DESIGN AND MANUFACTURE OF DYNAMIC ASSEMBLIES AND COMPOSITE DRIVE SHAFTS FOR TILTROTOR

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Abstract

The submitted paper aims to present the development of the components of a mechanical drive line using composite shafts between an engine and a rotor. This project is part of the STEADIEST (Supercritical Composite main Drive SysTem) project funded by H2020 Clean-Sky 2, July 2019 to November 2022. Project team was made up of Nexteam group, Conseil & Technique (compagnies in the aircraft industry - France) and Arts et Métiers (University - France). There were two objectives during this work. The first one was to identify and implement a global design method able to return optimized product by taking into consideration costs induced by design choices. The second objective was to develop and demonstrate the validity of our design process through STEADIEST case study. A design method has been proposed and demonstrate a good performance in solution space design exploration, leading on best technical choices based on current information. Finally, STEADIEST case study has been validated through critical frequency analysis and experiments.

1. NOTATION

Here are a quick overview of the symbols and acronyms used in this paper.

Symbols:

E	Young's Modulus, MPa
ρ	Density, kg/m3
SF	Safety Factor, -
Ω, Ω_{TR}	Main, tail rotor speed of rotation, rad/s

Acronyms:

NDOE	Numerical Design Of Experiments
AR	Architecture Review
PDR	Preliminary Design Review
CDR	Critical Design Review
FEM	Finite Element Modeling

2. INTRODUCTION

Tilt-rotor aircraft, or conventional helicopters, must solve the problem of transporting energy between the engine and the rotor over distances of several meters. In general, a single shaft is impossible. It is then necessary to use different sections which must be connected to each other. Drive shafts have the primary function of transmitting motion and torque. They must be capable of respecting a certain number of criteria in static or in fatigue, healthy shaft or damaged shaft. Connection elements such as supports or couplings are also to be integrated into the design process. And of course the entire rotating system must not generate vibration or instability problems. The design activity for the manufacturer is to choose an architecture, in the sense of the choice of the number of sections and the components used, which is efficient on all the dimensioning criteria mentioned above but also which makes it possible to minimize the mass and all costs (recurring and non-recurring). Many studies relate to the dynamic aspects, by proposing subcritical or supercritical shafts, by comparing metallic materials (steel or aluminum alloy) or composites. These studies mainly concern the automotive and aeronautical sectors. A certain number of articles are interested in the phenomena of instabilities induced by internal damping in the case of composites. For the dimensioning of composite shafts, a lot of papers present dimensioning methodologies according to the composition of the composite and the results obtained for the torque resistance or the calculation of the frequencies of the shafts. Few studies present a global vision of the design of the transmission chain by taking into account the different performance criteria (torque, vibration) associated with the criteria of mass, development time, cost to define a design approach.

The objective of this work was twofold. Develop and demonstrate through tests a concept of a transmission line of composite shafts with a high level of technical performance (low weight, user-friendly support, improved dynamic behavior reducing vibrations) and to present a global design approach allowing the optimization of costs (architecture and interfaces between parts, reduction of references and design to cost approach).

3. LITERATURE REVIEW

3.1. Driveline design methods

Minimizing the overall weight of aircraft (planes, tiltrotors, helicopters or even drones) has always been one of the main priorities in aircraft product development, including transmission driveline development. Supercritical driveline, including shaft(s) working above one or many critical frequencies, is experiencing renewed interest because a higher operating speed means a lower required torque with a constant power, thus a thinner or lighter shaft. These increased lengths also allow a reduction number of bearings, supports and couplings, but these weight gains are counterbalanced by important levels of vibration near the critical frequencies and the need of a damper to help driveshafts to accelerate through this critical speed. Wang et al. (Ref.3), as Huang et al. (Ref. 4) do, have detailed damper design recommendations for these types of supercritical shafts – which are increasingly deployed on aircraft.

In order to improve vibration and instability predictions, advanced models of supercritical drivelines have to be developed so as DeSmidt et al. (Ref. 5) and Huang et al. (Ref. 6) have studied the stability of a supercritical driveline made of two supercritical driveshafts connected by a flexible coupling. These studies deal about the influence of internal and external damping, misalignment and different types of loads on driveline instabilities. Their works provide a numerical model to predict instabilities therefore to prevent them since the driveline design.

An alternative to the supercritical driveline can be found in the literature about a single-piece helicopter driveshaft. Ross (Ref.1) and Henry (Ref. 2) present this driveshaft concept and optimization regarding whirling instability, composite capability to transmit torque and torsional buckling. Self-heating of the composite, induced by the flexion recovered by the shaft due to the lack of flexible couplings, needs be considered because of its impact on composite strengths.

3.2. Drive shaft design methods

It has been demonstrated that composite shaft internal damping is higher than metallic, and it could lead to instabilities as studied by Sino (Ref. 21), Sino et al. (Ref. 22) or Jacquet-Richardet (Ref. 23). Analytical models have been developed by Montagnier and Hochard (Ref. 34) and Montagnier (Ref. 25) in order to gain a better understanding of the vibrations or instabilities appearing in dynamic. Their models give a good indicator of instability for viscous and hysteretic damping but remains dependent of the chosen damping model since they cannot be simultaneously taken into consideration, neither frequency dependent damping. Other studies have been carried out with finite elements formulation like Mendonça (Ref. 26), Sino et al. (Ref. 22), Ri et al. (Ref. 27), Ben Arab et al (Ref. 28) and Ri et al. (Ref. 29) to better simulate and predict instabilities induced by an important internal damping with simple but efficient numerical models.

Regarding composite driveshafts design methodologies, they mainly rely on a single or multi-objective optimization algorithm. Wang et al. (Ref. 10) have summarized the main algorithms used for optimizing composite designs. Generic Algorithms (Rangaswamy et al. (Ref. 11), Rangaswamy et al. (Ref. 12), Roos and Bakis (Ref. 13) and Montagnier and Hochard (Ref. 14)) are the most applied, followed by Ant Colony, Particle Swarm Optimization (Manjunath and Rangaswamy (Ref. 15), Ok and Turkmen

(Ref. 16) and Hernandez et al. (Ref. 17)), Simulated Annealing (Gubran et Gupta (Ref. 18)), Artificial Bee Colony (Karimi et al. (Ref. 19)) and many others. Most of these design optimizations are made on automotive driveshafts, but method and tools can easily be extended to helicopter driveshafts. Composite driveshafts parameters to optimize are the stacking sequence and number of plies, based on an external diameter and a length of the driveshaft predetermined. Most of these studies aim to compare weight gains against an already existing metallic driveshaft, which is often higher than 50% of weight saving, up to 80%.

To come back to the global design strategy, Geromin (Ref. 7) has developed a methodological approach to design short flexible drive shafts based on different levels of maturity and on the integration of expert's knowledge. His work focused on short flexible high speed driveshafts design, whose specifications can differ from a driveline driveshafts design, but its main function remains torque transmission. His work is one of the few focused on the development workflow of a transmission driveshaft. To design composite for an identified level of stress, Castanié et al. (Ref. 8) summarize preliminary design method for shell composites based on Gay (Ref. 9) composite theory. This preliminary design can be applied for the tube section if the curvature can be assumed neglectable and aim to define the number of plies in each principal direction 0°, 45° and 90° to withstand the applied stresses, mainly shear stress induced by the torque and stress induced by centrifugal effects.

Only one study integrating manufacturing considerations has been found regarding composite driveshaft development. Feng et al (Ref. 20) have developed a fully composite tube with integrated composite flanges and they have detailed the manufacturing process. A Finite Elements Model has been developed to predict bending stiffness of the tube, and efforts have been made to predict with accuracy the loss of stiffness induced by fibers expansion in the flange during the molding, which is a manufacturing consideration used to better estimate final product properties.

3.3. State-of-the art summary

Several elements regarding the design of a transmission chain have been considered in the literature review above. In the first place, driveline transmission design and models have been studied. Then a review of driveshafts, mainly with composite material, has been carried out regarding the criterion of design, the methodology and their optimization.

From this analysis, it appears that there is a lack of studies referring to a global vision of the design of a transmission chain, including performance criteria like torque transmissibility or level of vibrations, but also design and development time and costs depending on technical choices made like the choice of an architecture and interfaces. Last but not least, literature does not abound about driveshaft design taking into account design specific knowledge from previous product development or manufacturing constraints, while it may cause heavy changes later in the development.

4. DESIGN METHOD

4.1. Overview

The state of research emphasizes that there is a need for a driveline chain design method able to take into account not only the functional specifications, but also the manufacturing constraints induced by tools, knowledge or any equipment at the company's disposal. In addition, market expectation and customers expect always faster and more efficient product development process, so there is a need of efficient composite driveshaft design method. There are already some transmission systems design processes but most of them remain focused on a local vision like the optimization of the sizing based on few criteria (up to half a dozen) or a short flexible shaft while a global design method is becoming needed to keep improving the design of these systems.

4.2. Product development process

The book written by the Defense Acquisition University (Ref. 30) outline the main phases of the product development process. It helps to evaluate each level of product development and control the advancement (including System, Item Performance, Item Detail, Process and Material specifications) thanks to technical reviews (SRR, PDR, CDR, TRR, SVR and many others). An overview of a typical project management, particularly in aeronautical and defense industry, is illustrated in the following figure.

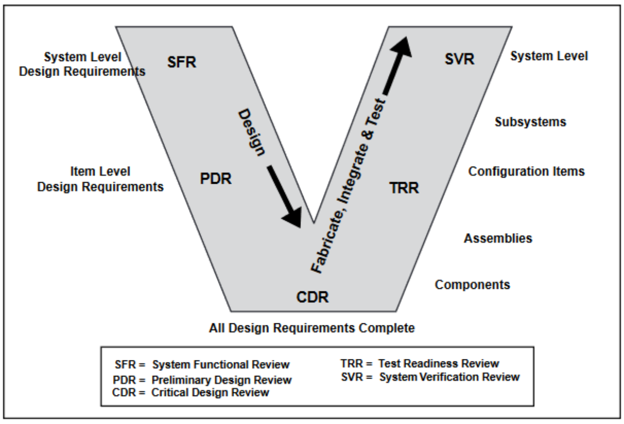


Figure 1: Systems engineering and verification extracted from (Ref. 30)

Pahl et al. (Ref. 31) had identified these product development main phases with planning and task clarification, conceptual design, embodiment design and detail design. They didn't integrate prototyping and production ramp-up in their overall process because their information or constraints may be needed during the whole design process. This global product design development process completed with project reviews is widely used in the industry as well as in scientific research like Tan et al. (Ref. 32) to study the impact of a change in the design depending on its advance.

4.3. Structured design approach

Our proposed driveshaft design process is also based on this decomposition in architectural principles, preliminary design, conceptual design, test program definition, prototyping and tests then a pre-serial launch if everything is conclusive, all with a control milestone at the end. The proposed methodology is focused on the two first processes detailed below, the architectural principles and the preliminary design phases. An overview of this design process is presented in the following illustration.

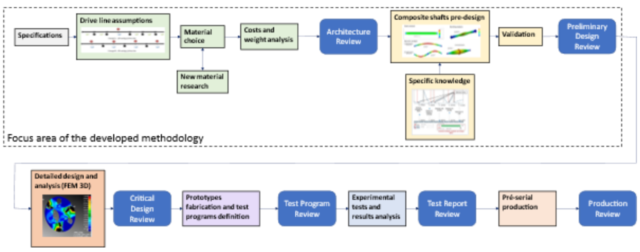


Figure 2: Composite driveshaft development process overview

4.3.1. Architectural principles

The very beginning of product development process starts with the reception of the technical specifications to fulfill as rich as possible. In most projects, these data come sparingly and engineers have to deal with incomplete and sometimes wrong information. Get the final specifications is an iterative process itself.

After receiving the specifications, one can begin to describe the driveline structure with different components. There are four main types of components which are shaft, support, coupling and interface pieces. Each one of them can have different variations or definitions, like a metallic or a composite shaft, or different geometries of coupling. In the first place, most of the value used will be arbitrarily chosen and will be gradually replaced by values obtained by tests or given by the supplier of a reused existing component. An important feature of this methodology and associated tools is keeping a trace of all the value used, even as assumptions, in order to validate values chosen. For instance, critical assumptions mainly concern support and structure stiffnesses, coupling stiffnesses, damping values of materials or boundary conditions. An assumption it also made on overall dimensions of shafts (length, outer diameter and thickness), il order to prepare a Numerical Design Of Experiments (NDOE) with various values of Young's Modulus and density.

Maintain critical frequencies of the driveline out of the avoidance bands is one of the most constraining design criteria in the majority of cases, and this is why it is the first criterion used to explore different architecture concepts. The NDOE aims to return to the designer a graph with bending critical frequencies of the transmission driveline function of specific modulus according to the following formula:

$$\text{(1)} \qquad Specific\ modulus = \frac{E}{\rho}$$

This method aims to use NDOE to build chart in order to help designer to identify suitable materials for a given architecture. This NDOE varies Young's Modulus and density of the shafts since these two parameters are used to build the specific modulus according to equation (1). Two NDOE can be computed, one with all identical shaft properties and one with two different shaft properties composing the driveline and can be visualized on the following figure.

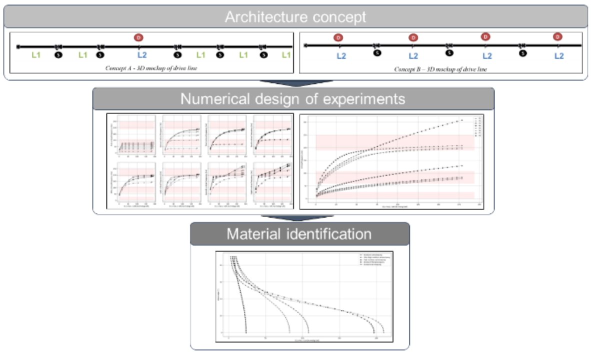


Figure 3: Process from architecture concepts to material identification

This figure details the NDOE process with two examples of driveline architecture, the left one made up of 2 different shaft properties (and length, but any length can be used) and the right one with the same shaft properties (and length) for the whole transmission. Results of the NDOE, plotted in graphs, can be observed above these architectures concepts. All bending critical frequencies can be represented in the same graphic if the driveline is compound of a unique shaft properties, as presented in the right-side plot of the NDOE results. Conversely, two different shaft properties lead to multiple graphs, each one displaying a bending critical frequency of the driveline, depending on the properties (specific modulus) chosen for one shaft. Three different shaft properties or more has not been implemented because such a configuration seems not in accordance with a costs rationalization strategy, where a maximum of identical components is researched to minimize manufacturing costs and facilitate maintenance plans. Designer's work is to identify interesting properties range that match critical bending frequencies avoidance bands.

Whereas this NDOE gives a direct indication about the compliant metallic materials, a transformation is needed to obtain final composites candidates. In fact, the Young's Modulus used to compute the critical frequencies of the transmission driveline corresponds to a homogenized Youngs' Modulus of the stacking sequence for a given composite material, according to the Gay theory (Ref. 9). It means that a stacking sequence has to be found for a chosen composite in order to match a specific modulus compliant with the avoidance bands. This stacking sequence is, by assumption, an even number of symmetric plies stacking at plus and minus a fixed angle. This assumption, coming from manufacturing constraints, is formalized knowledge capitalized on the experience of developing multiple composite products. Once specific modulus ranges are identified, the designer can compare them to the internal composite material database. A graph with winding angle depending on the specific modulus of the known composites is available and allow him to select suitable composites associated with wrap angles. In addition to the fitness of the composite material and its stacking sequence, attention needs to be paid to the compatibility of the material and the operational environment such as temperature, humidity conditions and ability to handle impacts without mechanical properties loss. If there isn't any suitable composite found suitable for the current architecture, two ways are possible: firstly, research of a new composite that can fit the specifications are made, secondly the architecture is changed to match known materials in the database.

Finally, when a material and the stacking sequence of the composite tube are identified, a preliminary cost and weight analysis can be made for the transmission architecture. Weight analysis is based on calculation of the sum of every component mass of the driveline, including fixation elements (bolts, screws, …), friction bush and balance rings. This weight can then be compared to a target weight of rate of mass saving defined in specifications. Costs analysis is made regarding our knowledge capitalization about manufacturing forecasts, mold cost, material cost, shaft manufacturing cost including scraps rate and unitary price for each component. The identification of a compliant solution regarding frequencies specifications, weight saving and with an interesting business case will lead to further analysis and development of the composite driveshafts. Even if the estimation is rough it allows to select between various interesting architectures. During this selection step, other criteria than cost and weight can also been taken into account like the risk inherent in each architecture or the engineering complexity for instance, and these criteria can be extended and refined by capitalizing specific knowledge.

4.3.2. Preliminary design

The preliminary design phase begins with a validated Architecture Review (AR). Several design calculations have to be made before the Preliminary Design Review (PDR) could be successfully passed, and concern every component of the transmission. PDR milestone could be achieved when the design is frozen and only needs tiny adjustments but the confidence in architecture is high enough to prevent any important design change. This method aims to give a precise overview and focus on composite driveshaft preliminary design, other components design – like splines, fixations like bolts, or any other element helping torque transmission, support's design or their bearings – will be set aside. This paper will focus on the sizing of the composite shaft, through the following sizing activities, but every new product needs to be validated by calculation. An overview of the different calculation made is presented in the figure below.

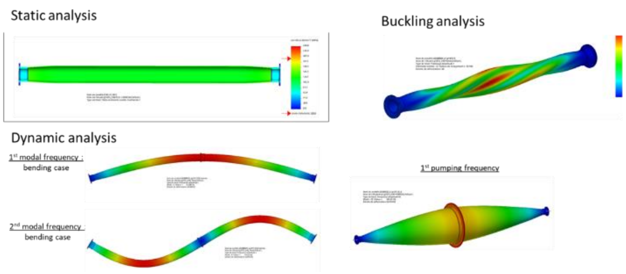


Figure 4: Various calculations made on composite driveshafts

Three main design activities will be detailed next: calculation of buckling critical torque, calculation of critical frequencies and calculation of torque transmissibility.

Buckling critical torque of the composite driveshaft needs to be above the ultimate torque applied on the transmission to prevent any instability induced by the loading of the shaft. This critical torque is hard to assess because most of composite failures happen because of delamination or fiber failure before buckling. The proposed design method aims to capitalize on experiments made, identify if the failure mode is due to buckling in order to improve analytical and numerical models. The more experiments we have, the more accurate the models will be.

Bending critical frequencies have to be computed more precisely than during the first phase of development. Finer analysis will be made with 2D and/or 3D Finite Elements Modeling (FEM) considering the real geometry and properties of each component. This calculation has to be realized on subassemblies of the driveline (isolation between two supports), on subassemblies representative of test bench configuration (should be the same as the previous subassembly but it can differ because of interfaces of the test bench), and on the whole driveline system. As it was previously said, bending critical frequencies are mostly one of the hardest specifications to respect. Knowledge capture also works well to help improve fidelity of models used during this design activity, because various experiments can enrich the simulation. Firstly, impact test on free-free shafts can be performed to identify bending and axial critical frequencies of the shaft alone, without the influence of any boundary conditions. This test allows to identify mechanical properties of the shaft to recalibrate models. Then, impact test or vibrating pot test can be used to perform the same kind of analysis but on a segment of the transmission with couplings and mounted on supports, integrating more representative boundary conditions. The correlation of these segments increase confidence in numerical models in order to validate full transmission driveline system, because the whole system is often hard to evaluate on a test bench because of its size and the number of components required.

Axial and torsional critical frequencies could also be checked although not as important as bending ones, because an external excitation could be near the system and vibrations caused by the eigenmode could induce damages. These models could also be enriched with experiments like impact tests.

The last main design activity refers to the torque transmission capability of the shaft. This analysis concerns three distinct area of the shaft: the tube section, the connecting radius and the flange. Each area has different geometry and/or stacking sequence and therefore has different properties and strength. Composite failures can appear in various ways since it can be because of delamination, fiber or matrix rupture in traction, compression or shear. Calculation of stress and strain indicate a level of breaking of the shaft based on input torque load, but identify the right breaking criterion is difficult, and our design method provide elements to check to check if a failure will appear or not. Another difficulty is added by the manufacturing process, which has an important impact on final properties and strength of the composite. The more gathered data about manufacturing, materials and structure of the shaft, the more precise the numerical simulations will be.

The core of the preliminary design activity is the capacity to enrich simulations with knowledge and data capitalized from previous design and experiments activities. This enrichment can take different forms, depending on the objective of the activity. A first example is about coupling modeling. A coupling can be done in 3D FEM, but it takes long time to model, to compute and also to change or adjust. A better approach is by using this 3D model once to compute main characteristics and properties (mass, damping, stiffnesses), and then reusing an equivalent model (superelement, simpler 1D element, reduced model). This also works with experimental data from couplings tested to measure their stiffnesses that can be reintroduce into a representative-but-simple model. A second example of this knowledge base is the capitalization of all main data about manufacturing, modeling, materials, experiments and non-destructive testing. It is also enriched by the test pyramid results, to refine mechanical properties and strength. Last but not least, dynamic analysis requires a certain amount of experience to have a good understanding of modeling and results. Results interpretation is improved by the knowledge base helping to identify types of modes, instabilities or vulnerable area sensitive to failure.

5. IMPLEMENTATION ON A TEST CASE

5.1. Case study specifications

The development process just described will be illustrated through an example of composite transmission design. Specifications and transmission design is not representative of the STEADIEST final design and only aims to illustrate the current method of driveline development. An assumption can be made regarding support and coupling definitions. Since this work doesn't intent to provide design method for both coupling and support, it can be assumed that these are reused parts given by the project leader. Another assumption is made regarding specifications completeness before any architecture design exploration. Based on this context, the table below provide specifications to guide design composite driveshafts of the transmission driveline.

Table 1: Aircraft transmission driveline specifications

Characteristic	Metric
Glass transition temperature	> 105 °C
Length of the transmission	9 686 mm
Angular misalignment	4°
Limit torque	1 900 N.m
Ultimate torque	2 850 N.m
Operational frequency	85 Hz
First avoidance band	0 - 26 Hz
Second avoidance band	60 - 108 Hz
Third avoidance band	190 - 250 Hz
Tube maximal diameter	65 mm

A first architecture that could be investigated is illustrated by the concept B of the following figure. This architecture is made up of 4 identical composite driveshafts in order to optimize manufacturing costs and to rationalize the architecture.

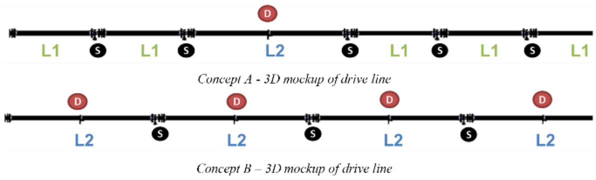


Figure 5: Two examples of transmission architecture concept

Following subsections will detail the two first phases of composite driveline development.

5.2. Test case on the first phase

The very first activity of the proposed design method, after getting specifications, is related to component properties in order to build a 1D FEM model of the whole driveline and to perform NDOE. Based on the specifications, it can be assumed that CAD files of the couplings and the support are available from which global properties of these components will be extracted by local analysis. Overall dimensions, mass and inertial properties could be extracted from either CAD file or 3D FEM model, while application of nominal forces and moments will enable stiffnesses characterization, whose an extensive overview is illustrated below.

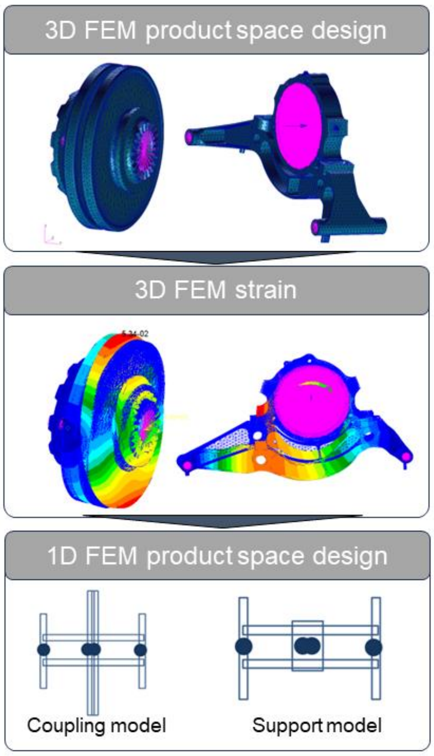


Figure 6: 3D to 1D model reduction

Based on this approach, the following tables summarize supports and couplings properties that will be used in 1D FEM models of the components and the driveline.

Table 2: Coupling properties

Property	English	Metric
Length	0.27 ft	81 mm
Length of rotation	0.13 ft	40.5 mm
Outer diameter	0.197 ft	60 mm
Thickness	0.008 ft	2.5 mm
Mass	3.31 lb	1.5 kg
Axial stiffness	5710 lb/in	1000 N/mm
Radial stiffness	3.14e+5 lb/in	55000 N/mm
Torsion stiffness	4.43e+6 lbf.in/rad	5.0e+8 N.mm/rad
Bending stiffness	2.66e+4 lbf.in/rad	3.0e+6 N.mm/rad
Material	Titanium	

Table 3: Support properties

Property	English	Metric
Length	0.295 ft	90 mm
Length of rotation	0.18 ft	55 mm
Outer diameter	0.197 ft	60 mm
Thickness	0.016 ft	5 mm
Axial stiffness	28551 lb/in	5000 N/mm
Radial stiffness	25696 lb/in	4500 N/mm
Torsion stiffness	0 lbf.in/rad	0 N.mm/rad
Bending stiffness	8.85 lbf.in/rad	1000 N.mm/rad
Material	Steel	

The last assumption regarding the transmission driveline definition is about shafts dimensions. Based on driveline and components lengths, architecture concept B from figure 5 induces shaft length of 2273 mm. Starting with maximum allowed diameter for the tubular section facilitates torque transmission. At last, a 3 mm tube thickness is set by knowledge capitalization, as it is needed regarding torque level and shaft length. Outer diameter and thickness of tube section will remain as assumption during this first phase of design, enabling composite and wrap angle identification and will be refined during the second phase. Once this setup preliminary phase is done, NDOE is run and gives the following results:

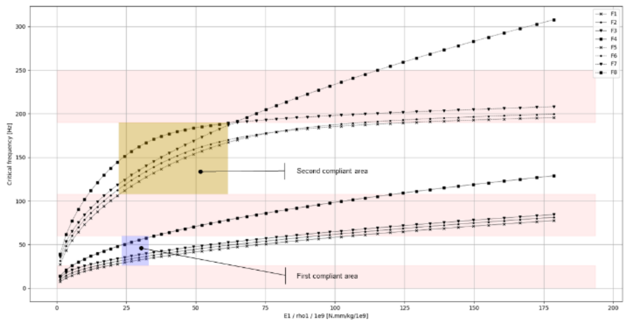


Figure 7: NDOE results for concept B architecture

Avoidance bands for critical frequencies are highlighted in red and designers need to identify areas between these avoidance bands where all driveline frequencies are compliant. As it can be seen on the above graph, there are two compliant areas: the first one is between 24 and 33 N.mm/kg/1e+9 (in blue) and the second one between 22.5 and 61.5 N.mm/kg/1e+9 (in brown). Only the intersection of these two (or multiple in general case) possible ranges gives the final available specific modulus range, which is between 24 and 33 N.mm/kg/1e+9 in this case, for this architecture. Once specific modulus range is identified, identification of an appropriate composite is done with the following graph:

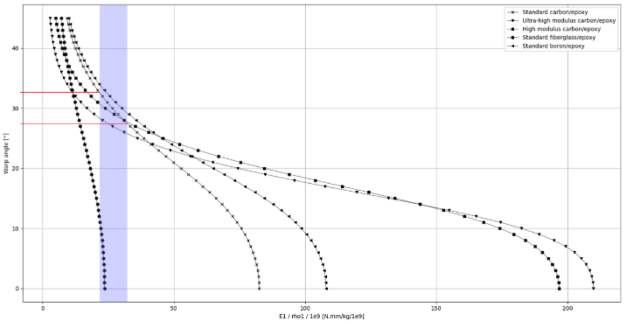


Figure 8: Composite wrap angle depending on specific modulus

This graph provides an extensive display of composites known in company's database, and allow to quickly identify what material suits specifications. For this architecture concept B, specific modulus range of interest is reported in figure 8 in blue. It can be observed that all composites cross the area, but not with the same wrap angle. For example, fiberglass/epoxy would be compliant with specific modulus if its stacking sequence is +/- 12° or less, which will lead to a very bad torsional strength so a poor torque transmissibility. However, a tube with a standard carbon/epoxy composite would be compliant if its wrap angle is between 27° and 32°, and will have a much higher torque transmissibility thanks to its higher strengths and wrap angles than a fiberglass/epoxy composite. Having such a length for a driveshaft lead to supercritical configuration since no material has high enough Young's modulus to keep very long driveshaft subcritical: operating range is between 60 and 108 Hz according to table 1, thus driveshafts will be automatically supercritical based on NDOE results for concept B. This induces the presence of dampers and friction bushes for every shaft to help them accelerate through their first bending critical speed. It can be imagined that friction bush mass is approximately 1.5 kg and is placed in the middle of the shaft. A new NDOE is run and produces the following results:

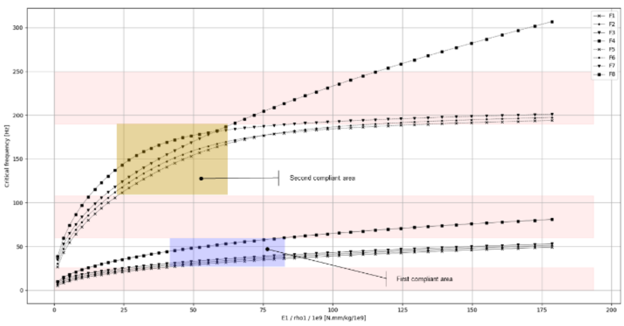


Figure 9: NDOE results for concept B architecture with friction bushes

While area overhead the operating range has not been significantly impacted since the additional mass of the shaft is positioned on a node of the second bending mode, but it has a good impact on the first frequency by flattening the curves. Thanks to that, a wider specific modulus range between 41 and 62 N.mm/kg/1e+9 can be considered to search for a composite material. Standard composite carbon/epoxy would be suitable with a wrap angle between 18° and 25°.

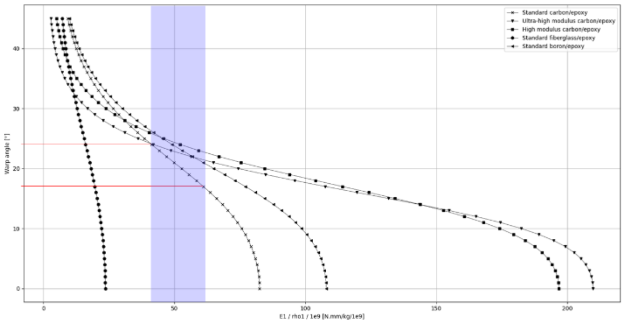


Figure 10: Composite wrap angle depending on specific modulus with friction bushes

Based on these two NDOE, a correct wrap angle range to considerer for the shaft would be between 25° and 30° depending on the mass of the friction bushes.

A cost and weight analysis for this case study is difficult to assess since there is no defined objectives. It will be assumed that the business case has been studied and an architecture such as presented in concept B would be relevant regarding both costs and weight goals. This composite driveshaft would be further analyzed and optimized in following development phase: preliminary design.

6. VALIDATION OF THE DESIGN METHOD

Technology readiness levels (TRLs) are the classical method for estimating the maturity of technologies during the acquisition phase of a program. For STEADIEST program, the technology concept has been formulated (TRL2), experimental proof of concept (TRL3) has been carried out on test shafts to validate:

- Torsional stiffness before and after impact,
- Torsional ultimate load after impact,
- Ability to exceed 10^6 cycles during torsional fatigue tests.

Then component validation in laboratory environment has been carried out on dynamic tests (TRL4) for shafts equipped with couplings and bearing supports.

The purpose of this paragraph is to briefly present the specimens under test, the means of the tests and the main results.

Two shaft configurations were tested (not corresponding to the solutions proposed in the previous chapter for reasons of planning and availability of specimens). A short subcritical shaft and a long supercritical shaft were selected whose properties are summarized in the following tables.

Table 4: Short shaft properties

Property	English	Metric
Length	3.789 ft	1155 mm
Outer diameter	0.213 ft	65 mm
Thickness	0.011 ft	3.4 mm
Mass	2.955 lb	1.34 kg
Winding angle	35.7 °	
Number of plies	16	
Material	AS4-8552	

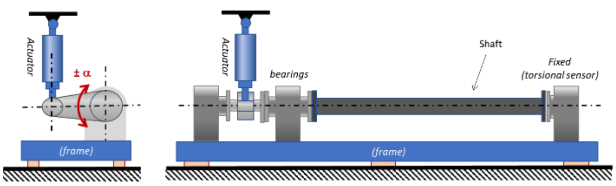
Table 5: Long shaft properties

Property	English	Metric
Length	7.457 ft	2273 mm
Outer diameter	0.213 ft	65 mm
Thickness	0.011 ft	3.3 mm
Mass	6.35 lb	2.88 kg
Winding angle	27.7 °	
Number of plies	16	
Material	AS4-8552	

6.1. Static and fatigue tests

6.1.1. Test bench

The LISPEN laboratory and NEXTEAM have developed a test bench for static and fatigue tests of a transmission shaft (tubular part with interfaces). During these tests, the objective was to test only the composite tubular part, the interfaces are not representative of the solutions adopted by the topic manager.



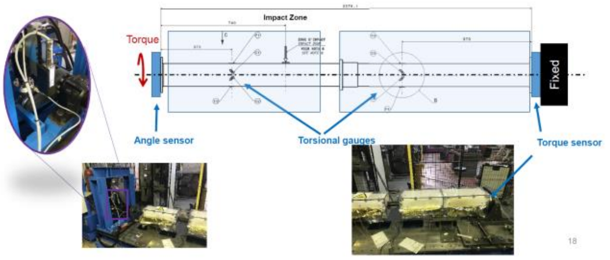


Figure 11: Static and fatigue bench

The tests were carried out at a temperature of approximately 70°C on several identical specimens (short shafts and long shafts).

6.1.2. Main results

It was shown through these static tests that the stiffness of the tubular part remained the same before and after impact. Within a few percent the measured stiffness is close to the stiffness calculated by FEM and little variation from one specimen to another.

Tests on short and long shafts after impact on Ultimate Load (UL) during 3s showed the ability to exceed the 2807 N.m required as provided by the specifications and correlate with the finite element calculations.

The fatigue tests showed that the configuration tested on the long shafts and the short shafts made it possible to exceed the 10^6 cycles at low frequencies (<5Hz) required by the specifications.

6.2. Dynamic tests

The dynamic tests were carried out according to two methods: impulse tests by impact hammer or rotation tests without torque.

The following dynamic tests were carried out:

- Impulse tests in free-free conditions for long shafts and short shafts,
- Long shaft impulse tests mounted on the ENSAM dynamic bench with STEADIEST bearing,
- Run-up tests for modes detection between 0 and 120 Hz, with or without STEADIEST bearing.

6.2.1. Hammer tests

Figure 12: Impact hammer test bench (free-free)

These tests allow designers to correlate FEM results with real properties since geometry is easily measurable and boundary conditions are perfect. Tests have presented a good repeatability between each component, and a good correlation has been achieved between FEM models and experimental results.

Table 6: Experimental results free-free hammer test

Mode	FEM frequency	Exp. frequency
Short shaft 1st mode	217 Hz	221 Hz
Short shaft 2nd mode	613 Hz	687 Hz
Long shaft 1st mode	71 Hz	76 Hz
Long shaft 2nd mode	214 Hz	219 Hz

Once shaft (alone) properties are validated, full bench configuration can be tested with hammer tests. These tests will deliver more information about connections between parts, stiffnesses of components and real boundary conditions. Such tests have been performed on the long shaft with the following configurations.

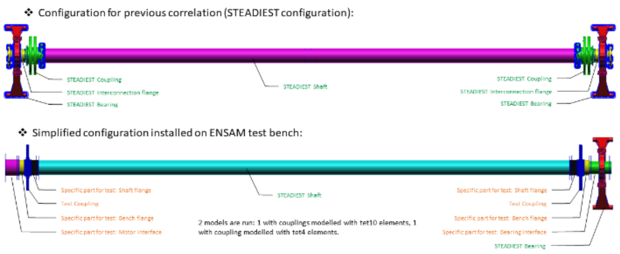


Figure 13: STEADIEST (up) and LISPEN (down) bench configuration

FEM results for the LISPEN configuration, further compared to hammer test results, are presented in figure 14.

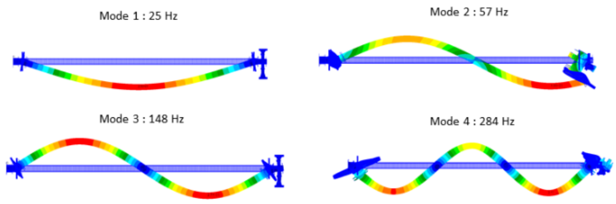


Figure 14: Normal modes FEM results for LISPEN configuration

It can be observed that mode 1, 3 and 4 are ideal beam bending modes while the 2nd mode is a mix of coupling bending and shaft bending, resulting in an intermediate frequency compared to analytical theory for beam bending critical frequencies. The validation of this intermediate mode will be completed with hammer test on bench structure with supports and couplings. An example of assembly is presented below, on figure 15.

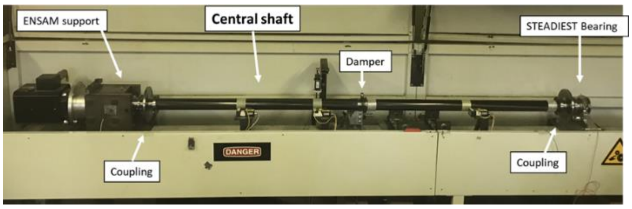


Figure 15: Hammer test - long shaft with LISPEN (ENSAM) & STEADIEST bearings

And an extract of hammer test results for this configuration is given on figure 16.

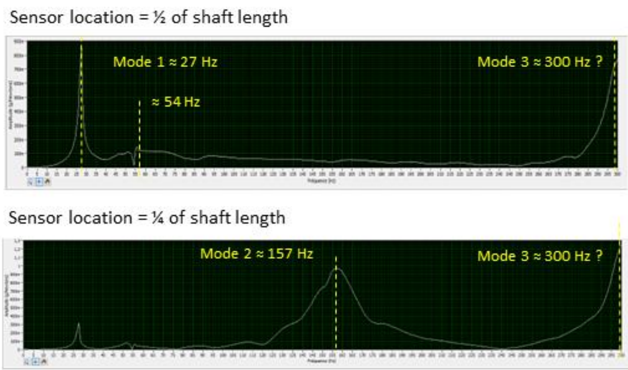


Figure 16: example of Impact Hammer test – LISPEN & STEADIEST bearings

This experiment confirms FEM results since all bending critical frequencies are present, even the intermediate one around 55 Hz. In addition, its amplitude is lower thus maybe it can express a less energetic behavior, that can be validated by a dynamic test in the following section.

6.2.2.Run-up tests

The purpose of these run-up tests is to detect shaft resonance modes associated with couplings and bearings excited by residual unbalance (1Ω) or residual misalignment (2Ω) and to detect instabilities induced by internal damping.

The sensors and measuring instruments are: Rotational speed, vibrations in the supports (4 accelerometers), beam deflection (4 laser proximity sensors).

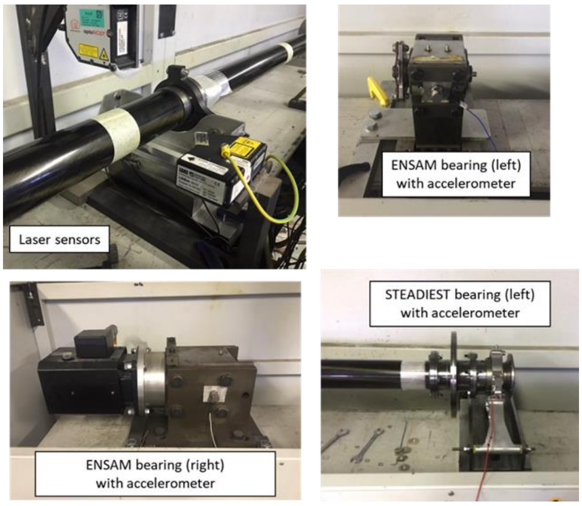


Figure 17: Instrumentation on LISPEN (ENSAM) bench

Ramp-up tests from 0 to 120 Hz (bench limit) were carried out to detect the modes for the configuration with two LISPEN bearings.

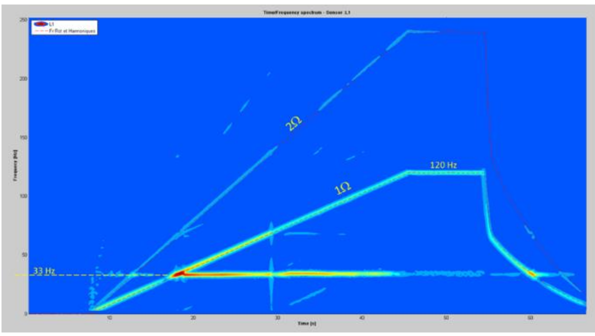


Figure 18: Ramp-up tests from 0 to 120 Hz – long shaft – LISPEN bearing

This dynamic test shows that the first mode is around 30 Hz (as previous by calculation), no other mode is below 120 Hz. No instability was observed below 120 Hz. This test confirms the weak energy associated with the 2nd bending mode observed by FEM.

The tests of short shafts showed similar good correlation on the prediction of frequencies and the absence of instability.

7. CONCLUSION AND OUTLOOK

After a literature review which highlighted the lack of global driveline and driveshaft development method, we propose a new process to bridge theory between product development process, NDOE and knowledge capitalization.

In the first part, the link between design theory and NDOE is fully analyzed in order to quickly identify composite and winding angle compliant with avoidance bands from specifications. This has been demonstrated on a fictive driveline specifications and results are promising thanks to the time saving in solution space exploration.

Then, a real case study has been presented based on work from STEADIEST project focused on critical frequencies validation. These shafts were designed and built based on the second part of the proposed design process. Other works related to torsional strength and stiffness, fatigue and environmental tests are not covered in detail. Critical frequencies correlation between FEM and experiments has been very good, even with unusual frequency like observed with the 2nd mode. Even if this frequency was also observed with hammer test, dynamic run confirmed the absence of excessive vibration amplitude or instability around its frequency area.

In conclusion, this design method has been proven very efficient to explore design solution space and to be very flexible and extensible to any general design problem – including driveline design. However, progress need to be made regarding knowledge capitalization and reuse to improve bridges between manufacturing and design processes in order to better predict final properties and behavior of the system.

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