

Improving Helicopter Flight Simulation with Rotor Vibrations

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1 Introduction

1.1 State of the art

Modern engineering exploits the capabilities of simulation both for design purposes, both for training. Software packages of Finite Element Analysis (FEM) and Computational Fluid Dynamics (CFD) are useful tools to simulate the behavior of a body when stressed by a load or when immersed into a fluid. The simulation applications to training concerns for example driving, flying, traffic and queue, manufacturing and robot motion. Virtual Reality [1] has been also used to support simulation realism [2], both in immersive, and in un-immersive systems. The research in simulation issues focuses the attention not only on the dynamic models (which compute the behaviour of the phenomena to simulate), but also on visualization systems. Head Mounted Display (HMD) [3], large field of view cluster of projectors, hemispherical monitors are widely used to increase simulation realism [4]. Problems like image blending have been solved using algorithm producing a better overlapping of image sides common to two or more projectors. Also techniques of geometry correction has been developed to correct the image distortion on flat or round screens. Virtual environments [5] like CAVE for stereo projection and 3D visualization has been developed and applied to simulation. Also the use of sounds is wide spread in simulator. But view and hearing are only two of human five senses: sight, hearing, touch, smell and taste. Modern simulators try to exploit also the other senses to improve the simulation. Devices dealing with touch are called “haptic”; an example of these is force feedback on simulators commands. A classical application of simulation is the helicopter flight [6]: the motivations lies in the high cost of such a machine, in the difficulty of flight learning and in the need to try hazardous manoeuvres in safe conditions. The helicopter flight simulators can be used also to improve the design cycle [7]: test pilot can in fact fly a new helicopter model provided the dynamic model and the cabin internal layout. Flight simulators [8] are made of four main subsystems: the commands interface, which provide the man/machine interface, the flight model, which compute the physical behaviour of the vehicle, the visual interface which provide a panorama of the external view perceived during a real flight, and an instrument panel, which reproduce the gauges and indicator required for flight. The most advanced simulators include also a motion system which provide the tilt and roll of the simulator cabin to increase the physical involvement in simulation. The most common device for motion is the so called “Stewart Platform”, which is composed by a series of six electromechanical actuator connected in the top to three edges of the simulator cabin floor. Other techniques used for motion are based upon the use of electric motors which rotate the cabin providing angular displacements about the roll and pitch axis. But, vibrations are of straightforward importance in the helicopter flight [9]: the different flight phases can be in fact recognised by pilot through the vibration level perceived in cabin. Thus, the simulation of the helicopter flight can be improved a lot with the application of a device simulating the vibration felt by the pilot in a real flight. The vibration over an helicopter [10] cabin are induced principally by the main rotor, the tail rotor, the turbine (or the motor in case of piston engine) and also by gearboxes. The vibrations induced by tail rotor and turbine lie in the high frequency zone; the so called “full body” vibrations are mainly due to main rotor whose excitation on cabin presents a frequency which is a multiple of the product of rotor speed times the number of
1.2 This Paper

This paper describes the conceptual design, the dimensioning, the CAD modelling, the FEM analysis, the final manufacturing and assembly of a device useful to simulate helicopter rotor vibration. The design started from an experimental phase in which the vibrations recorded onboard a full scale helicopter have been collected and analysed in the whole flight envelope. The analysis of the collected data and of helicopters specifications taken from bibliography, provided a better understanding of the vibration phenomena and of its perception by pilots and passengers. The device design has been carried out applying the classical formulas of machine dynamics and mechanical dimensioning. The device has been finally added to an helicopter flight simulator developed at the Bologna University. Results obtained show the correct ending of the project and confirm the design methodology effectiveness.

In the next second paragraph of the paper, the design methodology is described; the third section contains a brief introduction to the problem of vibrations in helicopter and describes the experimental phase of in-flight vibration data collecting. A brief list of the University of Bologna helicopter simulator main features is presented in the fourth section; the product design cycle of the shaker is reported in the fifth section, starting from conceptual design up to the installation; a conclusion paragraph highlights the project results, the advantages in realism gained and the drawback of the adopted solution.

2 Design Methodology

The design methodology followed for this work is represented in fig. 1, where all of the phases are listed.

One of the most important design phase is the determination of the frequency and accelerations amplitudes range of interest. The simulator should in fact reproduce the vibrations perceived on board a large part of helicopters. The smallest helicopter are ultralight machines, without sophisticated devices for vibration suppression; on the other side, large helicopters present a better rotor mechanics, high power turbine (or turbines), takeoff mass up to thousands of Kilograms. One of the design requirements is so the modularity, as to gain the possibility to simulate not only one model of helicopter, but a large class of machines. It is very important to study the vibrating phenomena over several helicopters so to cover the most significant vibrations which can be found in the frequency domain.

This kind of research has been performed using a powerful mathematical operation called ‘Fourier Transform’ (described in the next paragraph), which allow to split a generic time domain signal into the sum of many different fundamental signals, each carrying a different single oscillation frequency.

Once the vibration amplitudes and frequencies have been investigated, the conceptual design of the device can start. Many solutions for vibrations reproduction have been considered, comparing technical capacity and drawbacks. The most promising solution have been deeply analysed in a more detailed design: after a first rough dimensioning, the device has been modelled within a CAD environment. In the following a series of FEM analysis has been deployed to check the design robustness. The virtual model of the shaker system has been added to a 3D model of the simulator to verify the installation and the interface with the simulator frame. The follow-up work has been the manufacturing of a prototype and its installation on the simulator. A final phase of testing concludes the design workflow.

3 Vibrations on helicopters

There are different important sources of vibration on board of an helicopter: They can be in general grouped into three main categories:
- Main Rotor (shaft and blades) induced vibrations;
- Engine and relative gearboxes vibrations;
- Tail Rotor (shaft and blades) induced vibrations.

Each of these mechanical groups contain a considerable amount of moving masses which rotate at a significant angular speed: main rotor speed is typically between 300 RPM for large helicopters, up to 600 RPM for ultralight helicopters. The turbine rotational velocity can be up to tens thousand of RPM, depending on turbine diameters. Tail rotor speed is normally a multiple of main rotor speed, since a rigid mechanical transmission synchronize the two rotors. Normal rates of transmission are variable between 4 to 7, depending on the tail rotor
diameter and helicopter’s features. Following this consideration, it’s immediate to understand that all these rotating masses can easily introduce very high level amplitude vibration phenomena on the entire helicopter structure. The main rotor vibrations [12] are mainly due to an asymmetric distribution of the unsteady aerodynamic forces on the blades: the exciting frequency lies in the range 12-40 Hz, depending on number of blades and rotor shaft rounds per minuteln order to better evaluate the entity of this kind of vibrations, an experimental campaign has been carried on. A triaxial accelerometer has been rigidly mounted inside the helicopter fuselage (Fig. 2) of a series of helicopters. Using this sensor, it has been possible to collect vibration data during different typical helicopter flight phases like ‘Hovering Out of Ground Effect’ (HOGE), ‘Hovering In Ground Effect’ (HIGE), climb, descent, and cruise at various speeds. The test equipment employed for vibration measures is made by: a triaxial accelerometer (+-5 g), a Signal Acquisition Module (with a sampling frequency up to 50kHz, 24 bit of resolution) and a PC for data storing and synchronizing.

It’s not possible to identify the characteristic frequencies of the helicopter used for the test from the acquired acceleration signal, as can be seen from Fig. 3. The use of a mathematical operator is necessary to transport the time-domain measured signals in the frequency domain: this operator is called the Fourier Transform. Practically, this operation splits a single time varying signal in a sum of contributes, each one at a different frequency. The transformation calculated is not a continuous signal integration, but an optimized approximation called Fast Fourier Transform (or ‘FFT’): a saving in computational resources is gained by this way.

Figure 3 shows the analysis of a Fourier transform of the Z axis acceleration for frequencies up to 1000 Hz: main rotor, tail rotor and turbine contributions can be easily found.

Focusing the attention on the first 100 Hz of the vibration spectrum, the fundamental frequencies of the main rotor can be found. The main rotor speed is in fact 380 RPM, so the corresponding frequency is about 6.36 Hz; the number of blades is 2. The harmonic main rotor frequencies will be so:

\[ f_{rotor} = k \cdot N \cdot \left( \frac{RPM}{60} \right) = k \cdot N / rev \]

Where:
- Number of blades (N): 2
- Main rotor shaft speed (RPMshaft) in RPM: 382 RPM
- Main rotor shaft speed (1/rev) in Hz: 6.37
- Harmonic frequency order (k): 1, 2, 3

It follows (for the first 4 harmonics):

\[ f_1^{rotor} = 12.7; f_2^{rotor} = 25.5; f_3^{rotor} = 38.1; f_4^{rotor} = 50.8 \]

The speed of the main rotor is normally kept constant by a regulator but the amplitude of the vibrations are highly dependent of the flight phase. Figures 3, 4, 5 highlight the difference in amplitudes between the phases of Hovering Out of Ground Effect (in which the rotor aerodynamic and the air flow is quite regular), the flight at cruise speed (for which the vibration suppression system has been designed), and the flight at maximum speed (VNE) phase. In this helicopter the 1-N/rev amplitude is lower than the 2-N/rev and 3-N/rev because of in commercial helicopter damping systems are installed to reduce the vibrations in the low frequencies which are well felt by the human body.
As it can be seen from the above figures, is clearly visible that the amplitude peaks are positioned at the expected frequencies. Furthermore, the amplitude of the vibrations induced by main rotor is variable with flight phase. Results obtained are in agreement with bibliographic data, as Figure 6 and 7 taken from Bibliography [13], [14], [15] shows. Figure 7 in particular show the amplitude of vibration of the 4 blade German helicopter BO 105 as a function of the speed.

3.1 Vibrations and human body

The body perception of vibrations is mainly due to the resonance of the internal organs, which is about [16]:
- 7-19 Hz for eyes and head
- 3-5 Hz for heart
- 3-6 Hz for internal organs like liver, spleen, kidney
- 5-12 Hz for spinal column
Furthermore, vibration can be divided in
- “full-body” (industrial machines, air ground and marine transports, normally passing through feet or seat),
- “hand arm” (Industrial pneumatic and electric tools hand held, like drill or hammer drill)

Full-body accelerations [17] presents medium-low frequencies (2-20 Hz) and involves the whole body; frequency at higher frequency are felt only locally by skin.

Helicopters without vibration compensation systems, and helicopters with small weight presents a peak of vibrations amplitude for the 1-N/rev frequency.

The most critical vibrations [18] for the human body lie in the range 4-6 Hz, as the international norm ISO-2631 [19] explains; the interpretation of graphs of iso-exposition to vibrations shows how a vibration of 6 Hz and 0.3 m/s² RMS amplitude can be sustained for 8 hours. The same value of exposition is prescribed for a vibration of 2.5 m/s² (which is 8 time greater) at 60 Hz.

As a conclusion, the most critical frequency in helicopter flight (with reference to body behaviour) seems to be the first N/rev. This frequency is not so far from the critical 4-6 Hz range. The other N/rev harmonics, even if in certain cases (large helicopters with vibration suppression devices) present larger acceleration amplitude, seems to be less “felt” by the human body.

4 Helicopter Flight Simulator layout

The Bologna University flight simulator is made up of a cluster of 3 PC connected with a local net providing: dynamics computation, external visual representation, flight commands acquisition. The implementation of the vibration simulation required to dedicate a PC to this task; the simulation speed is not affected by this new device since the other two PC are not overcharged. The main PC1 is connected to the helicopter simulator flight commands (collective, rudder, cyclic) by a USB port so that all the commands excursions are acquired in real time. The flight commands are in fact endowed with potentiometer sensors so that each displacement can be acquired using a dedicated acquisition module, and then used directly in the simulation program.

A mathematical model of the flight is implemented in the PC 1. Different helicopter models can be simulated: the mathematical model is parametric so that new helicopter model data can be loaded each simulation run. Position (Lat Long Alt), attitude (Pitch Roll an Yaw), speed and accelerations are sent to the PC2. The PC 2 provides the visualization of the external view (based on the free simulator Flightgear) and of the instrument panel. A second UDP connection links PC1 with PC3.
The mathematical model implemented for the helicopter dynamics [20] has been taken from books [21], reports and journal papers. The following Figure 9 shows the simulator view taken from the pilot seat.

5 Shaker design

The shaker design has been carried out following four steps: the conceptual design, the dimensioning, the CAD modeling, the FEM analysis. As presented in the previous sections, vibrations over helicopters cover a large spectrum of frequencies. But the most important ones are the medium/low frequencies. It is assumed that the shaker will reproduce only the amplitude of the first frequency of the rotor. This is a strong simplification which can be justified by the following matters:

- the contemporary reproduction of more than one frequency requires expensive tools and a complicated control system;
- the amplitude of the accelerations at the first rotor frequency of the rotor vary with the flight phase;
- the frequencies of gearboxes and turbine are typically high and stress the structure; but are well over the resonance of the human organs;
- the first rotor frequency depends on the rotor shaft speed and on the number of blades (which are data available for all helicopters), and the amplitude can be guess comparing data of similar helicopter. By this way, an approximate simulation can be performed also without experimental data of the helicopter.

5.1 Shaker conceptual design and layout

The main requirements for the shaker are maximum acceleration peak and frequency; the second is a fixed value once defined the helicopter model, while the acceleration peak should vary during the flight depending on the flight phase. The possible shaker solutions which were considered for the helicopter simulator application, have been taken from bibliography [22] and by technical solutions widely spread in mechanics like:

1. Shaker with fixed eccentric mass and vibration intensity regulation through elastic elements deformation;
2. Shaker with magnetic eccentric;
3. Shaker with crank shaft mechanism to change vibration intensity;
4. Shaker with eccentric variable through cam profile shape;
5. Shaker with ‘Watt’ type variable eccentric.
6. Electrodynamic shaker
7. Hydraulic shaker
8. Pneumatic shaker

The best technical solution selection process was performed considering the following parameters:

- Technical complexity of electrical and electronics boards and devices needed for the regulation of the vibrations amplitude;
- Mechanical realization complexity;
- Weight of the system;
- Integration capability within the simulator frame and structure (small dimensions);
- Shape of the produced vibrations;
- Final cost;
- Adaptability to a widest range of vibration frequency and amplitudes.

At the end of the evaluation process, the Watt type variable eccentric device was preferred since it seemed to be a low cost solution able to satisfy the requirements.

The device is made by two eccentric masses linked by two counter-rotating shafts, moved by an electric motor. The eccentricity of the rotating masses is changed by a stepper motor which acts on a maneuvering screw (see also Fig. 21). By this way, moving back and forth the shaker regulator slider different values of eccentricity can be obtained.

The eccentric can be equilibrated as to obtain low unbalance; this can be useful in high speed rotating rotors, for which a small eccentric mass is required. On the contrary, the maximum amplitude of vibrations can be increased by adding a larger mass.

The following Figure 10 present a conceptual study of the eccentric device; in green the electric motor, in red the max volume occupied by eccentric mechanism, frame in white and green. The shaker in fact is made up by an electric motor, two counter rotating masses and a frame. The frame is required to connect the motor and the eccentric mechanism; at the top a series of screws rigidly connect the frame with the seat. The whole system frame and seat are suspended over the simulator frame by a series of springs. These springs can also be replaced in a short time, permitting to simulate non standard helicopters behaviors.
5.2 Shaker dimensioning

The equation describing the motion of a system excited by an harmonic force is:

\[ \ddot{x} + c\dot{x} + kx = F(t) \]  

(1)

Where:
M: mass of the system  
C: damping  
k: stiffness  
F(t): exciting force

The shaker can be schematized as a vibrodyne: it is a mechanical system made by two counter-rotating masses with a given eccentricity respect to the rotation axis. The Figure 11 presents two masses (m1; m2) with an opposite rotational speed (Ω; -Ω), and with the same eccentricity (e).

![Fig. 11: Vibrodyne scheme](image)

The centrifugal force acting on the masses are \( Fc1 \) and \( Fc2 \), defined as:

\[ Fc1 = m_1 e(\Omega)^2 ; \quad Fc2 = m_2 e(-\Omega)^2 \]  

(2)

But the forces \( Fc1 \) and \( Fc2 \) can be decomposed on the axis x and y of Figure 13, so that:

\[ \begin{align*}
Fc1\_x &= m_1 e(\Omega)^2 \cos(\Omega t) \\
Fc1\_y &= m_1 e(\Omega)^2 \sin(\Omega t)
\end{align*} \]  

(3)

\[ \begin{align*}
Fc2\_x &= m_2 e(\Omega)^2 \cos(\Omega t) \\
Fc2\_y &= m_2 e(\Omega)^2 \sin(\Omega t)
\end{align*} \]  

(4)

Summing the contributions of the mass 1 and 2 in direction of the x and y axis it follows:

\[ \begin{align*}
Fc_{total\_x} &= m_1 e(\Omega)^2 \cos(\Omega t) - m_2 e(\Omega)^2 \cos(\Omega t) \\
Fc_{total\_y} &= m_1 e(\Omega)^2 \sin(\Omega t) + m_2 e(\Omega)^2 \sin(\Omega t)
\end{align*} \]  

(5)

If \( m_1 = m_2 \) and if a new variable \( m \) is defined as \( m = m_1 + m_2 \) it follows:

\[ \begin{align*}
Fc_{total\_x} &= 0 \\
Fc_{total\_y} &= me(\Omega)^2 \sin(\Omega t)
\end{align*} \]  

(6)

The shaker dynamic equation will then be approximated to a single degree of freedom vibration, where \( x \) represent the vertical translation of the system:

\[ \ddot{x} + c\dot{x} + kx = me^2 \sin(\Omega t) \]  

(7)

This equation can be solved (after an initial transient) in terms of displacement \( x \), as:

\[ x = \frac{\left[ \frac{me^2}{k} \sin(\Omega t - \omega) \right]}{\sqrt{1 - (\Omega / \Omega_s)^2 + \left[ \frac{2\zeta (\Omega / \Omega_s)}{\Omega} \right]^2}} \]  

(8)

Where:
\( \Omega_s = \sqrt{k / M} \)  
\( \zeta = c / 2m\Omega_s \)

The dimensions required for the design of the shaker is the product of eccentric mass per eccentricity and the spring stiffness; the requirements are the whole system maximum acceleration (in terms of [g] or [m/s^2]) and a
rotating frequency (the N/rev, in [Hz]). The unknown quantities can be found by solving the equation (7) and (8) or by a simplified methodology, valid only for a rough estimation of the mass per eccentricity product in early design purposes.

In this simplified approach the stiffness and the damping of the system are neglected, so that the motion equation became:

$$M\ddot{x} = me\Omega^2 \sin(\Omega t)$$  \hspace{1cm} (9)

Ordering the terms it follows:

$$\ddot{x} = \frac{me}{M} \Omega^2 \sin(\Omega t)$$

Which can be recognized as an harmonic dynamics of the type:

$$\ddot{x} = x \cdot \Omega^2 \sin(\Omega t)$$  \hspace{1cm} (10)

By the comparison of the equations (9) and (10) it follows:

$$\ddot{x}_{\text{max}} = x_{\text{max}} \cdot \Omega^2 , \quad x_{\text{max}} = \frac{m \cdot e}{M}$$  \hspace{1cm} (11)

Now considering that the maximum acceleration and the rotational speed are known, the maximum displacement x can be found. The product mass per eccentricity is then expressed like:

$$m \cdot e = M \cdot x_{\text{max}} = M \cdot \ddot{x}_{\text{max}} / \Omega^2$$  \hspace{1cm} (12)

The spring stiffness can be computed by the formula of the displacement in mass-spring forced system:

$$x_{\text{max}} = \frac{me\Omega^2}{(K - M\Omega^2)}$$  \hspace{1cm} (13)

The only unknown is the stiffness of the spring, while other terms are known.

As an example, with mass $M=150$ Kg, 1-N/rev=13 Hz and max acceleration equal to 0.5 m/s², we have:

$$m \cdot e = 150 \cdot 0.5 / (2 \cdot \pi \cdot 13)^2 = 0.01 Kg \cdot m$$

which can be obtained for instance using two eccentric masses of 50 grams, and an eccentricity of 100 mm.

The stiffness of the spring in this case will be:

$$K=990000 \text{ N/m},$$

And with 4 springs in parallel, the stiffness of each one will result:

$$K=250 \text{ N/mm},$$

which represent a deflection of 1 mm with a force of 250 N.

An helical spring with a 3mm wire diameter presents similar values.

5.3 CAD modelling

All the shaker parts have been modeled within a 3D CAD, as Figure 12 shows.

Fig. 12: CAD modeling of the shaker, with its eccentric masses

All the parts of the 3D model has been assembled and coded with labels to allow the further assembly, as Figure 13 shows.

Fig. 13: Assembly drawing of the shaker

A series of 2D drawings have been realized to product parts. Materials used to build the device are: steel, nylon, aluminum.

Fig. 14: 2D constructive drawing of one of the shaker components

Also the frame of the shaker has been modeled as Figure 15 shows.
5.4 FEM analysis

A Finite Element analysis was required to check dynamic interference between the simulator frame and the shaker frame. The software used is Patran for preprocessing and post-processing, and Nastran as the main solver. For instance, the model of the shaker support structure has been modeled within the Patran software: mono-dimensional ‘beam’ type elements have been applied to simulate all of the metal tubes and sections in the structure. A single concentrated ‘mass element’ positioned on the motor CoG and joined to the support plate through ‘rigid elements’ has been used to model the motor itself. The complete model is made of 422 nodes and 458 elements, for a total of 2261 degrees of freedom (DOF). Element types used in the model are: Beam, Plate, Mass, Rigid.

Beam elements are used to represent tubes and bars, with the following dimensions: “L” type beam dimension 50x50x5 mm, Plain beam of 40x5 mm, Plain beam of 50x8 mm, Plain beam of 55x10 mm, Plain beam of 50x10 mm, Square beam of 40x40x5 mm, Bar with a diameter of D=20 mm.

The material used in the FEM modeling process is carpentry steel which has been chosen for cost and for weldability. Its main characteristics are:
- Young Modulus: 2.1 e11 N/mm²
- Poisson coefficient: 0.29
- Density: 7850 Kg/m³
- Max design Tension: 200 N/m²

The model (Fig. 16) constraints consist in 4 joints (each of them blocking all 6 degrees of freedom) positioned in the upper part of the structure, at the connection between the seat and the simulator.

The natural vibration modes have been found to be:
- Mode 1: 49 Hz
- Mode 2: 64 Hz
- Mode 3: 78 Hz
- Mode 4: 146 Hz

Among all these modes, the most interesting are those presenting a deformed which agrees with the direction of the shaker harmonic excitation force, because they can be subject to resonance.

5.5 Electric Motor

The electric motor needed to move the eccentric mechanism has been selected considering a maximum frequency for the first harmonic of 40 Hz. The torque of the motor is not a mandatory requirement since the only reaction is given by the friction of the bearings. Transitory dynamic is not so fast; the changes in amplitude of vibrations are small due to the inertia of the helicopter, which speeds up and decelerates in quite long times.

The characteristic of the motor (in Figure 18) are:
- maximum power output = 550 Watt;
- single phase, bipolar electric motor;
- max angular speed of 2440 RPM
- endowed with an inverter for speed settings.

The stepper motor is small enough to fit into the assembly: the torque required is very low and the correct angular position is provided by a look-up table which sets the number of motor steps needed to move the screw in order to obtain the desired eccentricity. A control board is linked to PC3, which drives the stepper motor and computes (in open chain loop) the current and required
position of the eccentricity. A calibration procedure is necessary to set the correct initialization of the control system.

Fig. 19: Stepper motor: CAD and real

5.6 Manufacturing and assembly

All the parts have been designed and manufactured aiming to lower the costs. The chosen materials are aluminium for non structural parts to be machined; nylon for non structural parts to be worked by lathe; steel for structural parts. Moreover, a large use of commercial items has been considered for bearings, screws, springs. The following Figure 20 shows the work in progress of bearings covers and support plates.

Fig. 20: Manufacturing process of the shaker radial bearing enclosures by automatic machine

In the following Figure 21, there are some picture of CAD and realized parts.

Fig. 21: View of the shaker eccentricity regulator (down) and housing for bearings (up) CAD model (left) and after manufacturing (right)

Finally, the shaker has been assembled and installed in the simulator, as Figure 22 shows.

Fig. 22: Image of the shaker frame (bottom part with shaker and motor- left) and upper part (connected to the seat-right)

The assembly seat / shaker has been suspended on the simulator frame by four helical springs.

Fig. 23 Shaker group (seat+frame+motor, in the left) suspended on simulator frame by springs (in the right).

6 Final Result

The following Figure 24 presents a CAD image of the device installed on the simulator structure.

Fig. 24: 3D CAD view of the simulator

The simulator has been evaluated by an operator both in static mode, and with vibration shaker on, returning a realistic feeling. In spite of the lack of higher frequency replication, the simulator has been successfully tested by several pilots with positive comments in terms of realism.
7 Conclusion

This work presents the workflow of a design activity which led to the implementation of a device to be used in helicopter flight simulators to reproduce flight vibrations. Helicopter pilot is in fact accustomed to feel different levels of vibrations during flight, depending on flight phases. The largest part of current simulator is not provided with such a device owing to high cost and complication of electrodynamic shakers.

A device useful to reproduce the first frequency of the main rotor has been so designed and tested. The design of the device started by the analysis of the typical frequencies felt inside the cabin during the whole flight envelope. Next, the data has been analyzed as to define a list of requirements. A conceptual design phase led to the definition of the shaker configuration; a 3D modeling and a further FEM analysis campaign was performed in the detail design phase. The manufacturing and the assembly of the device allowed the installation of the device over an helicopter simulator. The whole system have been implemented and tested; the reproduction of vibration increases the similarity with real flight and contribute to the realism of the simulation.

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References