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GEOMETRIC OPTIMISATION OF HINGE-LESS DEPLOYMENT SYSTEM FOR AN ACTIVE ROTORBLADE

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ABSTRACT

The Green Rotorcraft project (part of Clean Sky JTI) is studying the Gurney flap as a demonstrator of a smart adaptive rotorblade. Deployment systems for the Gurney flap need to sustain large centrifugal loads and vibrations while maintaining precisely the displacement under aerodynamic loading. Designing such a mechanism relies on both the actuation technology and the link which transmits motion to the control surface. Flexible beams and piezoelectric patch actuators have been chosen as components to design this mechanism. Flexible beams are providing an hinge-less robust structure onto which the piezoelectric actuators are bonded. A candidate topology is determined by investigating the compliance of a simple wire structure with beam elements. A parametrized finite element model is then built and optimized for displacement and force through surrogate optimization. The whole process does not requires many finite element analyses and quickly converge to an optimized mechanism.

INTRODUCTION

Adapting a rotorblade in-flight is the next step towards smarter, more efficient rotorcrafts. The Green Rotorcraft Consortium, part of European Clean Sky Joint Technology Initiative has chosen the Gurney flap as an active concept to modify the aerodynamic characteristic of a rotorblade during flight [1, 2]. The Gurney flap consists in a small flap deployed as close as possible to the trailing edge of the blade profile as shown in Fig. 1. This flap typically measures 2% of the profile's chord length. It improves the lift of the profile without significant increase in the

profile's drag [3–5]. This paper presents the optimisation work done on a flexible mechanism to provide sufficient displacement while satisfying mechanical and deployment time constraints. A procedure is setup to investigate suitable geometries and optimize the geometry of the actuated system.

FIGURE 1. SKETCH OF THE NACA 23012 PROFILE WITH A 2% LONG GURNEY FLAP AT THE TRAILING EDGE.

The active Gurney flap concept

A fully deployed Gurney flap increases the lift coefficient of the profile over a wide range of angles of attacks. It also improves both the static and dynamic stall behaviour of the profile [6]. Helicopter's performances are limited by the lift mismatch between the blade on the advancing side and the blade on the retreating side of the rotorcraft during forward flight as shown in Figure 2. Thus, actively increasing the lift for the retreating blade offers potential for improving speed and fuel-efficiency. The study carried out by the Green Rotorcraft Consortium's partners concluded the Gurney flap should be deployed within 10 degrees of sweeping angle for optimum performances [1]. The blade spec-

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ifications shown in Table 1 were defined in the project baseline. They provide the data needed to derive mechanical constraints for a deployment mechanism [7]. The speed required by the deployment mechanism is derived from the blade rotation speed. A flexible mechanism will take care of the centrifugal loads along the blade axis. An upper bound for the holding force applied on the deployed has been determined using flow simulations [8]. Those constraints are summarized in Table 2.

FIGURE 2. DURING FORWARD FLIGHT, THE AIRSPEED ON THE ADVANCING BLADE IS LARGER THAN THE AIRSPEED ON THE RETREATING BLADE.

TABLE 1. HELICOPTER BLADE SPECIFICATIONS.

Profile reference	Naca23012
Blade radius	8.15 m
Chord length	0.65 m
Rotation speed	26.26 rad/s
Tip speed	214 m/s

Actuators suitable in rotorcrafts

Pneumatic actuators, coil actuators and screw jack electrical motors are unsuitable for actuation, due to the space available and the rotation speed. Therefore an extensive review at the

TABLE 2. DESIGN CONSTRAINTS

Deployment speed	6.6 ms
Axial acceleration	573 g
Holding force	250 N

capabilities of piezoelectric based actuators was carried out in the scope of this project. Piezoelectric material's properties are highly anisotropic. The piezoelectric matrix, shown in Eqn. 1 relates applied electrical fields (E) to the strain state of the material.

$$\begin{pmatrix} \epsilon_1 \\ \epsilon_2 \\ \epsilon_3 \\ \gamma_{23} \\ \gamma_{31} \\ \gamma_{12} \end{pmatrix} = \begin{bmatrix} 0 & 0 & d_{31} \\ 0 & 0 & d_{32} \\ 0 & 0 & d_{33} \\ 0 & d_{25} & 0 \\ d_{15} & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} \times \begin{pmatrix} E_1 \\ E_2 \\ E_3 \end{pmatrix} \quad (1)$$

The material performances depends on the value of the strain coefficients ($[d]$) filling the piezoelectric matrix. The d_{33} coefficient is generally the highest coefficient [9]. Therefore, piezoelectric actuators taking advantage of the d_{33} coefficient achieve the best performance. Macro Fibre Composite (MFC) actuators consist in piezoelectric fibres embedded in an epoxy matrix as shown in Fig. 3. Electrical fields are applied along the piezoelectric fibres, in line with the d_{33} coefficient, through interdigitated electrodes. During the actuator selection process d_{33} piezoelectric actuators came out ahead [1, 8]. Their actuation speed met the speed requirement according to the blade rotation speed and their toughness is sufficient to withstand high centrifugal loads. MFCs are used in this study as the piezoelectric active material. Nevertheless the displacement generated by piezoelectric actuators is very small. A supporting mechanism has to be conceived to support the MFC actuator and enhance its displacement.

Mechanisms made with bending beams

To provide a sufficiently large displacement, piezoelectric actuators are often bonded onto bending beams. When a voltage is applied, the piezoelectric patch generates strain at the beam surface which bends the beam. The tip displacement of the bending system is much larger than the displacement generated by the patch actuators. This article details the geometrical optimisation of flexible mechanisms consisting of multiple bending beams actuated by piezoelectric patches.

FIGURE 3. STRUCTURE OF A MACRO FIBRE COMPOSITE PIEZOELECTRIC ACTUATOR.

METHODS

Early designs of a mechanism made for the Gurney flap consist in directly deploying the Gurney flap transverse to the flow direction [5, 10]. The efficiency of the Gurney flap as a lift improvement control surface, depends strongly on the Gurney flap position relative to the trailing edge. It has been shown that the Gurney flap efficiency is maximized when placed at the trailing edge [10]. However, the space available inside a beam profile is very limited close to the trailing edge. Therefore, a horizontal motion is needed on top of a vertical motion to deploy and place the Gurney flap at the trailing edge. It is possible to kinematically link the vertical motion to the horizontal displacement by a sliding mechanism. Therefore a single horizontal motion could deploy and bring the Gurney flap at the right position. Moreover, a sliding mechanism allows a cross flow deployment which requires less force than deploying the Gurney flap by rotation like a typical control surface.

The process followed to determine a topology and optimize a piezoelectric actuated system is detailed in the flowchart shown in Fig 4. Each step is briefly discussed except third block, discussed in detail.

Topology investigation

Possible topologies for the mechanism structures are investigated using a simplified wired structure with a limited number of nodes. A procedure tests the various possible connections between a defined square array of nodes as shown in Fig. 5. A few constraints govern the connections between the nodes: the two fixed nodes and the lower right node must remain connected to the structure. Once the connections are set, a linear system of equations is solved to calculate the deformation of the structure when a horizontal force is applied on the lower right node. The structures are ranked by the amount of horizontal displacement they could offer. This preliminary test leads to the structure displayed in Fig. 5.

FIGURE 4. FLOWCHART DETAILING THE PROCESS FOLLOWED TO DETERMINE A TOPOLOGY AND OPTIMIZE A PIEZOELECTRIC ACTUATED SYSTEM.

FIGURE 5. RESULTING STRUCTURE AFTER INVESTIGATING THE FLEXIBILITY OF VARIOUS CONFIGURATIONS.

Finite Element Model

From the resulting topology, a Finite Element Model is setup using the same set of connections as shown in Fig. 6. The finite element simulation is a quasi-static bi-dimensional simulation. The structure geometry is generated based on three parameters: the length of the upper arm, the length of the bottom arm and the curvature of the middle arm. The piezoelectric element's dimensions depend on the length of the supporting arm. The material's properties are tuned to match the MFC performances from the

manufacturer datasheets [11]. The left-side of the upper arm is clamped and the lower arm is subjected to a sliding boundary condition. The geometry of both arms follows the curvature of the profile to use as much available space as possible. The finite element model is embedded inside a procedure that returns the objective value that is going to be maximized by the optimization loop. This objective value is calculated according to the required constraints for the mechanism. The objective value is expressed as:

$$y = \sqrt{\left(\frac{d}{d_{pzt}}\right)^2 + \left(\frac{f}{f_{pzt}}\right)^2} \quad (2)$$

Where y is the objective value, d is the free displacement of the lower arm, d_{pzt} is the free displacement of the MFC actuator without structure, f is the block force of the mechanism and f_{pzt} is the block force of the MFC actuator without structure. Maximizing this objective leads to mechanisms with both large forces and displacements.

FIGURE 6. GEOMETRICAL PARAMETERS USED FOR THE FINITE ELEMENT ANALYSIS.

Optimisation scheme

The optimisation scheme chosen is a surrogate optimisation for its flexibility and the number of simulation it requires to approach the optimum. It consists in the determination of a surrogate function followed by a refinement of the optimum [12] as shown in Fig. 7. 30 finite element simulations are computed to explore the design space. A latin hypercube distribution is chosen to get a relevant distribution of the design parameters. An ordinary kriging model is chosen as a surrogate function to estimate the model [13]. An ordinary kriging model can be defined as a base function which represents the global trend of the data with a stochastic function that approximates the data computed at the sampling points:

$$\hat{y}(x) = \mu + \varepsilon(x) \quad (3)$$

$$x = \begin{pmatrix} a \\ b \\ c \end{pmatrix} \quad (4)$$

FIGURE 7. FLOWCHART DETAILING THE EVALUATION OF THE SURROGATE FUNCTION AND THE OPTIMIZATION LOOP.

where $\hat{y}(x)$ is the estimation of the surrogate function for a vector of parameters x , μ is a constant value corresponding to the base function of the ordinary kriging model and $\varepsilon(x)$ is the function which estimates data modeled at the sampling points. The surrogate function evaluates the objective value for a set of design parameters. After the surrogate function is calculated (cf. Appendix A), Matlab standard genetic algorithm searches the design space for a global maximum. The maximum is refined by a gradient-based search. Once the extremum is found, a new finite element analysis evaluates the displacement at the extremum. The loops stop when the objective set no longer improves significantly between two iterations. For this optimization scheme

the termination objective was less than 0.05 % of improvement between two iterations.

Performance evaluation

Force and displacement are not the only requirements. As mentioned in the introduction, the deployment speed is critical for a correct performance of the Gurney flap. The force and displacement capabilities of the mechanism are evaluated during the finite element analysis to calculate the objective value. However, a transient analysis is needed to obtain the displacement speed. It is performed on the optimized geometry.

RESULTS

The results of the geometrical optimization are presented in Fig. 8. After the 30 initial Finite Element simulations and the evaluation of the surrogate model, the problem converged quickly. The termination criteria is achieved within 9 iteration loops. The optimum geometry displayed in Fig. 9 shows a middle arm with an inverted slope when compared to the initial structure shown in Fig. 5.

FIGURE 8. VALUES OF THE GEOMETRICAL PARAMETERS USED FOR THE OPTIMIZATION AND THE OBJECTIVE.

The bender mechanism achieves a free displacement of 2.3 mm for a block force of 230 N. The deformed structure is shown in Fig. 9. The displacement remains insufficient for sliding directly the length of the Gurney flap (13 mm). However, the block force is sufficient to sustain the force of the airflow at the tip of the blade [8].

FIGURE 9. DEFORMATION FIELD OF THE OPTIMIZED MECHANISM

This optimization objective was focused on getting a structure that would maximize both force and displacement. Other requirements such as deployment speed are needed to have a suitable actuation mechanism. The deployment speed was investigated using a transient analysis of the finite element analysis set. The displacement of the bender mechanism subjected to a short step voltage (0.1 ms) is displayed in Fig. 10. 10 degrees of sweeping angle are completed in 6.6 ms. The displacement shown at this time step is 1.13 mm. The free displacement is achieved at 11 ms and increases further on due to the inertia of the motion. The displacement oscillations are undamped because no structural damping has been implemented in the finite element model.

CONCLUSION AND FUTURE WORK

The process described in this article successfully achieves to optimise an actuation mechanism according to both displacement and force. It manages to amplify the small strain generated in a MFC piezoelectric actuator into a significant displacement. The performance of the resulting geometry showed sufficient block force. However, the displacement and the deployment speed are still not sufficient for this application. The transient analy-

FIGURE 10. DISPLACEMENT OF THE BENDER MECHANISM DURING THE TRANSIENT ANALYSIS.

sis shows that the displacement of the structure is higher than the free displacement using the inertia of the motion. Future work involves redefining the objective's calculation to include and take advantage of the dynamic effects and implementing a more complex finite element model that takes into account damping, contact elements and has more MFCs. The described procedure can be applied to any actuated system requiring an optimisation based on performances.

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Appendix A: Calculation of the surrogate function with an ordinary kriging model

An ordinary kriging model can be defined as a base function which represent the global trend of the data with a stochastic function that approach the data computed at the sampling points.

A kriging model can be defined as:

$$\hat{y}(x) = \mu + \varepsilon(x) \quad (5)$$

Where $\hat{y}(x)$ is the estimation of the surrogate function for a set of parameters x , μ is a constant value corresponding to the base function of the ordinary kriging model and $\varepsilon(x)$ is the function which estimates data modeled at the sampling points. The surrogate function will evaluate the horizontal displacement of the bottom arm for a set of design parameters.

A gaussian correlation is used for this model:

$$R(x_i, x_j) = e^{-\theta(x_i - x_j)^2} \quad (6)$$

Where x_i and x_j are parameters values, θ is a scaling parameter for the correlation function, which value depends on the problem considered.

Then the base function is estimated:

$$\hat{\mu} = \frac{X^T \hat{R}^{-1} y}{X^T \hat{R}^{-1} X} \quad (7)$$

Where $\hat{\mu}$ is the estimation of the base function, $\hat{R}$ is the correlation matrix and X is a unit vector (ordinary kriging case).

Therefore the prediction of the kriging model is:

$$\hat{y} = \hat{\mu} + r^T \hat{R}^{-1} (y - \mathbb{I} \hat{\mu}) \quad (8)$$

Where $\hat{y}$ is the estimation of y , r is the correlation vector between the set of parameters y is evaluated at and the parameters at which the surrogate function has been evaluated and $\mathbb{I}$ is a unit vector.