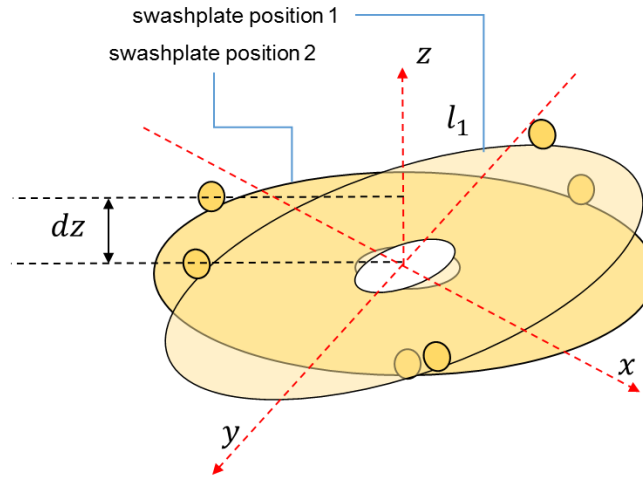


Figure 45 – Simplified scheme of swashplate position variation under cycling command.

Considering harmonic representation of pitch angle (α) for each revolution, $\frac{dz}{dt} \rightarrow \frac{d\alpha}{dt}$. Thus, for each revolution of main shaft, the variation of α can be modeled as Equation (18).

$$\frac{d\alpha}{dt} = \alpha_{max} \sin(\Omega t) \quad (18)$$

4.2.4 Linkage forces of blade mechanism - analytical and numeric approach

The static forces due to mechanism functioning and the dynamic force due to aerodynamic effects is depicted in Figure 46.

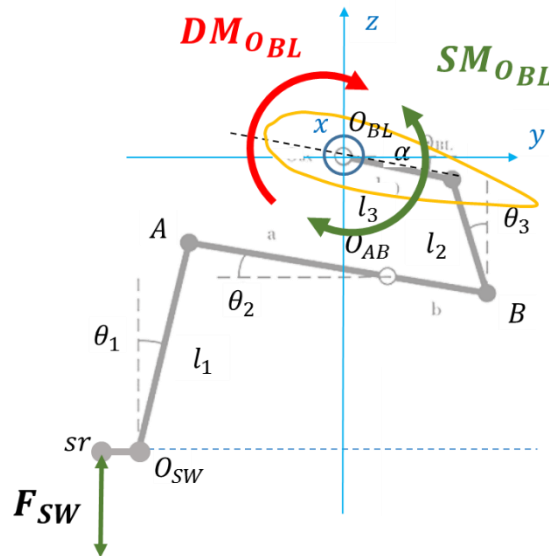


Figure 46 – External static force and moment of blade mechanism

In current operation, the static force (F_{SW}) applied in the linkage ball is transmitted through the mechanism to the rod l_3 , generating a moment $SM_{O_{BL}}$ in the blade root (point O_{BL}). Helped by

CAD simulation in Autodesk Inventor, with a 3D representation and the mechanism dimensions (Table 7) was obtained the maximum internal static moment $SM_{O_{BL}}$ of mechanism. Figure 46 illustrates, among others, the moment with double direction because the amplitude and direction of vector $SM_{O_{BL}}$ vary during each revolution of the main shaft, according to Equation (18).

The dynamic forces stem from aerodynamic effect of lifting and drag. Based on the blade element theory, with Equation (7), and considering as input data the whirl tower frequency and geometric parameters, it obtains the moment $DM_{O_{BL}}$ centered in O_{BL} , taking into account the airflow from leading edge to trailing edge of the blade. The Matlab code in Appendix E demonstrates how to calculate obtain the aerodynamic moment. The blade node O_{BL} is placed on the center of pressure of the profile NACA0024, so that the lifting force does not induce any moment in O_{BL} . Thus, the resistant force against F_{SW} corresponds to the internal mechanical resistance of mechanism and aerodynamic effect due to $DM_{O_{BL}}$.

4.2.5 Experimental linkage forces and displacement

In order to determine the internal forces in blade linkage is performed an experiment with whirl tower in severe tip path plane angle.

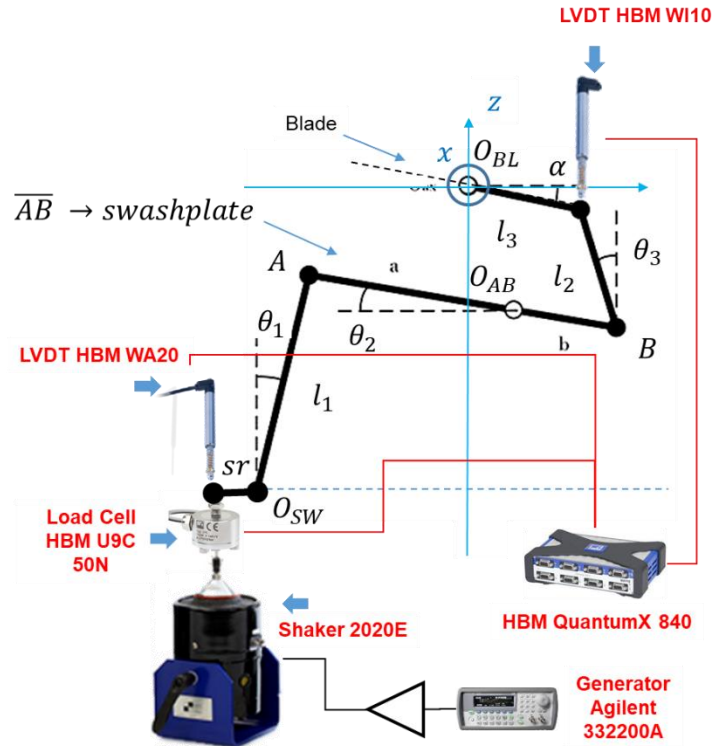


Figure 47 – Experimental scheme to measure input force and output displacement on blade feathering mechanism.

Figure 47 shows the experimental scheme to measure the input force applied under swashplate linkage ball (point O_{SW}) through a Load Cell. The output movement is measured by a

displacement sensor (LVDT) installed in the end of rod l_3 witch rotate de blade profile around origin O_{BL} . The Load Cell measure F_{SW} force, so that LVDT helps to obtain $SM_{O_{BL}}$.

The amplitude of internal force from the modal shaker is the minimum required to shift the attack angle α of a range from $(0 - 10^\circ)$. The frequency tested follow the range of $(1 - 12)\text{Hz}$ according to the limits of 4.1.4.

4.3 Shape memory alloy device requirements

The device design to vibration reduction has to follow a list of requirements, aligned with general objective, state of the art and scientific contribution, as follow:

- a) Interchangeable assembly between original parts and modified prototype;
- b) Commercial SMA geometry application;
- c) Fabrication on UFCG/UFPB facilities;
- d) Operation below the maximum frequency limitation of SMA in order to preserve the performance
- e) Design that allows further helicopter parts replacement in pre-existing models.

The following lines show each requirement itemized in the previous list.

4.3.1 Interchangeable assembly between stiff pitch link and SMA pitch link

Taking into account the original geometric space and clearance available of the stiff pitch link rod, the proposed device can replace the stiff pitch link without any modification to the mechanism. Thus, the proposed device preserves the stiff pitch link length and original attachment with eye bolt. The original pitch link assembly is depicted in Figure 48.

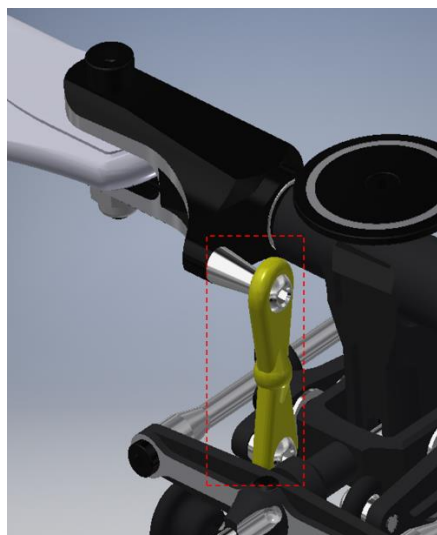


Figure 48 – stiff pitch link positioning

4.3.2 Commercial SMA micro coil springs - geometry application

The state-of-the-art demonstrate satisfactory results with commercial SMA-SE springs, originally employed in the orthodontic application as depicted in Figure 49. In orthodontics, these springs are employed to aid the teeth repositioning due to internal and constant forces when these devices are attached on the orthodontic device.

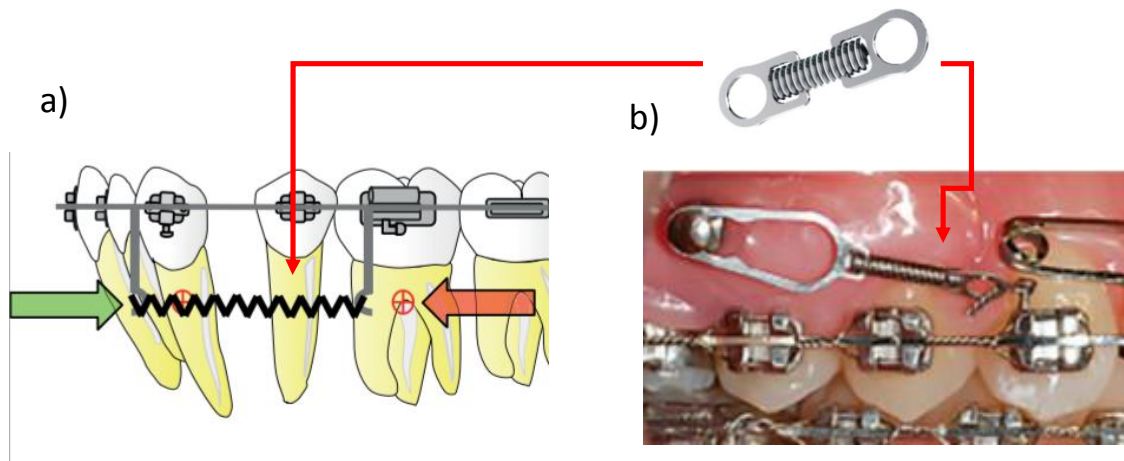


Figure 49 – Example of micro coil springs employment in orthodontics: a) illustration of internal constrain force performed by the spring installed in the teeth; b) example of clinical case with coil spring application. Adapted from Ribeiro et al.(2016)

In addition, the orthodontic coil springs SMA-SE has been applied in vibration control as illustrated by two examples of Figure 50.

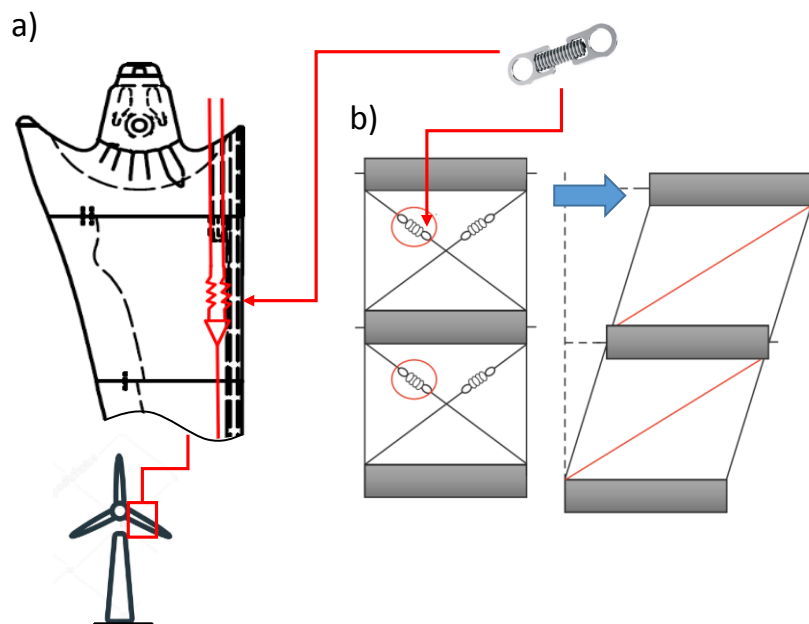


Figure 50 – Examples of vibration control SMA applications: a) micro coil SMA springs incorporated in blades structure of aerogenerator; b) micro coil springs works as damper in a prototype building structure. Adapted from Moraes et al (2018) and Reis(2018)

According to the two examples depicted in Figure 50, Reis(2018) employs SMA-SE micro-springs to passive vibration attenuation of aero generator blades (Figure 50a). Moraes et al (2018) to passive vibration control in a prototype structure that simulates a two-floor building (Figure 50b). Taking into account the interest in adapts mini coil springs for mechanical systems in other fields, besides orthodontics, Grassi *et al.* (2015) study the effect of heat treatments on the thermomechanical behavior in those SMA springs. The prototype dimensions and the experience of LVI/LAMMEA-UFCG with mini coil spring, support this research towards employs mini coil spring SMA-SE M9 standard so that increase the probability of success in a short time disregarding the spring fabrication.

Thus, is employed mini coil spring of Morelli manufacture which thermomechanical characterization was performed by DSC and quasistatic load test.

4.3.3 Fabrication at UFCG/UFPB facilities

The steps of SMA device manufacturing, since springs to final assembled device has to be implemented using UFCG/UFPB facilities considering the experimental apparatus available to design, characterization and test. The structure of smart pitch link is fabricated by additive manufacturing. The assembly of SMA springs and the structure is performed at LVI and all dynamic characterization.

4.3.4 Operation below the maximum frequency limitation of SMA in order to preserve the performance.

The time response frequency of SMA may not a concern, but need to be considered in the engineering applications. Machado *et al.*(2008), in an introduction about SMA even highlights that the good energy density of SMA actuators, compared with other functional materials, is no observed when this material is analyzed in frequency wise. The performance in frequency is practically the opposite: SMA frequency response tends to decrease as soon as frequency rise. Even though, in the passive application, Reis (2018) performs experiments with micro coils SMA springs. The results demonstrate a decrease in the peak energy dissipation close to 15Hz, operating as an energy dissipator, according to the results exhibited in Figure 51. Taking into account the use of the similar micro coil springs for this research, the strategy lies to study the span frequency of the prototype.

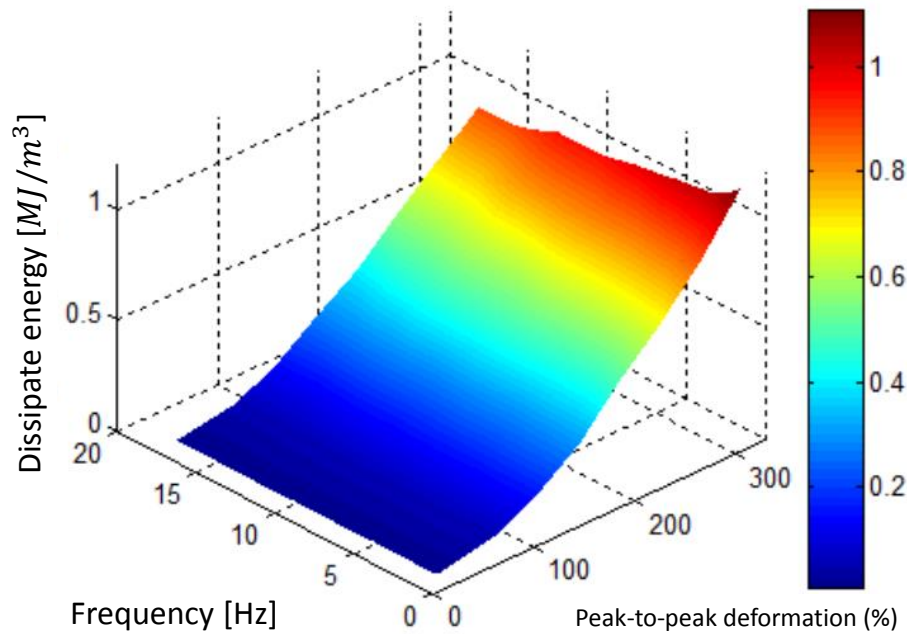


Figure 51 – Energy dissipation based on frequency and deformation amplitude of micro coil SMA springs. Adapted from Reis (2018)

Expanding for real helicopter rotors, the angular speed varies according to the rotor dimensions and tends to be reduced as soon as the rotor dimensions increases. Considering a large variety of vertical flight aircrafts, since real manned helicopters, helimodels, and drones the main shaft frequency, directly related by 1/rev parameter of the blades, depends on the aircraft model. Based in manned models as Esquilo AS-355F2, the main shaft frequency is approximately 6.5Hz. In helimodels as T-Rex 500, the main shaft frequency is 38Hz. Indeed, the main shaft frequency depends on rotor diameter and the required lift force. In SMA pitch link designed to helicopter application, the frequency range is an important parameter considering its dynamic performance (BRAMWELL *et al.*, 2001; DAMY, 2006; ROSATA, 2015).

Literature reports the superelastic effect in vibration attenuation depends more from stress than frequency. Reis(2018) demonstrates a reduction of 0.76dB of vibration energy dissipation due to frequency increasing from 5 to 15Hz, against a rising of 9.65dB due to a stress increasing of 300% with a mini coil SMA-SE. Above these limits, the rate of phase transformation does not reach satisfactory results. In the same direction, Rao (2015) recommends SMA applications in frequency below 10Hz. Lagoudas (2008) declare a relatively low-frequency response, until approximately 12Hz when SMA is compared with piezoelectric materials. Thus, the project highlights the limitation of SMA device in operational conditions bellow to maximum frequency limits recommended.

4.3.5 Design that allows further helicopter pitch link replacement in pre-existing models

The design should not affect the general rotor framework, so that can replace the stiff pitch links in pre-existing aircraft, which addresses a further commercial alternative to vibration reduction. In order words, the smart pinch link has to replace the original pitch link considering that, in further application aspects of safety must be considered. Aiming the concept test performance, for prototype installation the methodology not analyses the possible pitch link structural failure neither .

4.4 SMA device concept

Based on literature, especially on successful researches of Anusonti-Inthra(2000), Nitzsche(2015) and REIS(2018), was design a concept of device, which may replace the stiff pitch link (Figure 48) in order to reduce vibration due to transmissibility from blade root to swash plate mechanism. The device have to follow the requirements of section 4.3. The concept design of SMA pitch link is depicted in Figure 52.

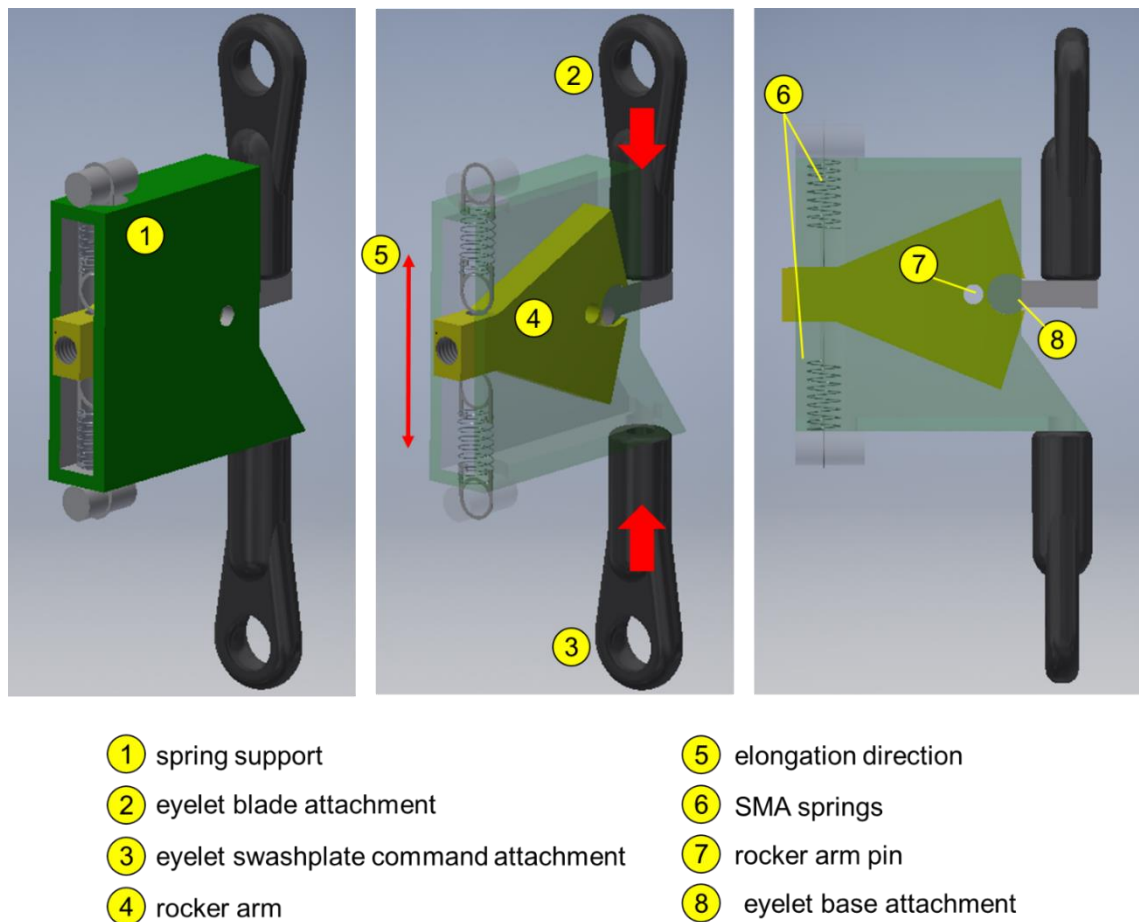


Figure 52 - SMA-SE device to replace stiff pitch link

The SMA device works as a “SMA pitch link” employing two SMA-SE orthodontic springs (NiTi spring, model M9), installed in opposition, pulling a rocker arm. The longitudinal and relative displacement between root blade attachment and swashplate command is transmitted to the rocker arm, which converts this movement in elongation of the SMA-SE springs. The elongation of springs induces phase transformation of SMA-SE, so that couple damping in the system. The SMA springs are commercial without any thermal treatment before installation.

The replacement of original pitch link to SMA-SE pitch link is illustrated in Figure 53 for a virtual helicopter model employing the rotor hub prototype.

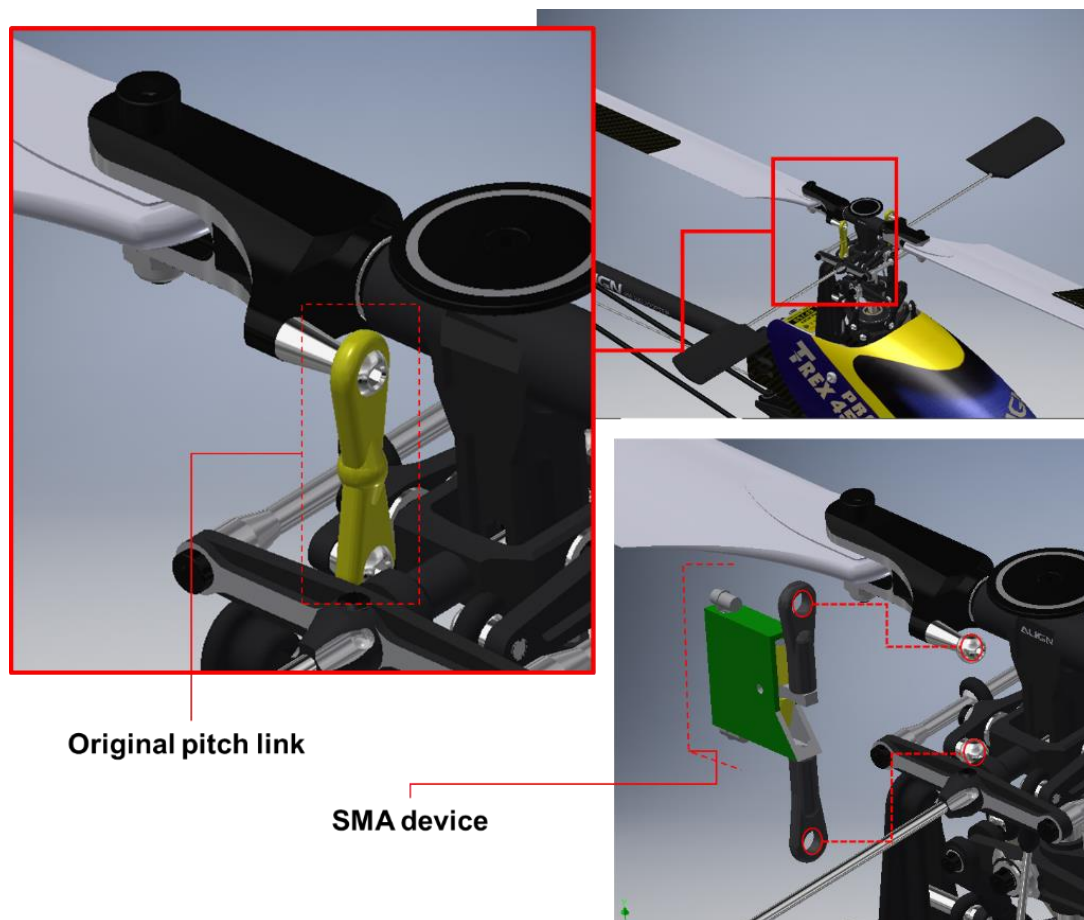


Figure 53 - Illustration of definitive positioning of SMA-SE pitch link

The SMA pitch link proposal has to mix two important and opposite requirements: ensure phase transformation of the material by stress and avoid large relative displacements between eyelets (points 2 and 3 in Figure 52). The solution was design a geometry with asymmetry in rocker arm pin position, so that amplify the small displacements of point 8 of Figure 52. Figure 54 depicts the evolution concept of Figure 52.

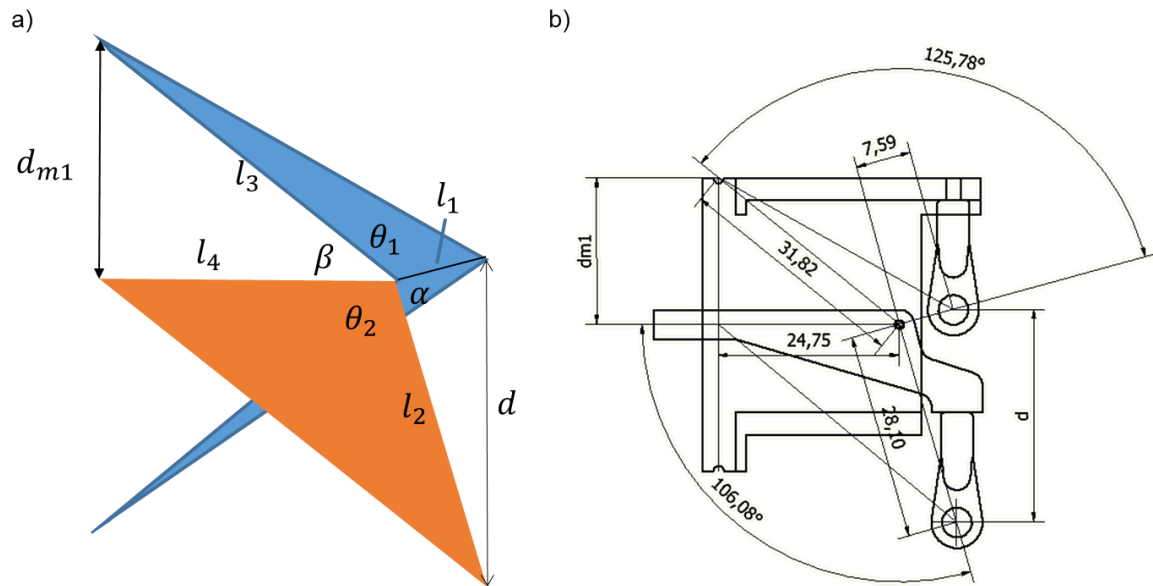


Figure 54 – SMA pitch link concept: a) cinematic kinematic diagram; b) dimensioning of prototype necessary to replace with original stiff pitch link.

The geometry amplifies the input displacement in 3.37X, named as boost factor. When in equilibrium the springs are assembled with deformation offset (pre-stroke) in 300%, setting the material in approximately the middle of a phase transformation region, even before dynamic operation, according to the strategies adopted by (HUANG *et al.*, 2014; MORAES *et al.*, 2018; REIS, 2018).

To avoid interference in the synchronism of the blade mechanism, the SMA pitch link should not perform great lengths of shifting. Thus, the maximum displacement of SMA pitch link should be limited. On the other hand, the device amplifies the input displacement by a factor of 3.37 (Figure 54), so that follows both criteria of low input displacement and spring large output deformation.

The commercial SMA springs were thermomechanical characterized aiming the temperature phase transformation in order to verify the Superelastic behavior in the environmental temperature. Sample SMA Spring was analyzed by Differential Scanning Calorimeter (DSC).

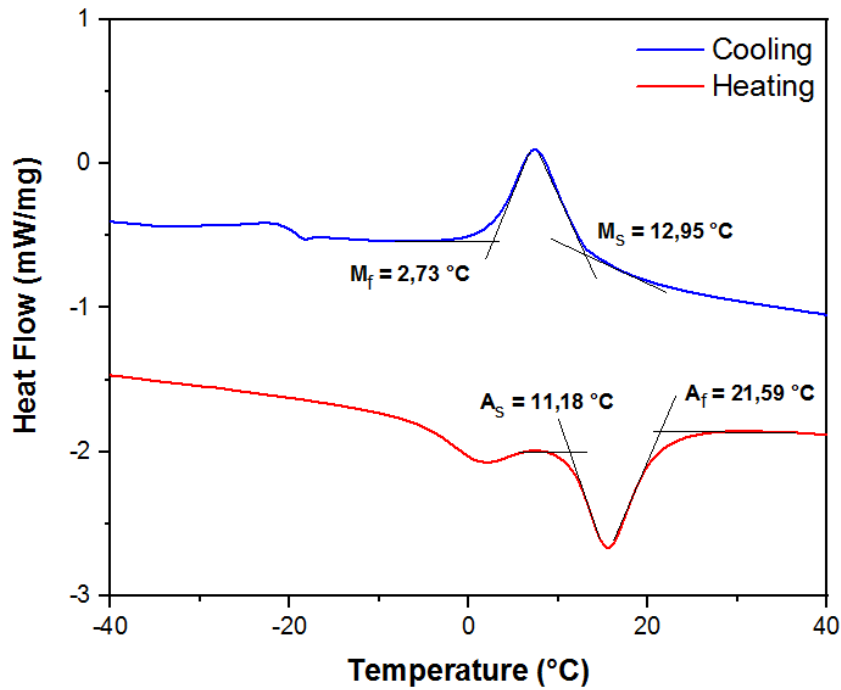


Figure 55 – SMA micro coil spring thermogram

During cooling the material exhibit phase transformation from martensite to austenite ($M_s \rightarrow M_f$). In environmental temperature (21°C) the material is completely austenitic ($A_s \rightarrow A_f$), which allows superelastic behavior. In laboratory is feasible to ensure temperatures above A_s , and superelastic response. Yet, in real external application, minimum distance between environmental temperature and A_f has to pain attention. The DSC results indicates no additional thermal treatment need before prototype installation. Isothermal conditions are assumed once considering Superelastic effect.

5 Results and discussions

5.1 Prototype assembly

According to the general requirements, the main parts of whirl tower are identified in Figure 56. The rotating parts compounds blades, rotating frame (mechanism of rotor hub). The fixed frame compounds electrical motor, tower, protection net and inertial table, depicted in Figure 57.

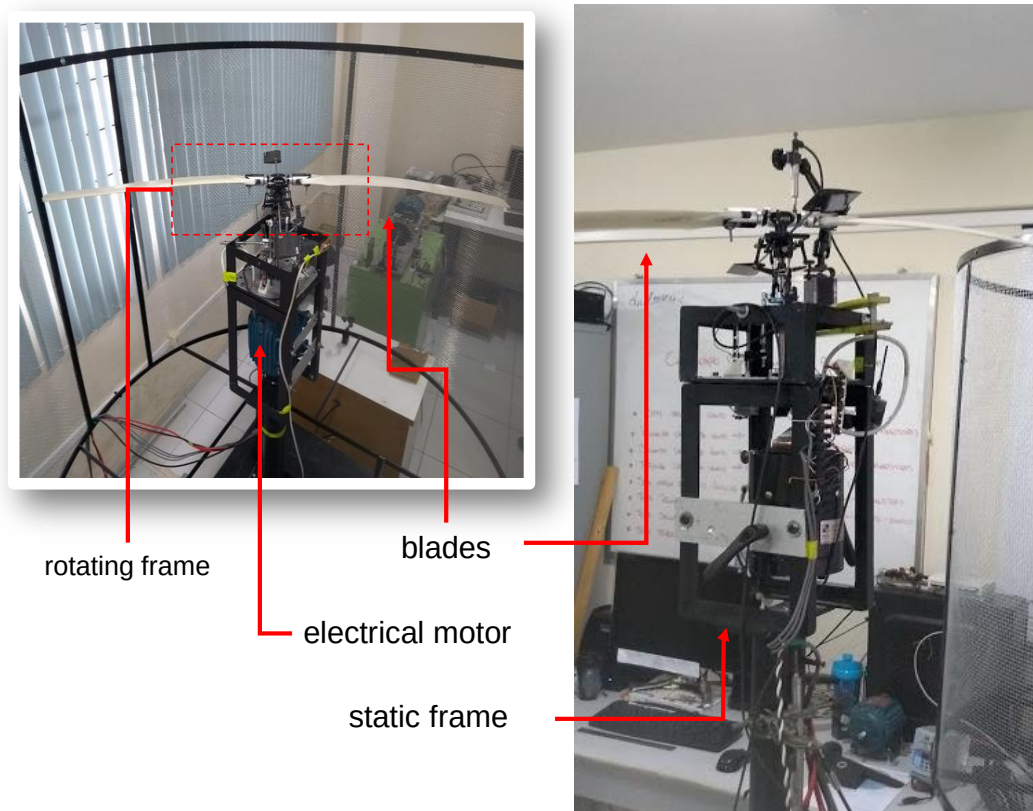


Figure 56 - Prototype main parts

Figure 57 exhibits the experimental apparatus installed in the Laboratory of Vibration and Instrumentation of Academic Unit of Mechanical Engineer at Federal University of Campina Grande (LVI-UAM/UFCG), ready for the measurements with sensors and net protection around its frame. The prototype, including a circular protection net, is 1.4m in diameter, 1.88m in height. The acquisition and control system is installed close the prototype and cabled for sensors and controls.

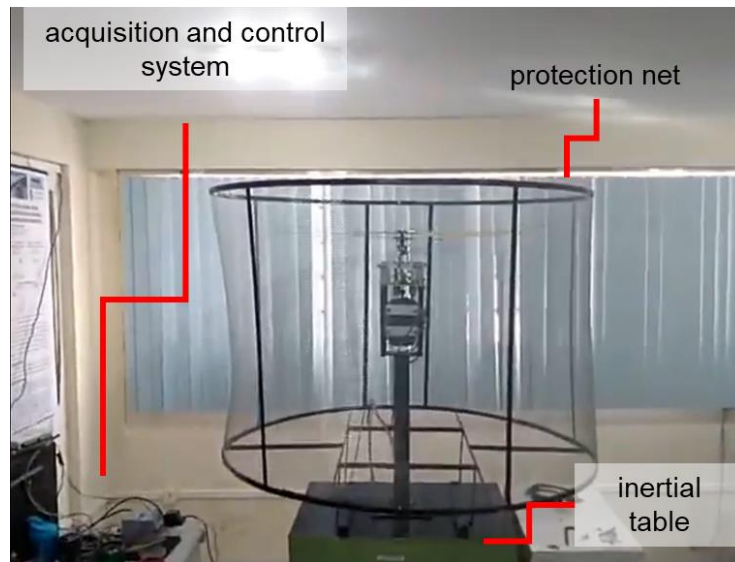


Figure 57 - Experimental apparatus showing the Whirl Tower

5.2 Functional tests

The functional test performs a sweep in the main rotor frequency. The main rotor was excited from minimum frequency possible to generate by the motor (3Hz) until maximum safety frequency. In real time, is measured the maximum vibration amplitude for each frequency tested, obtained by Labview interface (Figure 39). Sweep was performed until the tests detected a high increase in acceleration amplitude, when exciting frequency reach 12Hz. The high level of vibration suggests operational limit. Thus to preserve the safety of tests and integrity of structure was defined 12Hz as de maximum safety frequency to perform tests. In addition, the test helped to identify loose parts, gaps, clearances, and abnormal noises. On functional tests it was possible to identify eventual electrical problems, as electrical interference between the sensor signals and alternate current supply of peripheral systems.

The functional tests motivated improvements on the coupling between AC motor shaft and main shaft due to clearances. Clearances were eliminated with bolts and nuts changing. In general, based in visual inspection, no several damages or failures were detected on the functional tests. Summarizing, tests feasible the following:

- No loosed parts;
- Clearances in linkages on the first tests;
- No torque decreasing;
- Acceleration signal measurements has been successfully;
- Electric interference in analog inputs, solved with capacitors low-pass filters;
- Defined 12Hz as the maximum safety main rotor frequency.

5.3 Modal analysis of the blade

Numeric modal analysis was performed with CAD model in Ansys®. The numerical analysis parameters are itemized in Table 9.

Table 9 - General parameters of blade numerical modal analysis

Material applied for Ansys bench	Polyethylene
Young's Modulus (edited) determined experimentally (Figure 43)	1.8GPa
Mesh	1.376.810 nodes 827.755 elements

In addition, was employed the experimental scheme depicted in Figure 44 to obtain the first and second natural flapping vibration frequency in order to compare with numerical results. Figure 58 shows the modal frequencies obtained numerically and experimentally.

According to the numerical modal analysis, the natural frequencies in feathering and lagging is above operation frequency of the prototype and did not motivate experimental analysis in those frequencies. The span frequency of the experiment is limited to 60Hz once the theoretical whirl tower maximum frequency is 60Hz, in practice, the operational the frequency did not exceed 12Hz.

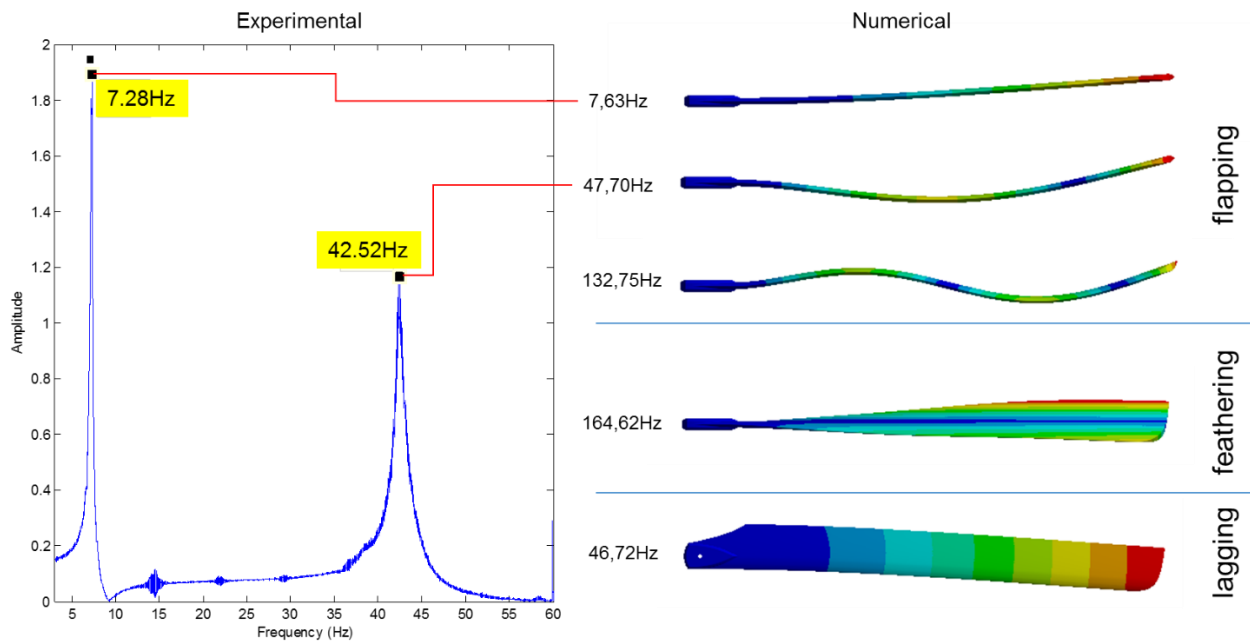


Figure 58 - Comparison between FFT from experimental results and numeric modal analysis by Ansys, with first and second natural frequencies highlighted.

5.4 Aerodynamic moment on the blade

The aerodynamic effect of the blade consists mainly, in drag and lift forces. Yet, the effect these forces may induce internal forces on the blade mechanism. Employing the mathematical approach of the blade element theory in vertical flight, the Matlab code of Appendix E demonstrates how to calculate the aerodynamic force and internal compressive load F_R . The algorithm returns internal compressive load $F_R = 2,01N$ due to aerodynamic loads of the blade, considering pitch angle in 10^0 and $\Omega = 7Hz$.

5.5 Blade linkage forces and displacements

The linkage forces and displacements are experimentally obtained from scheme of Figure 47. The results represent static measurements of mechanism, so that, input force is from internal mechanical resistance, disregarding, the aerodynamic load. In dynamic condition, the aerodynamic force F_R (exhibited in section 5.4) and the input force F_{SW} compounds the compressive resultant forces in the pitch link. Table 10 shows the input force and displacements to five frequencies. These results are important to estimate the internal force induced by the mechanism, assigned to friction, which vary due to inertial forces according to frequency shifting. These measurements were performed with a maximum pitch angle of 10 degrees.

Table 10 – Experimental results of maximum input forces and displacements in static test with blade mechanism.

Frequency (Hz)	Maximum input Force F_{SW} (N) (Load cell)	Maximum displacement Input (mm) (WA20)	Maximum displacement Output (h) mm (WI10)
5	0.7101	1.2325	0.4059
6	0.5639	1.2088	0.4215
7	0.5549	1.1620	0.4240
8	0.4483	1.0877	0.4278
9	0.3808	1.0018	0.3987
Standard deviation	0.1256	0.0943	0.0126

Following, carried out an analysis of the resultant force in pitch link (rod l_3 in Figure 59) for 10° of pitch angle. Aimed with a CAD model of the mechanism in Autodesk Inventor® it obtained angles: $\theta_1 = 4,94^\circ$, and $\phi = 9,18^\circ$, for $\alpha = 10^\circ$.

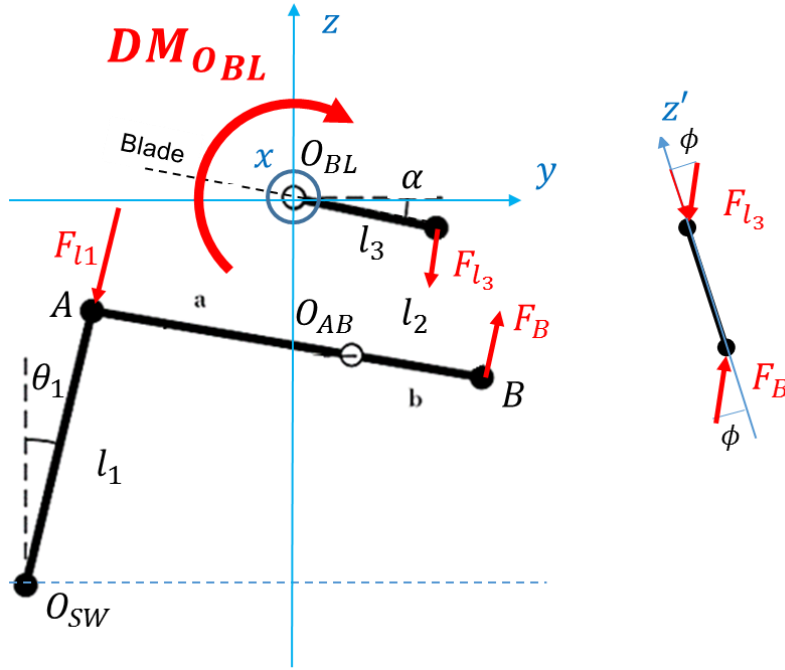


Figure 59 – Blade mechanism and free body diagram for $\alpha = 10^\circ$

Considering the deviation in Load with frequency, it assumed the input force experimentally measured in 7Hz (Table 10) due to the proximity with the first resonance of the blade (Figure 58). Thus, based on free body diagram of l_3 , using Equation D5, as detailed in Appendix D, it obtain the resultant maximum force under l_3 , $F_R = 2N$. It's important to explain that F_{SW} vary due to rotational position of main shaft when the swashplate is inclined (Figure 39), thus F_{SW} used to calculate F_R consider F_{l1} direction according to depicted in Figure 59, that generate the maximum compressive load in l_3 . It's important to highlight that these forces are steady-state loads and do not consider eventually loads due to lateral gusts, for example.

5.6 Numerical and experimental modal analysis of whirl tower

Were performed modal static and dynamic experiments aiming to identify the critical frequencies that may affect the machine's response and which limits the operational test. Modal tests with hammer impact find the structural natural frequency to investigate the contribution of these frequencies to the dynamic spectrum response. Dynamic tests took place according to the limits established after functional tests. This test aims at the spectrum of rotating parts (mainly the rotor) mixed with the structure natural frequencies.

5.6.1 Numerical and experimental modal analysis (free vibration)

The modal analysis performed by Hyper Mesh® consider an applied input force of 1N on each point of Figure 41, attempting to reproduce the experimental procedure performed with the prototype. The frequencies from simulation are exhibited in Table 11.

Table 11 – Mode shapes of whirl tower modal shape by numeric simulation

Results of frequency (mode shape)	1	20.40 Hz
	2	72.00 Hz
	3	107.80 Hz
	4	115.4 Hz

The experimental modal analysis generates four FFT's according to the points of Figure 41. The results are depicted in Figure 60.

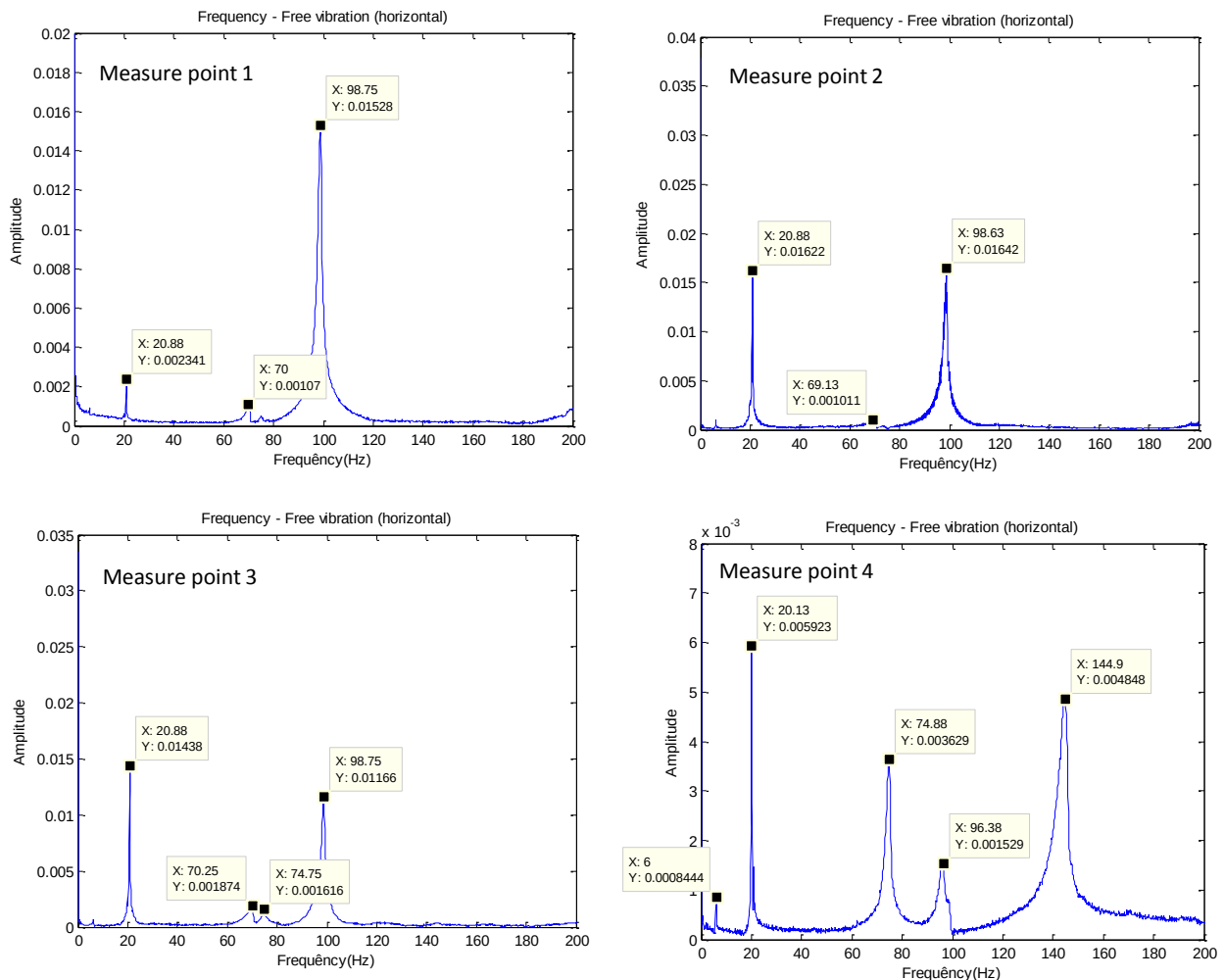


Figure 60 – Dynamic response of experimental modal analysis in four measure points

The spectral response of the experimental results in frequency of 20.88Hz, 70.0Hz and 98.0Hz (approximately) is observed in all of measured points. Comparing with numerical simulation, the first frequency close to 20.0Hz converges. The second mode, close to 70Hz even though detected in the simulation did not exhibited amplitude to allow comparison with experimental result. The numeric method performs results in four directions, yet the simulation of point 2 demonstrates results more precisely than points 1, 3 and 4. Thus, the simulation of point 2 was considered as comparative with experimental result. Considering the operation frequencies of prototype below to 60Hz, the comparative adopts a span of 0 to 60Hz, depicted in Figure 61.

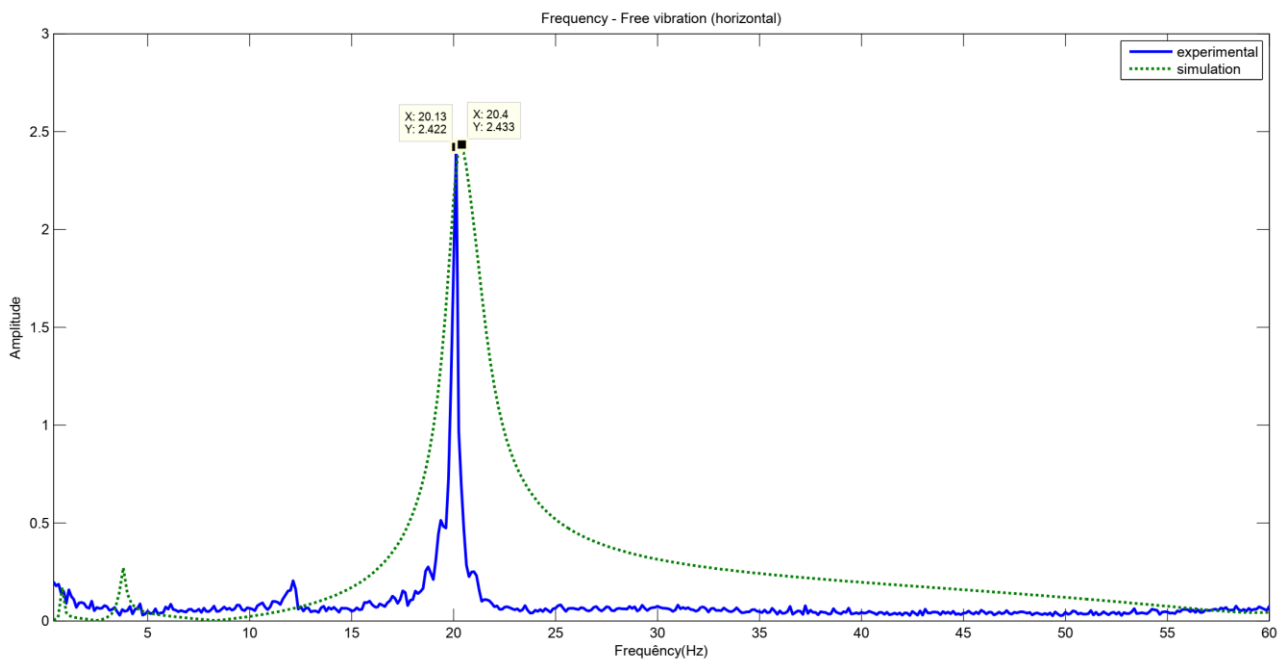


Figure 61 - Frequency in free vibration of whirl tower without rotor head – simulated and experimental

As observed in Table 11 and Figure 60, after the second mode, the modes from experimental diverges because the geometric complexity of the prototype induces differences between experimental and numeric simulation. In addition, the simplifications of numeric modal contribute to these discrepancies. The numeric model considers an input force of 1N. For experimental with hammer, is not possible to ensure the constant force. Therefore, the amplitude of experimental and numeric are adjusted regards to compare the frequency converge, despite the amplitude.

5.6.2 Forced vibration with a frequency sweep in rotor head and blades

A forced vibration experiment, with the prototype completely assembled, employing a sweep frequency was conducted according to the limit established in functional test. Thus, sweep run rotor until 12Hz, with frequency rate at 4Hz/min. Figure 62 depicts the Power Spectrum Density of this sweep highlighting the interval (0-20Hz). The most relevant power peaks are observed in frequencies 5.80, 12.48, 13.65 and 14.81Hz. Considering that the modal frequencies of the structure take place above 30Hz, as detailed in the previous section, the frequencies obtained from the sweep test are related dynamic behavior of the rotating parts. In other words, the sweep demonstrate that the highlighted frequencies of Figure 62 corresponds to the natural frequencies of the mechanism compounded by the blades, linkages (among these, the pitch link), and the rotor hub. In a real steady operation, these frequencies should to be avoided.

Taking into account $N\Omega$ ($N=2$ blades), the first natural frequency of the blade (7.28Hz - experimental) induces a correlation with frequencies between 12 to 15Hz ($2 \cdot 7.28 = 14.56\text{Hz}$) highlighted in detail of Figure 62.

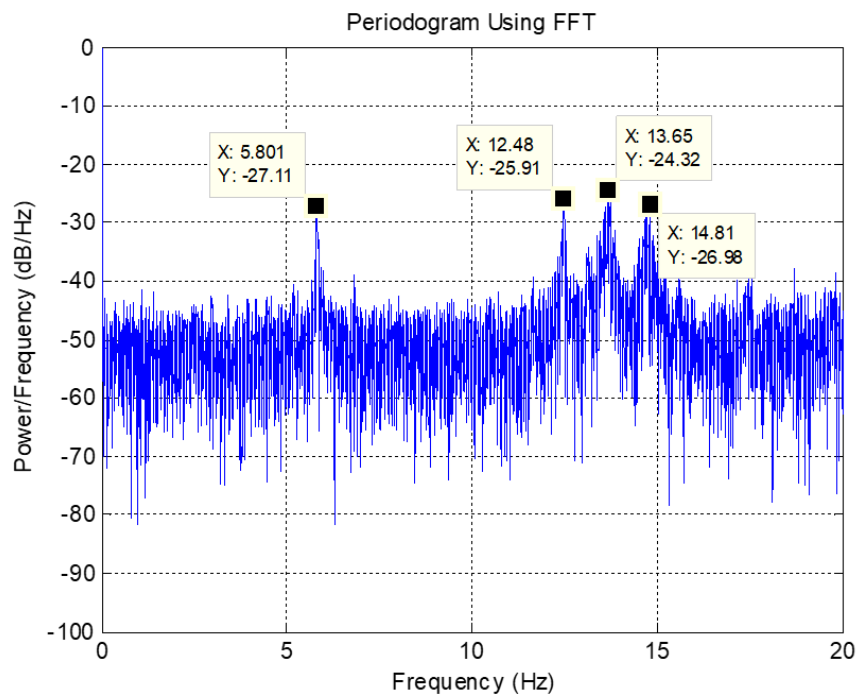


Figure 62 – Sweep frequency experimental Power Spectrum Density

5.6.3 Dynamic response at stationary state

This experiment performs tests with the rotor in constant rotational speed (stationary state). Functional tests specified 12Hz as the maximum safety operation frequency due to the prototype limitations. The sweep tests, which spectrum is depicted in Figure 62, aid to observe that the operational test will take place with frequencies below the first shape mode of tower. In addition, the interest is to study the dynamic effects in operation frequency near to the resonance of rotor head components, as example, the blades. Because of this, for dynamic analysis was adopted 10Hz of main shaft frequency, as a safety region between 5.8 and 12.48Hz. This frequency chosen ensure a long period of measurements without influence of transient structure behavior. The power Spectrum Density from dynamic experimental test at $\Omega = 10Hz$ is depicted in Figure 63. The highlighted area in red represents the most relevant amplitudes and frequencies, directed linked with fundamental rotational frequency Ω , and $N\Omega$. The Spectral response of highlighted area of Figure 63 is showed in Figure 64.

Although the frequency inverter is configured to 10Hz, the spectrum denotes a variation of 7.7% in Ω , imputed to motor slip, and the aerodynamic resistance. Supported by the simulation and the experimental results with the tower structure, its inferred that the frequencies in the region of (20-35)Hz correspond to the natural structural frequencies (in dynamic tests is modified due to rotor head mass) and harmonic 2Ω .

The accelerometer is installed on non-rotating frame (Figure 40). According to Table 1, all of vibrating contributions originated in blades are transmitted to nonrotating frame at $N\Omega$. Considering $N=2$ for this prototype, any strategy to reduce vibration in rotor head will verified at $N\Omega$.

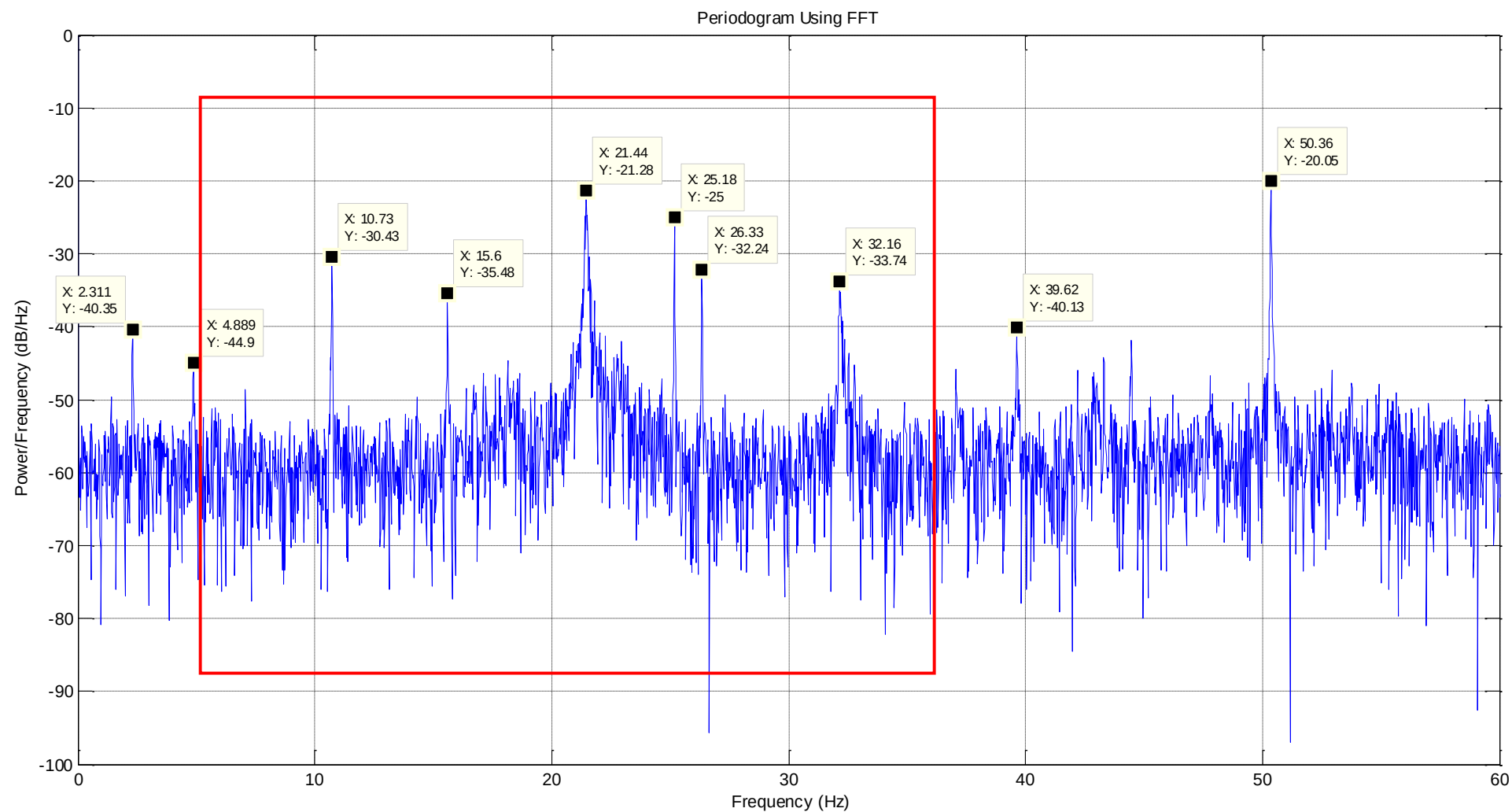


Figure 63 - PSD with rotor operating at 10Hz

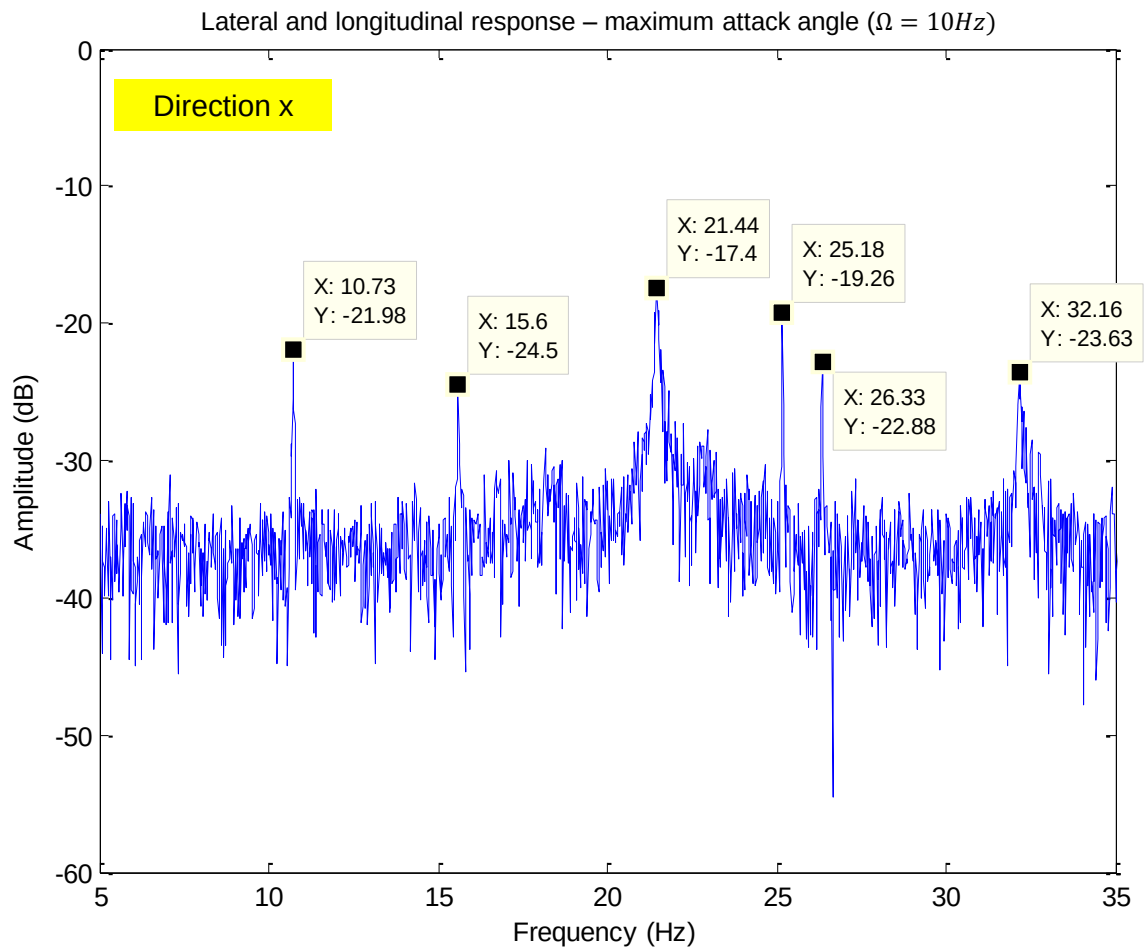


Figure 64 – Zoom of spectral response with rotor at 10Hz

5.7 SMA pitch link

According to the concept was fabricated the prototype by additive manufacturing in PLA (Polylactic Acid), as shown in Figure 65. Using this SMA pitch link carried out experiments in order to analyze the coil spring assembling, dynamic tests with modal shaker (forced vibration) and hysteretic behavior of device. The position of the static eyelet determines the offset of the springs. In Figure 65 a detail of closed coil spring before offset demonstrates the length of each element. To obtain the length of 5mm the original M9 closed spring (Figure 65a) was sectioned so that employing 6 coils, as depicted.

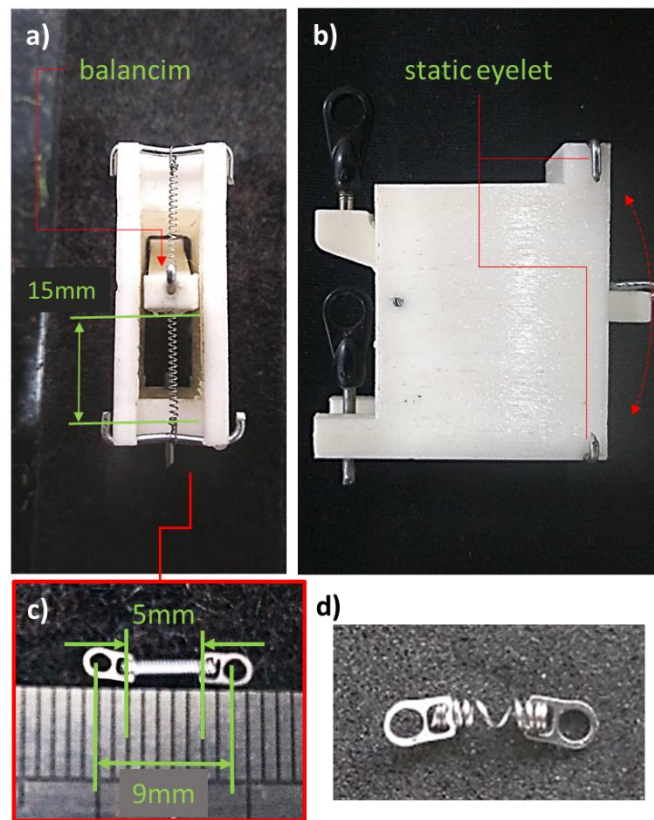


Figure 65 – SMA pitch link: a) PLA prototype (front view); b) PLA prototype (side view); c) dimensions of closed coil spring before assembly; c) distalizing spring (detail of one opened coil)

The offset 15mm (Figure 65a) is adopted both to closed spring and to distalizing spring (Figure 65d). The theory behind the distalizing spring employing because this model of spring would demand less strain to induce phase transformation when compared with the closed spring for the same conditions. Because of this, the tests were performed with closed and distalizing spring with any geometrical changing except by the spring replacement.

Figure 66 illustrates the device and exhibit its summarizing general characteristics. The total mass exhibited consider the closed coil spring. The mass difference with when uses distalizing spring is about 7mg for each element.

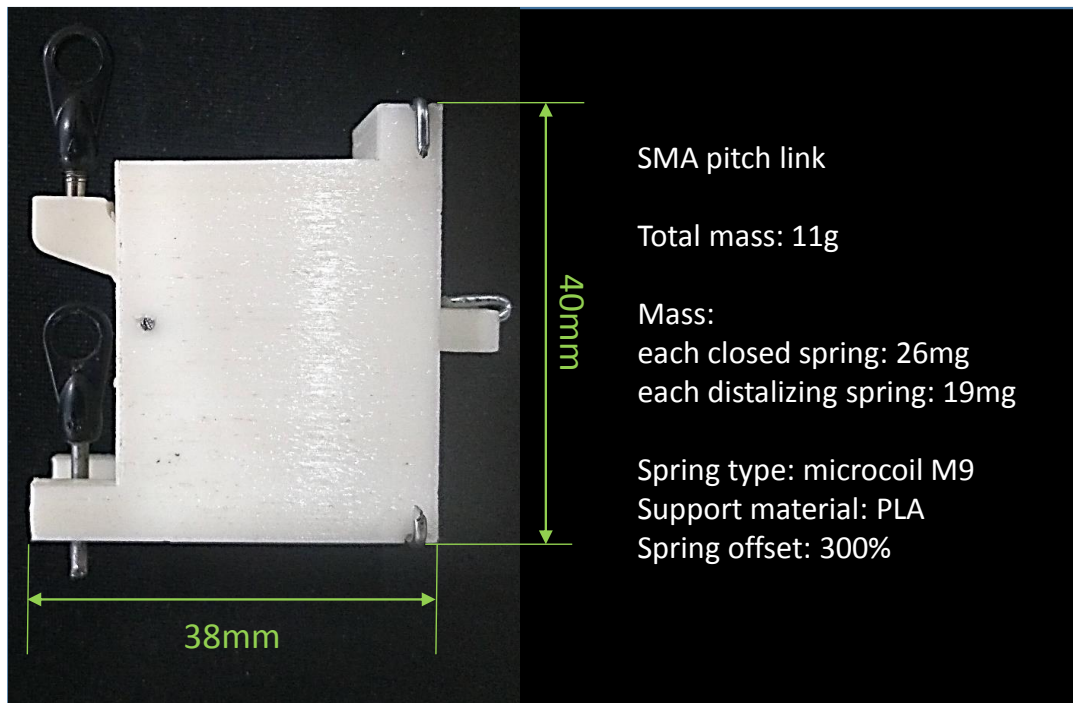


Figure 66 – SMA pitch link: general characteristics

Figure 67 illustrates the replacing of original rigid pitch link to the SMA pitch link. The additional area required by the new device do not cause any mechanical interference in the original disposing of the rotor mechanism.



Figure 67 – SMA pitch link replacing the original stiff pitch link

The experimental data with SMA prototype, reported in literature, demonstrates maximum Superelastic deformation about 500% (GRASSI *et al.*, 2015; MORAES *et al.*, 2018). In this particular application would be necessary maximum length of 25mm to induce a plastic deformation, an undesirable condition considering the geometric limitations. The available length of 15mm in the balancim will not enough to induce plastic deformation of coil springs so that avoid permanent damages in the springs due to extremely deformations. Yet, the minimum input forces under device ensures hysteretic behavior of the SMA.

5.8 Dynamic tests with SMA pitch link

Before device installation in the prototype, corresponding to the first functional test, a bench test performs input force and measure displacement to simulate the real operation as shown in Figure 68. The kit employed (modal shaker, load cell and LVDT) was the same of scheme from Figure 44. The input peak force not exceeded the maximum load calculated in 4.2.6, $F_R = 2N$.

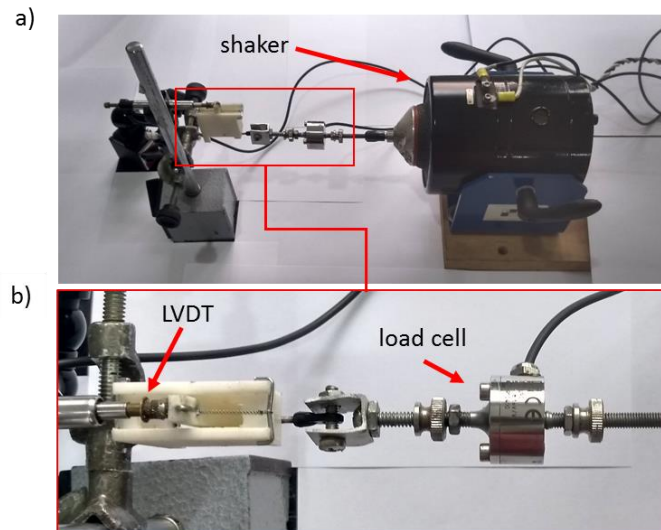


Figure 68 – Scheme for receptance measurements with SMA pitch link: a) assembly with modal shaker kit connected to device; b) detail indicating load cell and LVDT positioning.

The experiment generates the curve depicted in Figure 69 when pitch link employs closed micro coil spring and in Figure 70 when the device employs distalizing micro coil spring, both under a sinusoidal input signal, generating maximum peak force in approximately 2,5N. The graph exhibits the variation of distance between pitch link eyelets (distance d , depicted in Figure 54b) for the correspondent input force. Because of LVDT positioning, in the opposed tip of balancim, a correction of the LVTD measurements, based on the boost factor (Figure 54) calculates the effective displacement in d . Thus, the distance d and force F (load cell) generates the force/displacement curves.

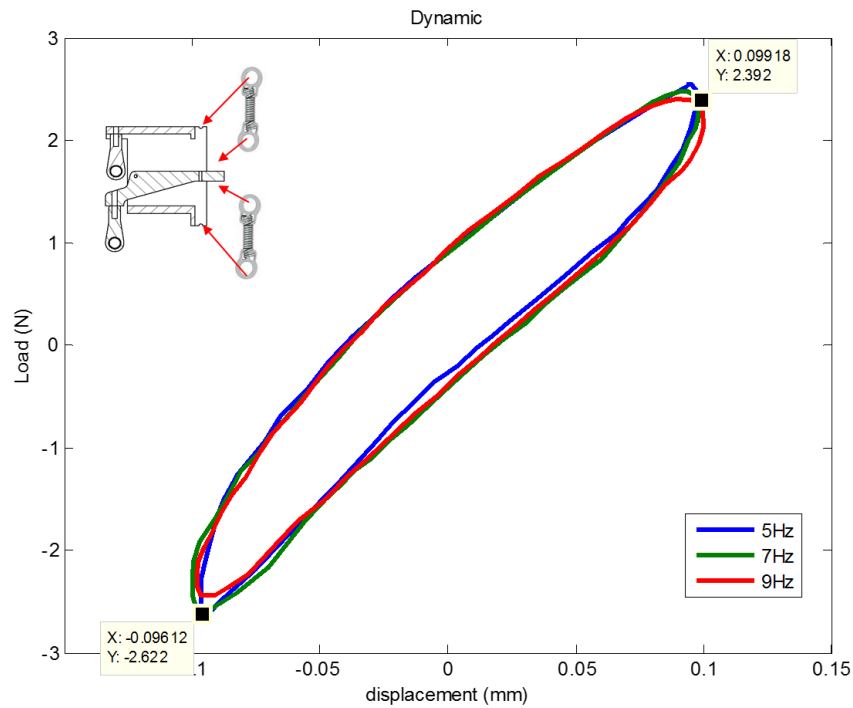


Figure 69 – Force/displacement curve of SMA pitch link employing closed coil spring with 300% offset.

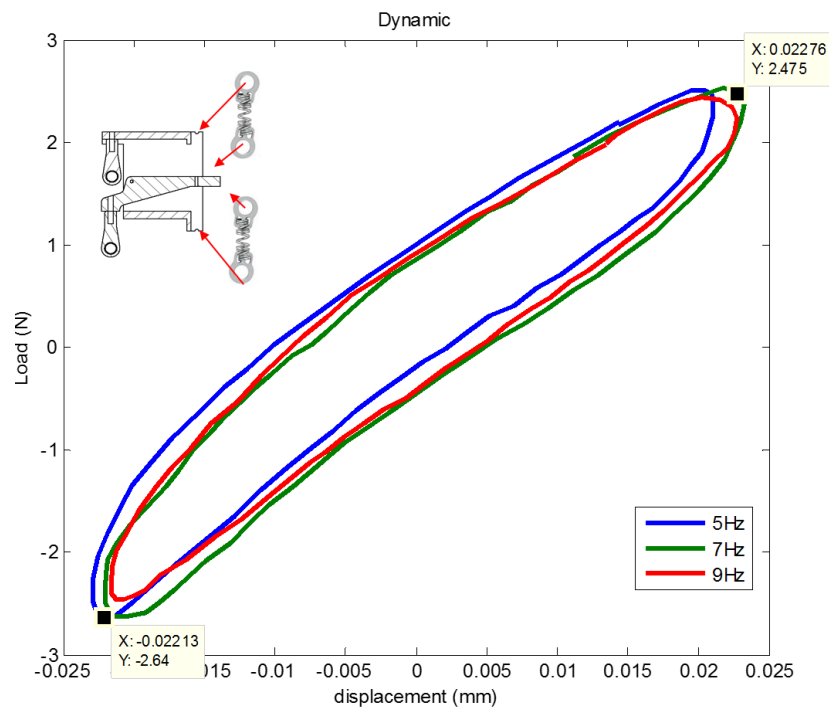


Figure 70 - Force/displacement curve of SMA pitch link employing distalizing coil spring with 300% offset.

The input frequencies in 5, 7, and 9Hz (interesting frequency span) were tested to verify the presence of hysteretic behavior. For all of these frequencies, the experiment indicates hysteresis, although the maximum device displacement, and consequently the springs, exceed about 10% of the

offset (peak-to-peak). This result demonstrates the capacity of material's hysteretic behavior, even in lower displacement when adopted the strategy of pre-stroke. The distance between eyelets did not exceed 0.20mm (peak-to-peak) for closed springs and did not exceed 0.05mm (peak-to-peak) for distalizing springs. Table 12 exhibits the dissipated energy and secant stiffness of each pitch link: employing closed and distalizing SMA springs. The dissipated energy represents an increase of about 2.33% with distalizing assembly compared with closed SMA springs. On the other hand, the stiffness rises 344% with distalizing SMA springs.

Table 12 – Dissipated energy and secant stiffness for each type of coil spring adopted in the SMA pitch link

	Spring type	Pitch link active coil spring volume (m^3)	Dissipated energy (MJ/m^3)	Pitch link stiffness (KN/m)
	Closed	$2.726 \cdot 10^{-9}$	0.257	25.67
	Distalizing	$0.964 \cdot 10^{-9}$	0.263	113.94

5.9 Dynamic tests with SMA pitch link in the mechanism with non-rotating main shaft

In this test the SMA pitch replaces the original pitch link. Due to the external instrumentation, as LVDT exhibited in Figure 71, is not possible to perform the measurements with the main shaft in rotation. As comparative, the tests were performed with stiff pitch link, and repeated with the SMA pitch link. Figure 71a shows the simulation of SMA pitch link assembly. In Figure 71b, the real assembly demonstrates similarities, except the eyelets repositioning.

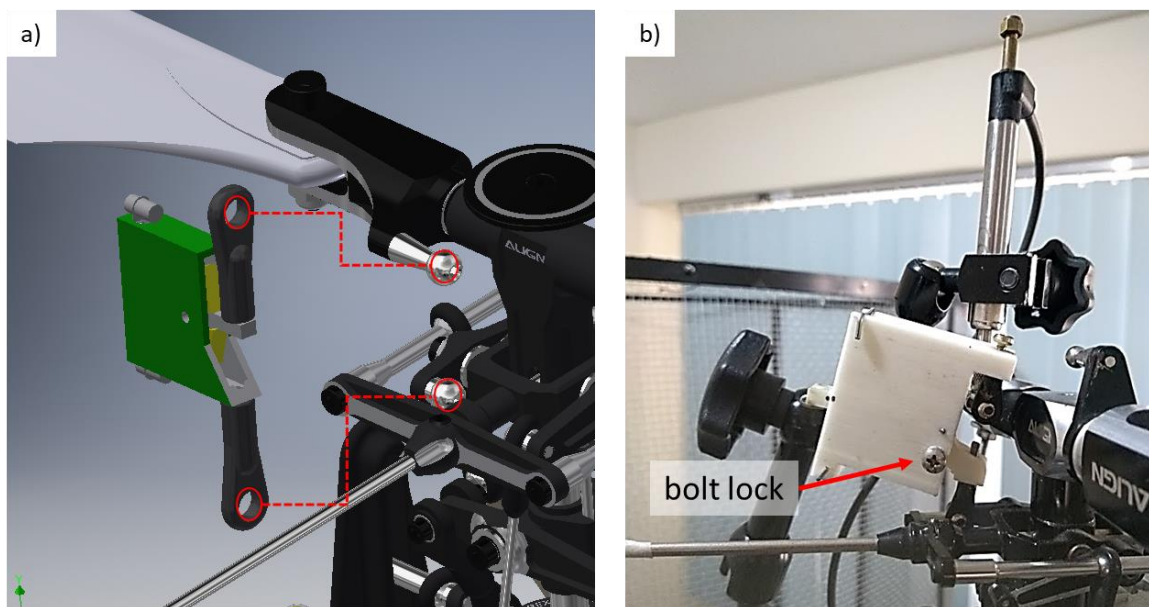


Figure 71 – SMA pitch link prototype assembling: a) simulation of assembly to verify available closure; b) real assembly in the prototype

To avoid constant components replacing and an accidental assembly interference among tests it was adopted the following strategy: the device named as “stiff pitch link” is the SMA pitch link with it balancim locked by a bolt as indicated in Figure 71b. Thus, there is no spring movement so that the device performs as a rigid link. In sequence, to test the SMA pitch link the lock was removed.

As mentioned, these tests were performed with the rotor stopped, with rotational parts exited by a shaker to simulate the cyclic load in the blade mechanism according to the scheme depicted in Figure 47. The first dynamic experiment with the assembly of Figure 71. Following the span (0 to 20Hz) adopted in the experimental modal analysis (as exhibited in Figure 62), was performed a sweep frequency during 180s with sample rate in 600Hz. The reference condition was based on stiff pitch link. Both spectral responses doesn't correspond to the helicopter prototype because disregard the dynamic contribution of the whirl tower, main shaft motor, and bearings. Thus, the response corresponds to superimposed effects of the blades and the blade mechanism.

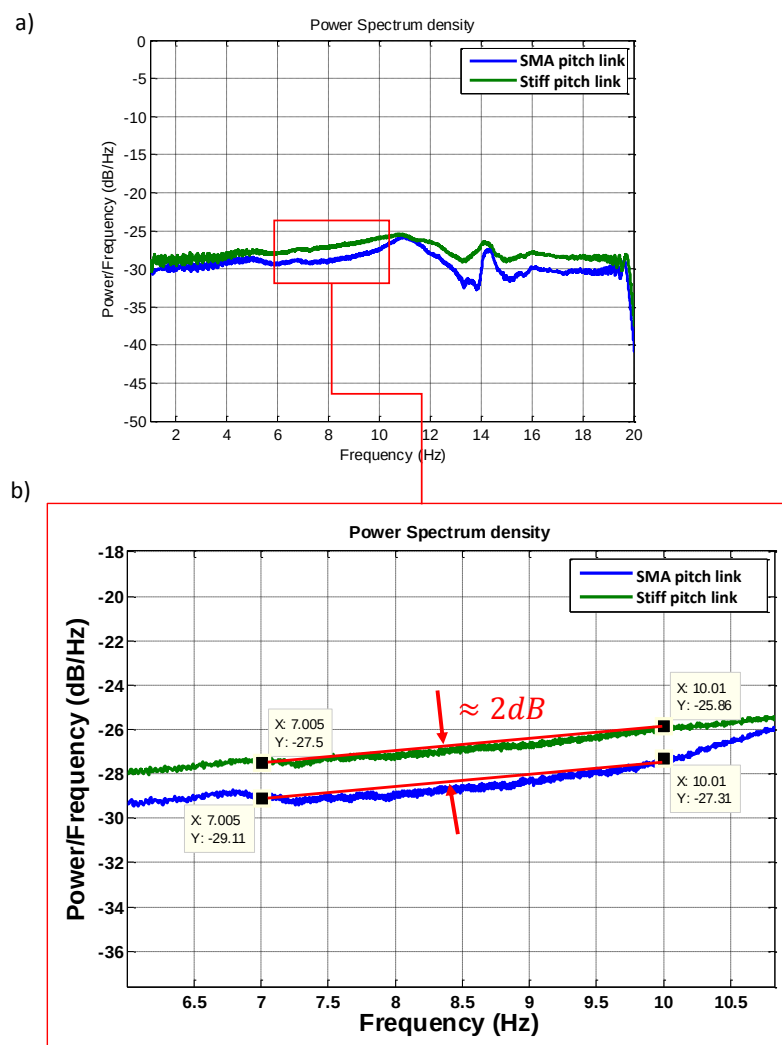


Figure 72 – Power Spectrum Density of sweep in functional comparative test using stiff pitch link and SMA pitch link:
a) large span 0-20Hz; b) detail in span of 6-11Hz highlighting attenuation of approximately 2dB in span of 7-10Hz.

The sweep depicted in Figure 72a reveals other region with relevant relative attenuation, between 13-15Hz, according to detail in Figure 73. The peaks in 13-15Hz is present in response of Figure 62, and corresponds to vibrating modes of rotating parts.

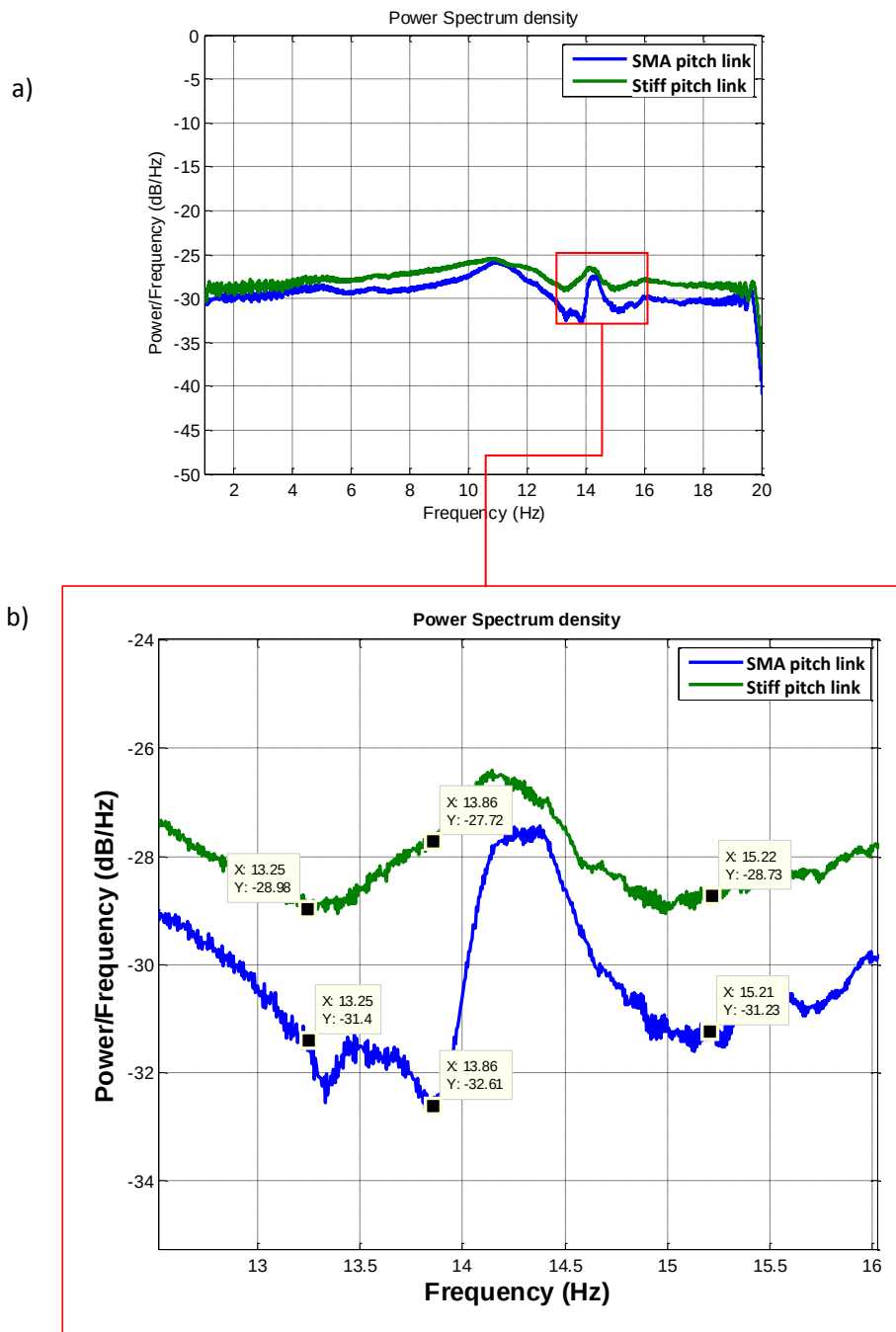


Figure 73 - Power Spectrum Density of sweep in functional comparative test using stiff pitch link and SMA pitch link:
a) large span 0-20Hz; b) detail in span of 12-16Hz highlighting attenuation in span of 13-15Hz.

Figure 74a and b represent respectively, the periodogram in time domain and a spectrogram of the response with stiff pitch link, respectively. In Figure 74c and d are represented the periodogram

and spectrogram response with SMA pitch link. Considering the time domain representation (Figure 74e), in measured time interval of 180s, the RMS value is 0.18 for the stiff pitch link (blue lines) and 0.15 for SMA pitch link (green line). Observing the spectrogram, the color bar indication aims to realize that greatest power density of signal take place in about 90s of sweep. This power density matches when frequency crosses the region close to 10 Hz (approximately between 8 and 11 Hz) as observed in Figure 74a. Similar results is observed in Figure 74c and Figure 74d.

Considering the results with and without SMA pitch link, for comparative using the spectrogram, the orange area of SMA is less intense when compared with stiff pitch link, which confirm the attenuation due to SMA device. Figure 75 details the spectrogram in span of 7-10Hz where the data tips indicates the relative attenuation of 2dB approximately, confirming what is represented in PSD (Figure 72). It's important to explain that in spectrogram the horizontal axis do not represent the time interval, but the samples overlapped. Thus, the time interval of the span between 7 to 10Hz is the same represented in the Figure 74.

The PSD and the periodogram demonstrate no resonance shifting, which indicates that the most relevant contribution of SMA is regard damping, not affecting the equivalent system stiffness. In addition, no resonance shifting is verified in time domain response.

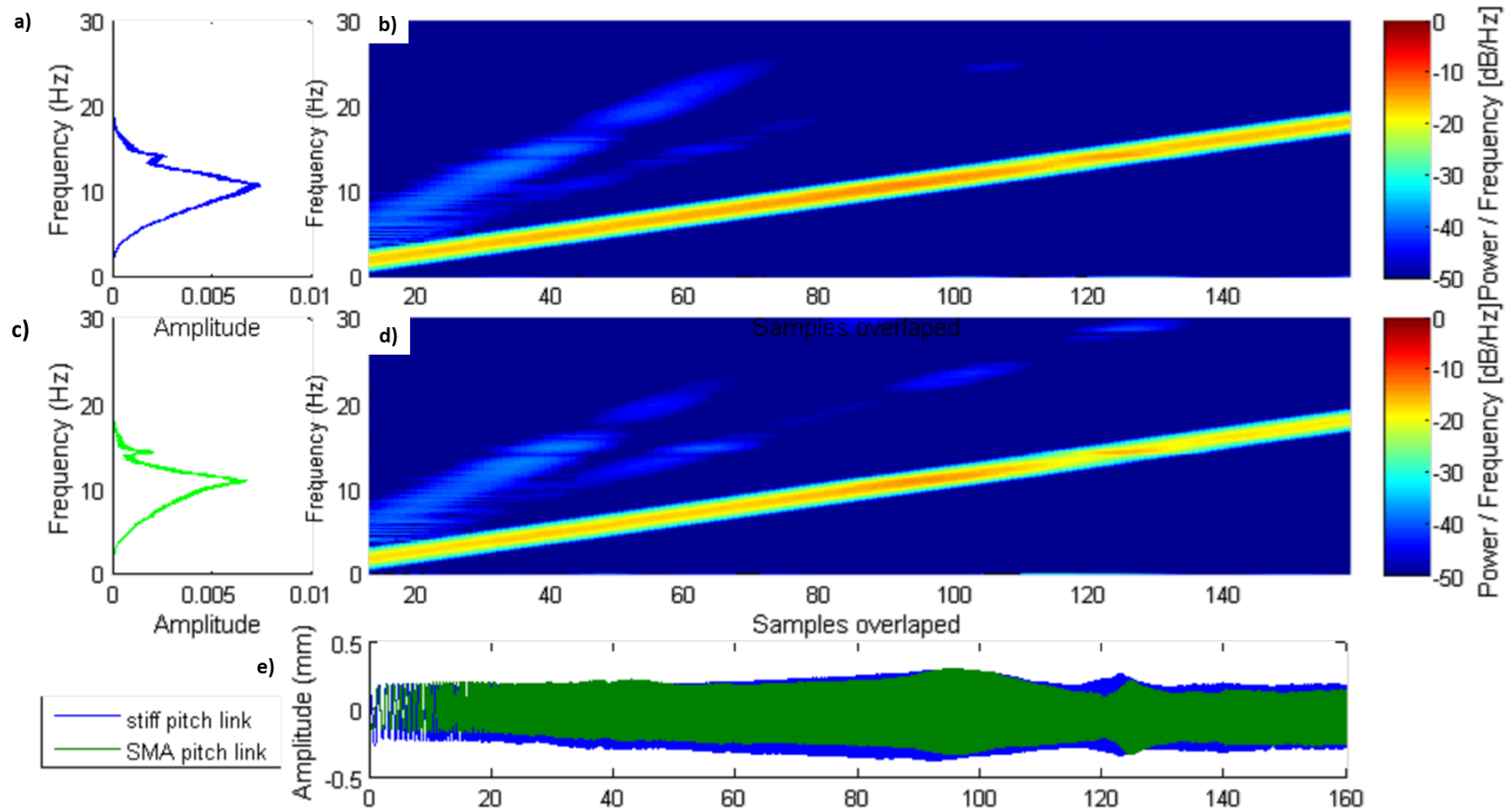


Figure 74 – Tridimensional response in sweep for 0-30 span during 160s: a) Power Spectrum Density with stiff pitch link; b)) Power Spectrum Density of stiff pitch link; c) periodogram with SMA pitch link; d) spectrogram with SMA pitch link e) time domain comparative between stiff pitch link (blue line) and SMA pitch link (green line).

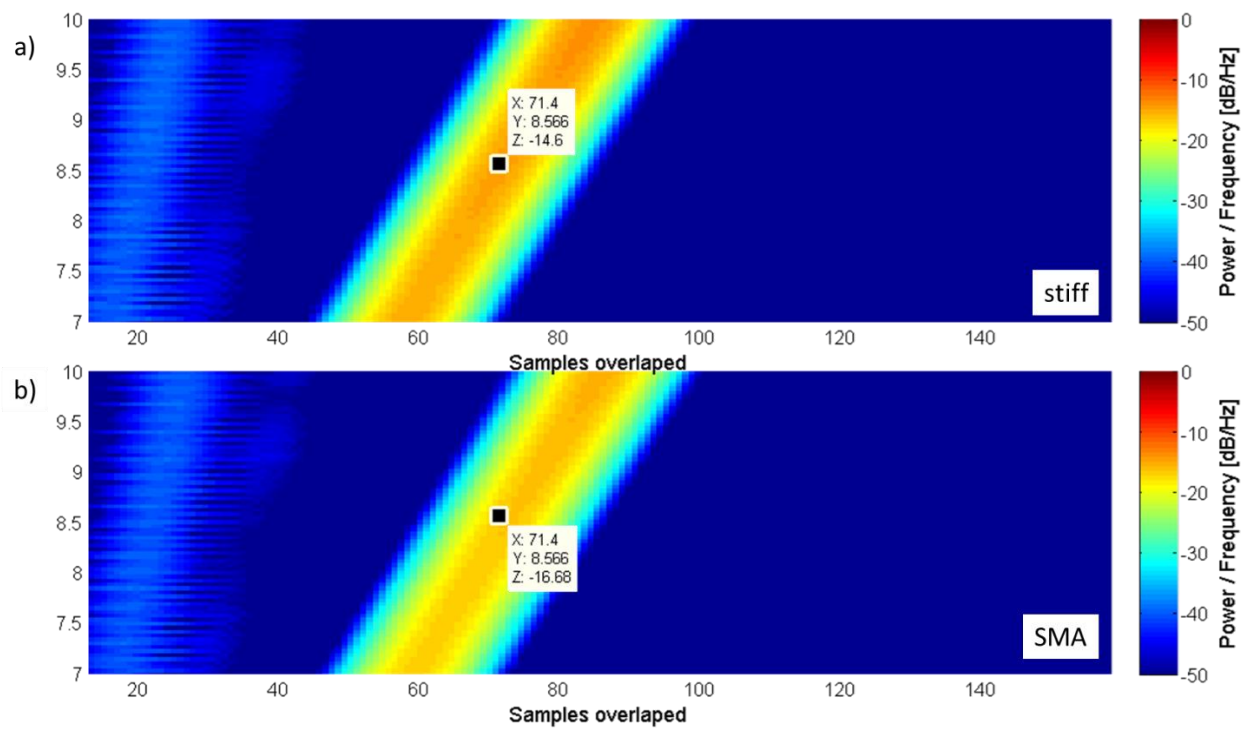


Figure 75 – Sweep detailing spectrogram of span 7-10Hz

The power densities measured in the interval of 7-10 are exhibited in Table 13 with maximum attenuation in bold.

Table 13 – Signal power density measured in the interval of 7-10Hz highlighted in Figure 75

Frequency [Hz]	Power density [dB/Hz]		
	Stiff pitch link	SMA pitch link	Attenuation (dB)
7	-15.3	-16.9	1.6
8	-14.8	-16.6	1.8
8.56	-14.6	-16.6	2.0
9	-14.3	-16.0	1.7
10	-13.6	-15.1	1.5

5.10 Dynamic tests with SMA pitch link with main shaft in rotating

To verify the effect of the SMA pitch link in the prototype in operation, with the main shaft in rotating to simulate an eminent hover flight, were performed an experiment using a pair of SMA pitch links assembled in the rotor, as illustrated in Figure 76. The pair of pitch links follows the general design of Figure 66, except for the modifications in the support. These modifications regards to material reduction in both sides aiming mass and drag reduction during rotation. The total mass of each pitch link is 9 grams.

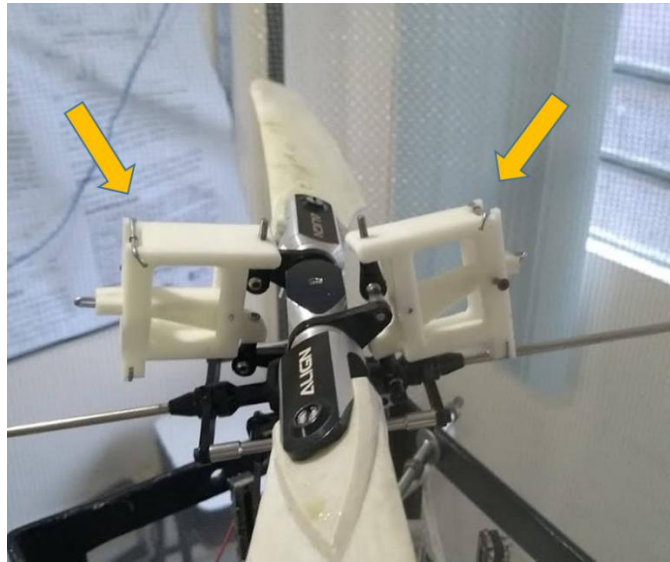


Figure 76 – SMA pitch link pair assembled in the rotor to the rotational tests.

The dynamic tests generates partial results compounded by a sweep in frequency, which accelerates the rotor from zero to 10Hz, adopting the blades in maximum pitch angle (to maximize the drag and load in the blades). The tests were performed with SMA pitch link configured as a stiff pitch link (with lock) and repeated with free pitch link (without lock). The Figure 77 depicts the power spectrum density (PSD) of sweep frequency in the rotor witch the accelerometer was installed in the rotor chassis (according to the positioning exhibited in Figure 40).

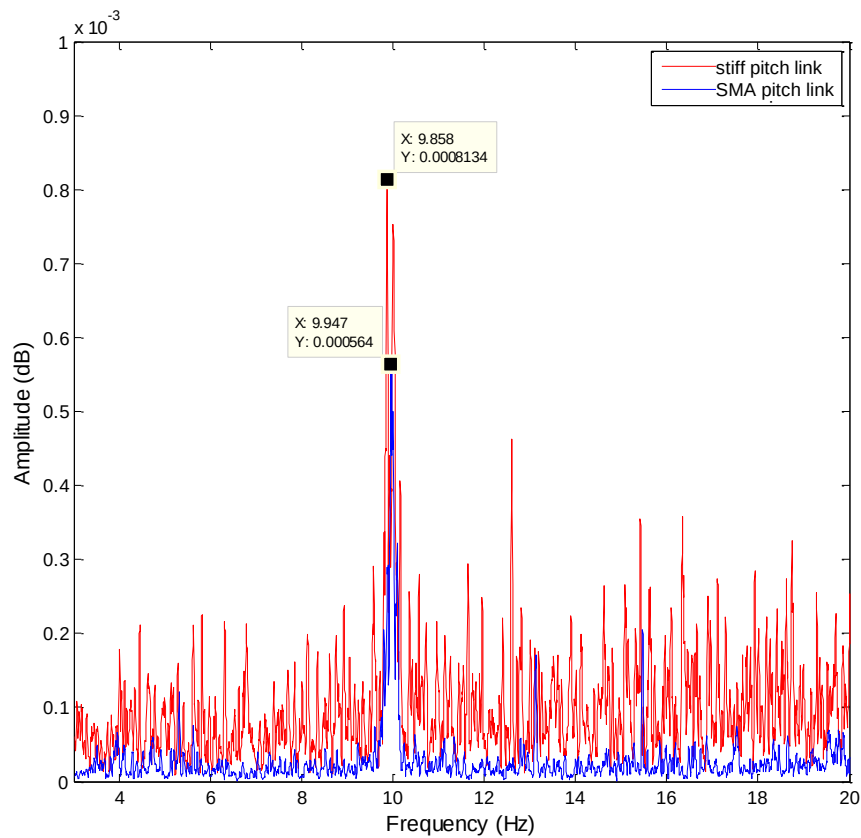


Figure 77 - Power Spectrum Density comparison of sweep in frequency for SMA pitch link in rotation tests

The PSD of Figure 77 exhibit a wide variety of harmonics when pitch link is configured as a stiff pitch link. These frequencies may originate in the flapping, lagging and feathering movements of the blades (according to Figure 11) transmitted through the cyclic and collective commands to the prototype structure, as established in Table 1. When the pitch link works with SMA (unlocked) the dynamic response changes so that highlights some frequencies possible to observe in Figure 78, without superposition, that exhibit the results only with SMA.

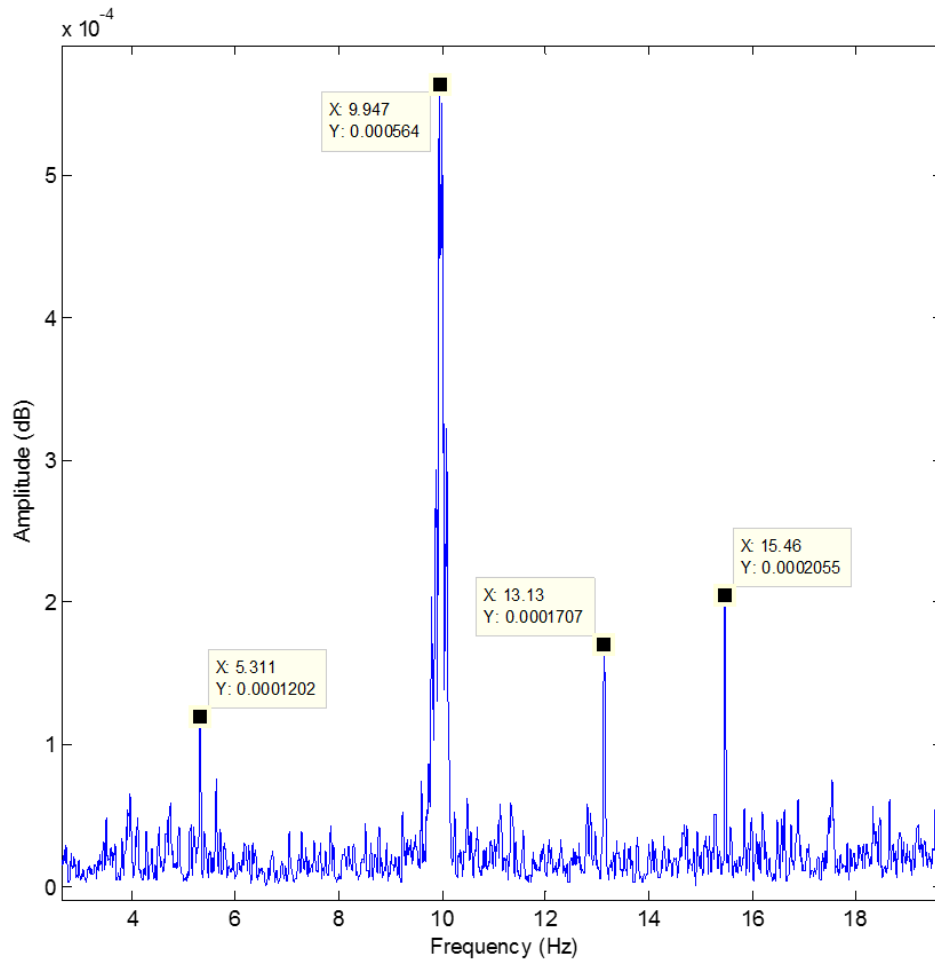


Figure 78 – Power spectrum density of sweep in frequency – response with SMA pitch link.

The response of this sweep is also observed by a periodogram and a spectrogram with stiff pitch link, respectively in Figure 79. Instead of response of Figure 74, in sweep of Figure 79 were obtained from acceleration measurements and corresponds to the overall prototype behavior. According to the Figure 79e is verified an amplitude increasing after 60seconds so that the maximum signal energy is verified near 10Hz (as observed in Figure 79a, and Figure 79b). Comparing the response with and without SMA pitch link is verified an amplitude reduction in all sweep span assigned to the SMA damping effect.

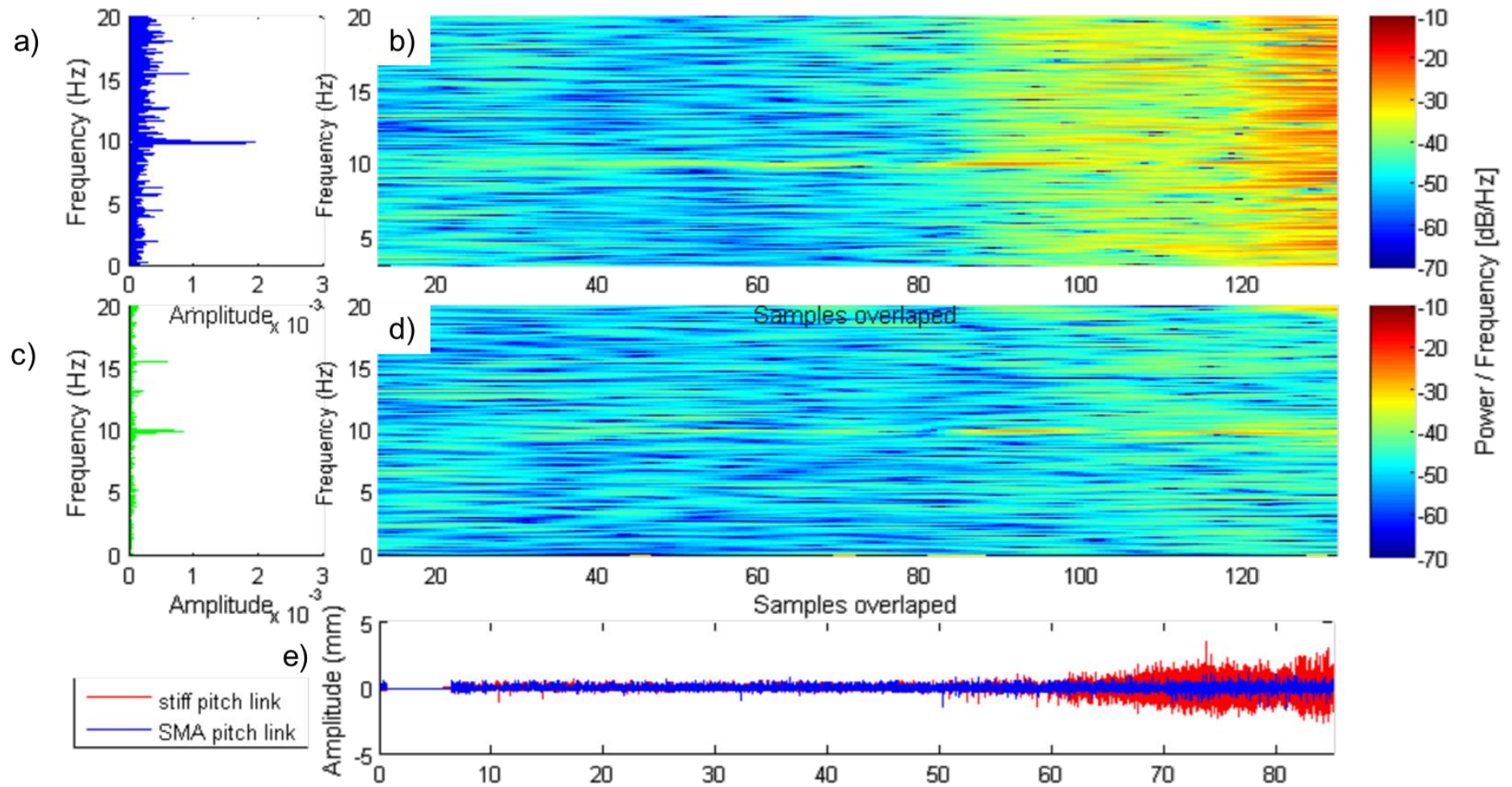


Figure 79 - Tridimensional response in sweep for 0-20 span during 90s: a) Power Spectrum Density with stiff pitch link; b) Power Spectrum Density of stiff pitch link; c) periodogram with SMA pitch link; d) spectrogram with SMA pitch link e) time domain comparative between stiff pitch link (red line) and SMA pitch link (blue line).

After the sweep, was executed other test with fixed frequency in 7Hz in order to observe the behavior in a frequency next to the blade flapping frequency and measure the RMS value from time signal. The parameters and the result of fixed frequency test are exhibited in the Table 14.

Table 14 – Test parameters and RMS result for fixed frequency with SMA pitch links were tested with the rotor in operation

Frequency	Pitch angle	Capture time	RMS (acceleration) (stiff pitch link SMA pitch link)
7Hz	12°	45s	0.2361 0.1975

The routine of tests induces the prototype to wear so that in last measurements are indicated the adjustments and replace the coupling between motor and main shaft, among other repairs. Thus, after a complete prototype maintenance revision, the number of tests may increase and should repeated to confirm the preliminary results.

5.11 Dynamic tests with cyclic commands shifting

According to treated in Chapter 2, the cyclic command is responsible to change the pitch angle of each blade, individually, in a fraction of the complete rotor revolution. The rotor model employed in this research works with three actuators, as shown the Figure 31 and Figure 33. Thus, in case of a unique cyclic actuator activating, simulating a rolling aircraft movement, the temporary shifting in the pitch angle, through the pitch link, works as a step function with the input force from the actuator and the output response on the blade.

The circumstance simulated in this test starts as a hover flight (rotor plane parallel to the ground) followed by a cyclic command in one of the actuators, holding this condition for 10 seconds. The main rotor frequency is 10Hz because this frequency is the secure limit for dynamic tests, according to the parameters performed in the item 5.10. The dynamic response is represented by the spectrogram and the timeline of the cyclic command, depicted in Figure 80. The cyclic command is activated after 15 seconds (Figure 80b) holding this position until 25 seconds. At the same time, the spectrogram performs a comparative in power during this period, with (SMA) and without SMA (stiff) pitch link (Figure 80a).

Assuming a reference point, placed in 22s, as detailed in Figure 80c (item 1 and 2), the SMA pitch link performs about 4dB/Hz in attenuation due to the cyclic input excitement. This attenuation is greater than observed in the results exhibited in Figure 72 (about 2dB) what suggests the attenuation

proportional to the input excitation displacement will be aligned with the fundamental theory about hysteretic damping of Chapter 3.3.

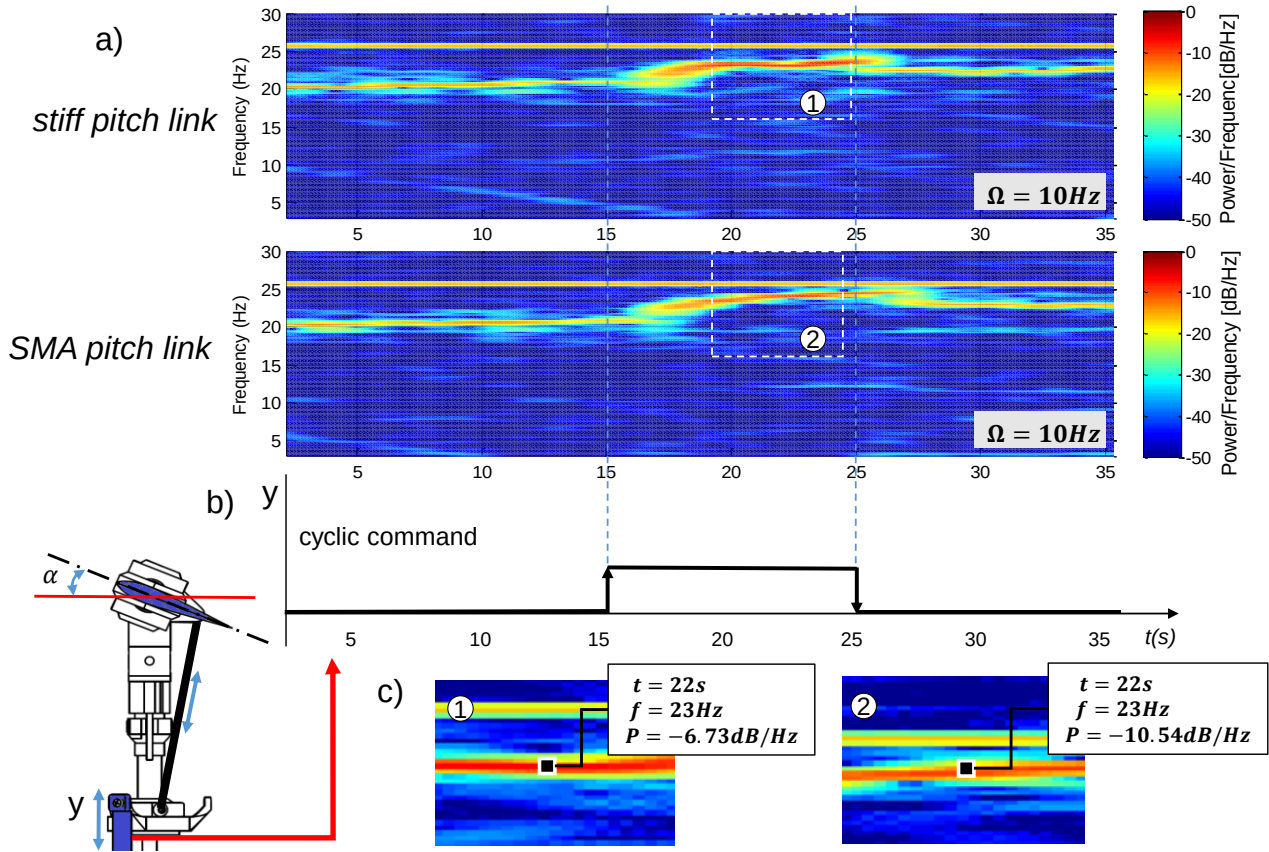


Figure 80 – Graphical representation of dynamic response due to cyclic activating: a) spectrogram; b) cyclic command timeline; c) detail of the spectrogram as a comparative with and without the SMA pitch link.

In order to simulate an extreme circumstance, where the experiment of Figure 80 would be repeated with the prototype under a lateral wind, as a continuous gust, the cyclic variation method was repeated with the prototype subjected to a lateral wind flow with constant velocity in 10m/s. This condition induces a dissymmetry of lift that exaggerate the asymmetric load effects applied by cyclic commands. The dynamic behavior is represented by spectrograms of Figure 81.

In all experiment long, the peak of energy is observed for two times the main shaft frequency, according to the experiment without lateral gust. Yet, when the system employs the SMA pitch link the peak energy is about 3dB/Hz less than using the stiff pitch link.

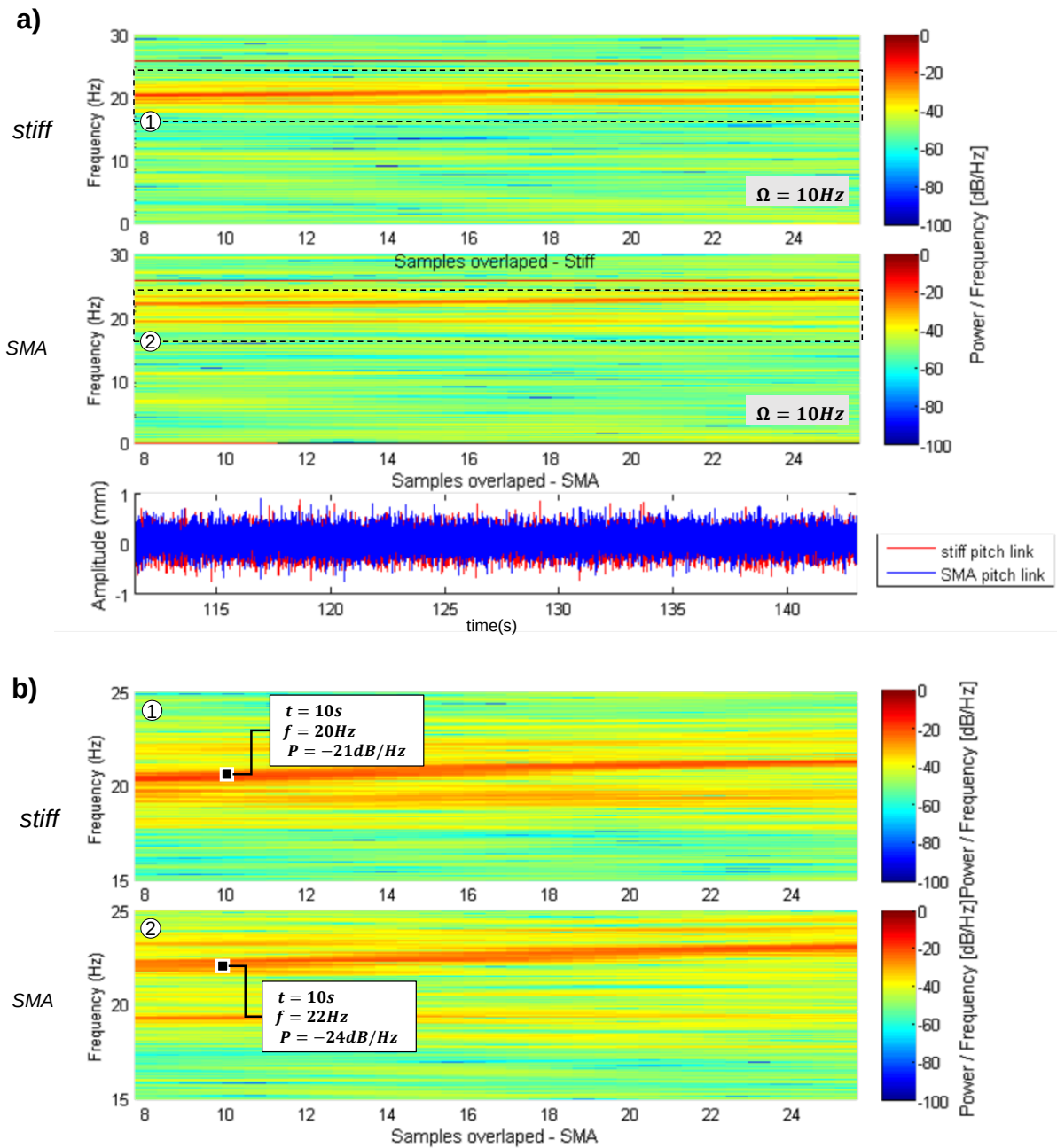


Figure 81 - Graphical representation of dynamic response due to cyclic activating under lateral gust: a) spectrogram comparing the performance without SMA pitch link and with this device; b) detail of the spectrogram as a comparative with and without the SMA pitch link.

5.12 Fatigue tests with SMA coil springs

Taking into account that the main component of SMA pitch link is the spring, closed and distalizing springs were tested in fatigue. The test was performed with each spring, without pitch link mechanism to avoid influence of other movable parts. Each spring was subjected to amplitude of 1mm (peak-to-peak) under a pre-strain in 300%, similar to when assembled in the pitch mechanism. The frequency adopted to fatigue tests was 7Hz because this value corresponds the critical

circumstance of pitch link operation due to the natural frequencies of the blades, disregarding the operational frequency of the prototype. The functional fatigue was performed with an electrical mechanic shake table, model Shake Table II of manufacture Quanser. The experiment is depicted in Figure 82.

In the experiment the shake table, performs the horizontal displacement, while one side of the support was attached on the table base and the other support end displaces under table movement according to a synodal signal in 1mm peak-to-peak amplitude, according to the expected operational conditions of SMA device. The displacement was measured with a displacement sensor. At the same time, a load cell registers the input force. The experiment was uninterrupted and performed under temperature of 21°C. The test took place with three units of each spring type.

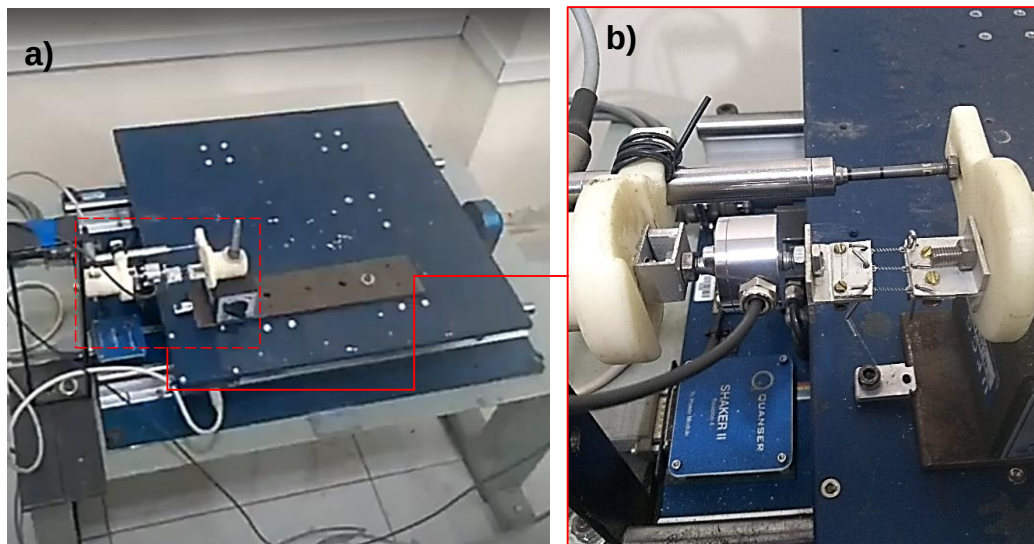


Figure 82 – Fatigue experiment in SMA springs: a) general view of shake table and supports assembly; b) assembly detail

The system HBM MX840A was employed as platform to registers the sensors under a sample rate of 100Hz, so that the number of cycles was counted at the final test based on the test spent time.

The results are presented from Figure 83 to Figure 86. In both experiments, the springs were subjected to the same experimental parameters, differing only on spring type: closed and distalizing coil spring. For closed springs, the result of Figure 83 indicates the stabilized behavior from $2 \cdot 10^5$ cycles to $6 \cdot 10^5$ cycles considering a stable maximum force for infinite cycles. The test was interrupted without structural failure in any specimen tested.

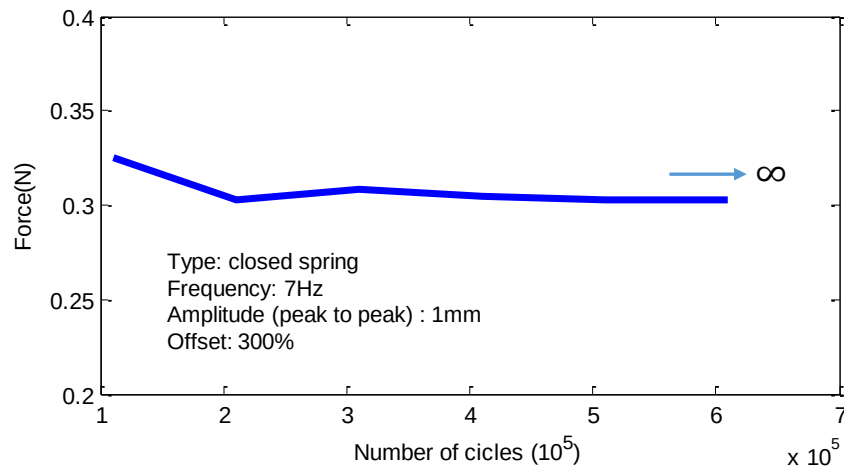


Figure 83 – Maximum peak force of closed coil spring under fatigue test stopped in $6 \cdot 10^5$ cycles, without structural failure

The force measured depends on the maximum elongation applied. This force tends to fall after the initial cycles and stabilizes until the end of the experiment.

In Figure 84 is depicted the evolution of superelastic behavior of closed coil spring demonstrating the hysteretic behavior until the end of the test. Taking into account that the load cell operates under tension during all test long, the force was registered in negative values.

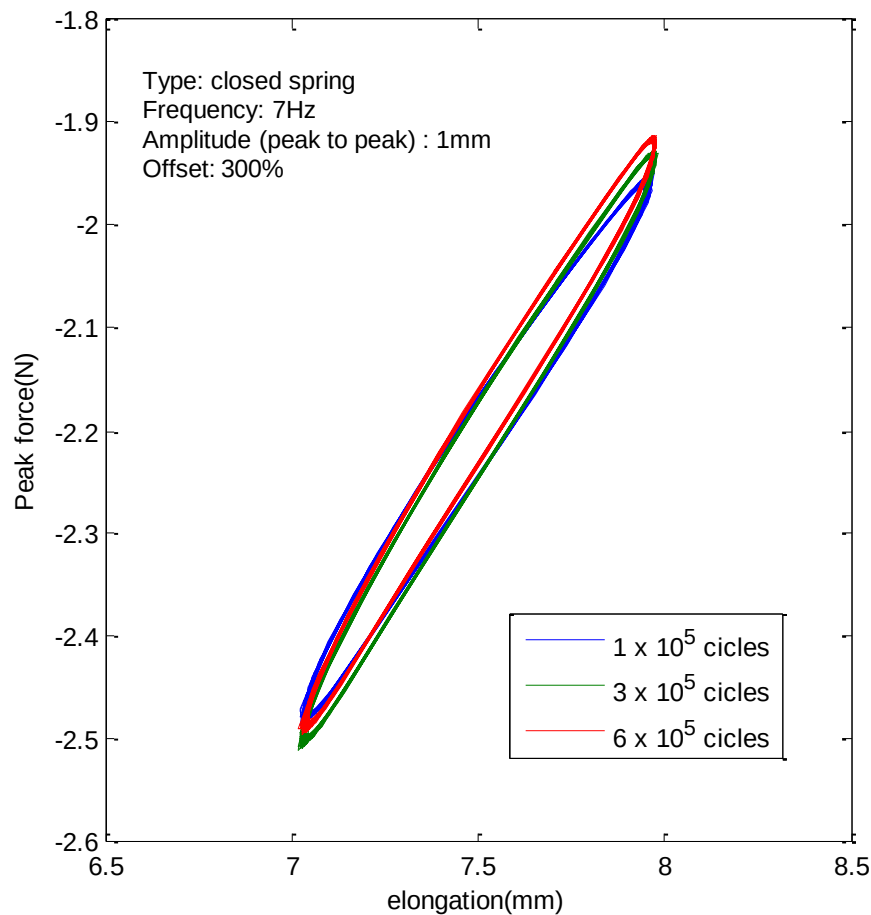


Figure 84 – Evolution of superelastic behavior of closed coil SMA spring with measurements stopped in $6 \cdot 10^5$ cycles without structural failure.

In Figure 85 is depicted the maximum peak force of fatigue test with distalizing springs. The test was interrupted due to structural failure that take place in about $0.6 \cdot 10^5$ cycles for wall the three specimens tested, exactly 10 times the less the closed springs. Similar to the Figure 84, the hysteretic behavior of the distalizing spring is depicted in Figure 86.

In practice, the tests employ only new orthodontic springs, acquired only for this fatigue test. In addition, any thermal treatment was performed.

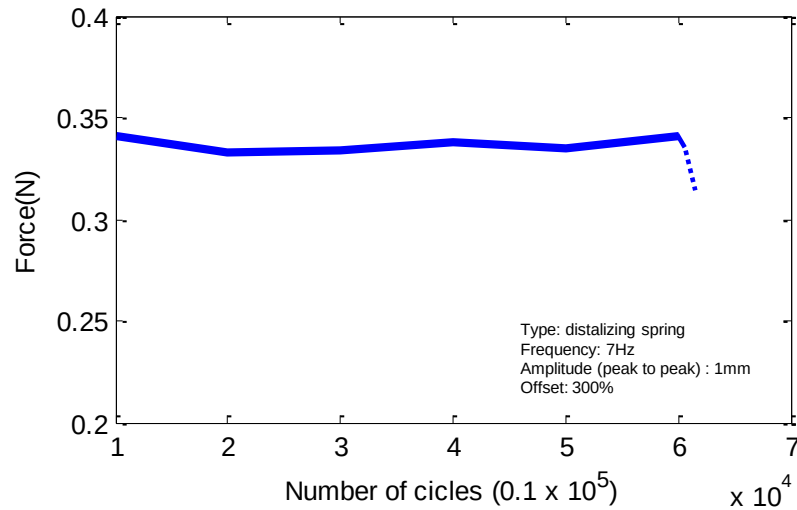


Figure 85 - Maximum peak force of distalizing SMA coil spring under fatigue test stopped in $6 \cdot 10^4$ cycles, after structural failure

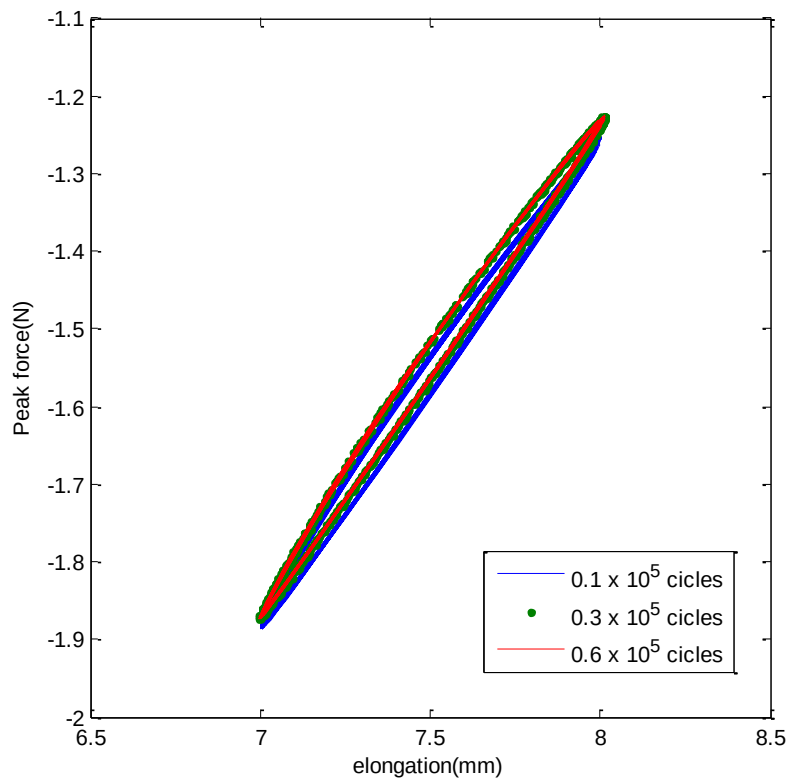


Figure 86 - Evolution of superelastic behavior of distalizing SMA coil spring with measurements stopped in $0.6 \cdot 10^5$ cycles after structural failure.

6 Conclusions

The chapter summarizes the discussion and conclusions about many points of this research, once despite the final results aimed with the general objective, the path to reach the final results rely with related results that it's worth to discuss.

6.1 Prototype performance

The whirl tower as a prototype aimed to perform the behavior of a helicopter in hover flight reached the proposed goal. The size of overall facility, including the protection net did not exceed the available workspace, dismissing several modifications in the laboratory layout, which represents an important heritage, aiming future researches. Cyclic and collective commands of whirl tower work with good response time been the control system a motivation for future improvement projects.

The acceleration measurements exhibit low noise, good compatibility with Matlab, and feasible a good spectrum analysis what testify the effectiveness of measurement systems as also as the kit accelerometer ADXL+ Labview feasibility for this application. The ADXL results precision were confirmed with the accelerometer PCB model 352B10, validated the results.

The blades manufacturing testifies the effectiveness allows experimental modal analysis with mechanical parts obtained by rapid prototyping open opportunities for improvements.

The natural frequencies of structure are far from Ω and $N\Omega$ frequencies performed in forced vibration tests with the rotor in operation. The dynamic behavior in operational tests confirmed the statements declared by the literature: so that not affected directed the overall dynamic response of rotor because the individual dynamic loads of each blade has to be canceled itself, considering a balanced rotor system. The modular constitution of the prototype opens the opportunity to its improvements in forward works.

6.2 The SMA pitch link proposal

The SMA pitch link demonstrates the performance of minicoil springs in vibration reduction employing SMA-SE even when the material undergoes few deformations. The tests using a pair of smart pitch links indicates satisfactory performance.

One of the challenges in this prototype is to perform large deformations (enough to induce hysteretic damping) with low displacements. The asymmetric geometry used in balancim (Figure 54)

ensure the boost factor in order to promote proper springs elongation even though imperceptible displacement between mechanism eyelets.

The horizontal area necessary to perform the balancim mechanism (Figure 65) may represent a drawback in the well geometry utilization for this particular application, differ from the slim geometry of a classic pitch link. Thus, this characteristic motivates the second-generation design in order to promote the proximity between the original pitch link design and the SMA pitch link.

6.3 SMA pitch link dissipated energy and stiffness

The primary function of a pitch link is to transmit the movement from the static frame to the rotating frame. In these circumstances, clearances and linkage length variations may cause delay in the commands (due to synchronism in the pitch angle changing, as detailed in Figure 8 and Figure 10), so that corresponds to a undesirable behavior. In other words, the ideal device requirement is to keep a non-changing eyelets distance (rod ends), like a perfect stiff rod.

In energy dissipation wise, both SMA pitch link version (with closed and distalizing springs) demonstrates few differences of 2.33%. Still, the rising stiffness of distalizing springs (344%) version suggests that this type of spring is the best choice for this pitch link geometry. Thus, the optimized version of SMA pitch link may employs distalizing springs to ensure the highest stiffness combined with the proper energy dissipation.

The best results with SMA pitch link, under the configuration presented in the experimental tests, reach the maximum energy dissipation of $0.263\text{MJ}/\text{m}^3$. Comparing these results with dissipated energy in preview researches, mainly those which uses micro coils SMA springs, as exhibited in Moraes (2017), was observed a maximum of $0.350\text{MJ}/\text{m}^3$ in dissipated energy, considering dynamic loading under low frequency (about 3.25Hz). In the same field, Reis(2018) obtains, results close to $1\text{MJ}/\text{m}^3$, in dynamic loading (about 15Hz). Both results (greater than the obtained in the current research) subjects the microcoil springs to large deformations, what explain these energy dissipation values.

6.4 The general performance of the SMA pitch link concept

The hysteretic damping is a function of the displacement amplitude as treated in the theory. Regarding to the SMA pitch link, the most severe conditions induces more energy dissipation, most perceptible in dynamic tests than the static tests, performed in bench. The tests extend the understanding of the poor influence of the frequency on the device performance and its large dependence on the displacement's amplitude.

Compared with an original stiff pitch link the stiffness of a SMA pitch link may change the natural frequencies of the structure, shifting it to minor values. Even though, the changing on the spring geometry may ensure a good energy dissipation without strict stiffness reduction. Thus, taking into account helical springs, geometric improvements may guide better results in terms optimizing the operation into the hysteretic area due to the phase transformation induced in the material.

Finally, the application field of SMA pitch link can be expanded to other systems dependent on rotating pitch angle shifting mechanisms like blades of aerogenerators and airplanes with variable-pitch propellers. In this sense, the concept was subject to a patent register due to large potential application field.

The major contribution of this SMA pitch link device is to improve the energy dissipation, with large relation weight/damping (considering all of the structure), without no engine energy consumption.

6.5 Suggestions for forwarding researches

Listed and summarized below are suggestions to future researches employing the damping concept registered in this thesis, as follow:

- Employ experimental methods to measure the vibrations and its amplitudes directed in the blades by wi-fi sensors installed on the blades or no-contacting technics that performs measures on the blade's deformations.
- Perform flight tests using SMA pitch link in helimodels instrumented to register the vibrating spectrum in flight.
- Explore new geometrical alternatives of SMA pitch link, aided by numerical simulation to stimulate new generations and pursue improved results in terms of energy dissipation.

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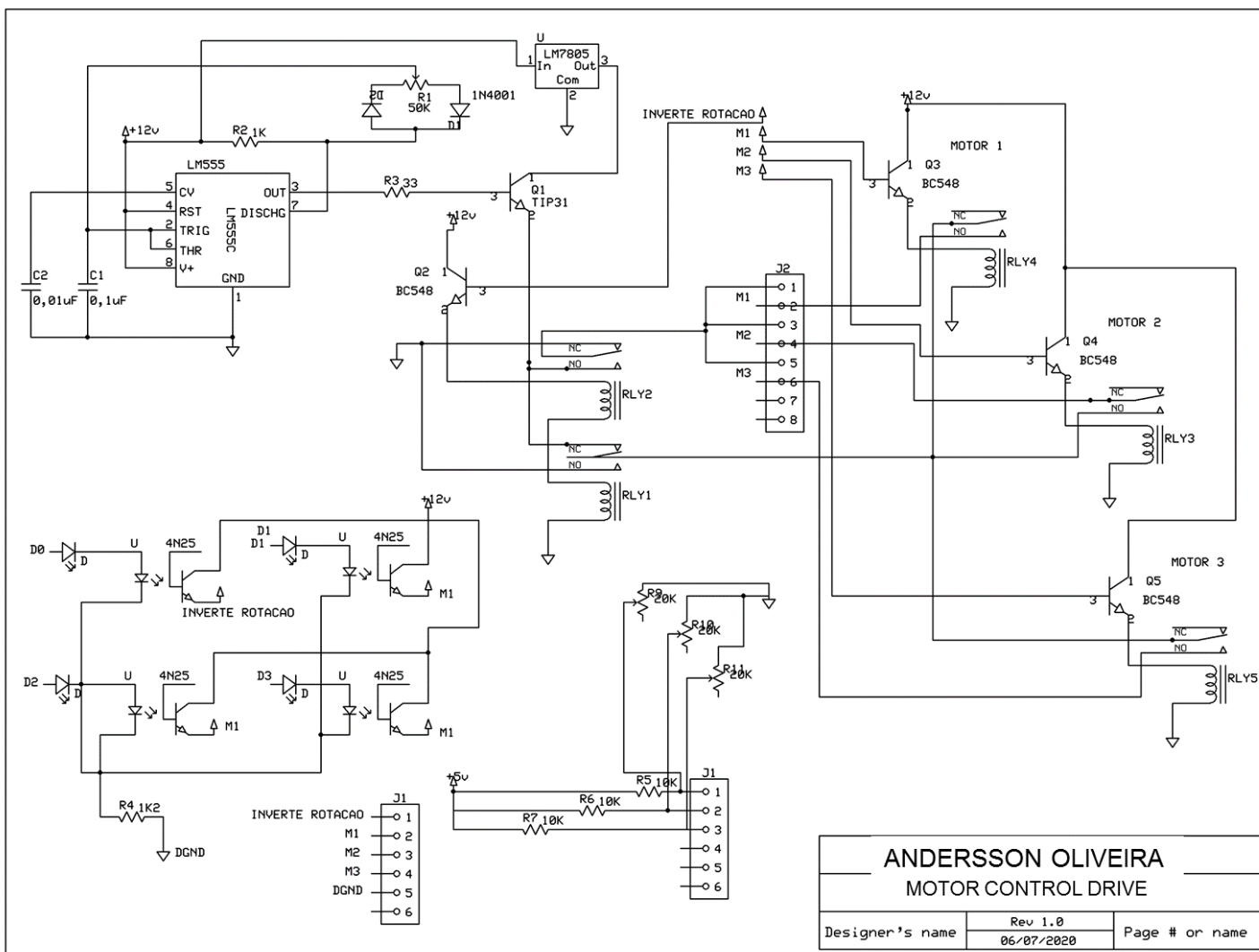
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APPENDIX A – Control board schematic



APPENDIX C – Determination of vertical input force of blade mechanism

The Equation (14c) describes the y_A coordinate relates to the horizontal position of swashplate linkage ball (y_{SW}) and position of rod l_1 .

$$y_A = y_{SW} + l_1 \cos \theta_{1x} \sin \theta_{1y}$$

The Equation (14d) describes the relationship between vertical position of the swashplate (z_{SW}), vertical coordinate of $z_{O_{AB}}$ and angle positioning of rod l_1 , thus:

$$z_{O_{AB}} + a \sin \theta_2 = z_{SW} + l_1 \cos \theta_{1x} \sin \theta_{1y}$$

Replacing $l_1 \cos \theta_{1x} \sin \theta_{1y}$ of (14d) in (14c).

$$z_{O_{AB}} + a \sin \theta_2 = z_{SW} + y_A - y_{SW}$$

Assuming $y_A = y_{O_{AB}} - a \cos \theta_2$, z_{SW} is:

$$z_{SW} = z_{O_{AB}} + a \sin \theta_2 - y_{O_{AB}} + a \cos \theta_2 + y_{SW} \quad (C1)$$

In hovering flight, considering experimental conditions, flybar position is constant, and $\theta_2 \approx \alpha$, thus:

$$z_{SW} = z_{O_{AB}} + a \sin \alpha - y_{O_{AB}} + a \cos \alpha + y_{SW} \quad (C2)$$

APPENDIX D – Calculus of internal maximum resultant force in rod l_3

Input data

- Length of segment “a” from AB (Figure 32): $a=25.2$ mm
- Length of segment “b” from AB (Figure 32): $b=13.8$ mm
- Length of rod $l_3=17.00$ mm
- Aerodynamic moment $DM_{o_{BL}}=0.0173$ Nm
- Servo input force in (7Hz), $F_{SW} = 0.5549$ N
- Angles obtained graphically by mechanism in CAD, for $\alpha = 10^\circ$: $\theta_1 = 4,94^\circ$, $\phi = 9,18^\circ$

The geometrical relations are established according to the model of Figure 59.

$$F_{l1} = F_{SW} \cos \theta_1 \quad (D1)$$

$$F_{l1} a = F_b b \quad (D2)$$

$$F_{l3} = \frac{DM_{o_{BL}}}{l_3} \quad (D3)$$

Based on free body diagram of rod l_3 , the resultant force is represented as Equation D4.

$$F_R = F_{l3} \cos \phi + F_b \cos \phi \quad (D4)$$

Substituting D1, D2 and D3 in D4, equation D5 express F_R in terms of external force F_{SW} , and external moment $DM_{o_{BL}}$.

$$F_R = \frac{DM_{o_{BL}}}{l_3} \cos \phi + F_{SW} \cos \theta_1 \cos \phi \left(\frac{a}{b} \right) \quad (D5)$$

APPENDIX E – Code to determination of aerodynamic moment and internal force in pinck link

```

clc, close all, clear all

% Aerodinâmic moment on the blade.

a= 5.7; % lift slope
ro=1.1839; % air density kg*m^-3 at 20°C
freq = 7; % main shaft rotation freq (Hz)
omega=(freq*2*pi); % main shaft rotation freq (rad/s)
c=0.040; % blade profile chord (m)
teta_graus=10; %pitch angle degree
teta_rad=teta_graus*(pi/180);
R=0.485; % blade length (m)
N=1; % number of blades

% lift force
T=(1/6)*N*ro*a*(omega^2)*c*teta_rad*(R^3) % considering a=5.7

DMobl=(c/4)*T % considering the center of pressure at c/4

% maximum compressive force under l3

a=25.2e-3;% lengh of segment a of bar AB (m)
b=13.8e-3;% lengh of segment a of bar AB (m)
l3=17e-3; % length of rod l3 (pitch link) (m)
Fsw=0.5549; % input force from servo - experimental (N)

Fr=((DMobl/l3)*cosd(9.18))+((Fsw*a*cosd(4.94))/b) % maximum axial
force under pitch link (APPENDIX D)

```