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Structural Analysis and Study of Failure Modes of Ornithopter

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Abstract—*Some of the failure modes that occur in the ornithopter are said to be studied. These limiting failure modes that are encountered in the wing and primary structure are namely Euler Buckling, composite laminate failure, Torsional buckling failure are analyzed. A study comprising the set of formulas which when implemented in this analysis characterizes the section properties of each structural member. This study also includes a process through which the evaluation of the stress and proximity of failure of material which under combined bending, axial and torsional loads give the framework through which the failure analysis of ornithopter has been carried out.*

Keyword : *Buckling, failure analysis*

1. INTRODUCTION

Flapping wing propulsion integrates into two separate systems namely aerodynamic lift generation and propulsion. Inconvenience, both these conventional design are separate and gives lesser degree of interdependence. A flapping wing aircraft produces thrust by allowing the wing to twist and utilizes several strategies to ensure time average normal force vector of wing is the thrust rather than drag. This complex relationship makes an ornithopter an ideal application for an aero structural optimization scheme.

If there is hinging of the wing at one centre point or at multiple points (engine powered ornithopter),Human powered ornithopter (HPO) wing achieves its flapping deflection entirely by the elastic bending of the main spar.

Primary structure design consideration

In case of HPO the design was similar to many aircraft given either the stiffness or failure constraints for each degree of freedom or structural member. Some of the driving design considerations in the case of the HPO are

- Ensure optimal torsional compliance for wing to generate sufficient thrust .
- Limit bending deflection in the chord wise (in plane) direction of wing to control flapping kinematics.
- Limit bending deflection in the out of plane direction of the wing to maintain thrust efficiency.
- Design of optimal flapping kinematics while mitigating relevant failure modes.

Primary structure and its fabrication

The wing design and its structure are the major concern for the structural analysis . The primary structure fabrication was used to provide optimal design flexibility. The wings spar has been fabricated using carbon prepeg fibre. In most of the cases composite laminates also used.

Determination of Material Properties

The determination of composite laminate material properties is very different from most materials for several reasons, including, but not limited to:

- The materials are non-isotropic, that is their properties are not the same in all directions due to their fibrous composition;
- The material properties are highly dependent on the rotational orientation of the laminate plies;
- The strengths and stiffness of composites often differ in compression and tension, even in the same axis, due to dramatically different failure modes and mechanical forces in action;
- The material properties depend highly on the conditions under which the layup is cured.

2. CLASSIFICATION OF ENGINEERING PROBLEMS

According to S. H. Krandall (1956), engineering problems can be classified into three categories:

- equilibrium problems
- eigen value problems
- propagation problems

Equilibrium problems are characterized by the structural or mechanical deformations due to quasi-static or repetitive loadings. In other words, in structural and mechanical systems the solution of equilibrium problems is a stress or deformation state under a given load. The modeling and analysis tasks are done to obtain the system stiffness or flexibility so that the stresses or displacements are computed accurately.

Eigen value problems can be considered as extensions of equilibrium problems in the solutions that are dictated by the same equilibrium states. There is an additional distinct feature in Eigen value problems: their solutions are characterized by a unique set of system configurations such as resonance and buckling.

Propagation problems are said to predict the subsequent stresses or deformation states of a system under the time-varying loading and deformation states. It is called initial value problems in mathematics or disturbance transmissions in wave propagation.

Model testing is perhaps the most widely accepted words for activities involving the characterization of mechanical and structural vibrations through testing and measurements. It is primarily concerned with the determination of mode shapes (Eigen vectors) and modes (Eigen values), and to the extent possible the damping ratios of a vibrating system. Therefore, model testing can be viewed as experimental solutions of Eigen value problems.

There is one important distinction between Eigen value analysis and model testing. Eigen value analysis is to obtain the Eigen values and eigenvectors from the analytically constructed governing equations or from a given set of mass and stiffness properties. There is no disturbance or excitation in the problem description. On the other hand, model testing is to seek after the same Eigen values and eigenvectors by injecting disturbances into the system and by measuring the system response. However, model testing in earlier days tried to measure the so-called free-decay responses to mimic the steady-state responses of equilibrium problems.

Table 1: Comparison of Engineering Analysis and System Identification

	Engineering Analysis	**System Identification**
Equilibrium	Construct the model first, then obtain deformations under any given load.	Measure the dynamic input / output first then obtain the flexibility.
Eigen value	Construct the model first, then obtain eigen values without any specified load	Measure the dynamic input/output first, then obtain eigen values that corresponds to the specific excitation
Propagation	Construct the model first , then obtain responses for time – varying loads	Measure the dynamic input/output first, then obtain the model corresponds to the specific load

It is observed from the above Table that the models are first constructed in engineering analysis. In system identification the models are constructed only after the appropriate input and output are measured. Nevertheless, for both engineering analysis and system identification, modeling is a central activity. It is also observed that, in engineering analysis, once the model is constructed it can be used for all of the three problems. On the other hand, the models obtained by system identification are usually valid only under the specific set of input and output pairs. The extent to which a model obtained through system identification can be applicable to dynamic loading and transient response measurements depends greatly upon the input characteristics and the measurement setup and accuracy.

3. STRUCTURAL MODELING BY SYSTEM IDENTIFICATION

As noted in the previous section, modeling constitutes a key activity in engineering analysis. For example, the finite element method is a discrete structural modeling methodology. Structural system identification is thus a complementary endeavor to discrete modeling techniques. A comprehensive modeling of structural systems is shown in Fig. 1. The entire process of structural modeling is thus made of seven blocks and seven information transmission arrows (except the feedback loop).

Testing consists of the first two blocks, Structures and Signal Conditioning along with three actions, the application of disturbances as input to the structures, the collection of sensor output, and the processing of the sensor output via filtering for noise and aliasing treatment.

FFT (Fast Fourier Transform) and Wavelets Transforms are the software interfaces with the signal conditioners. From the view point of system identification, its primary role is to produce an accurately possible impulse response functions either in frequency domain or in time domain variables. It is perhaps the most important software task because all the subsequent system realizations and the determination of structural model parameters also depend on the extracted impulse response data.

System realization performs the following task:
For the model problem of

Plant : $\dot{x} = A x + B u$

Give measurements of

Input : u

Output : $y = Cx + Du$

Determine system characteristics : A, B C and D

Structural modeling block is to extract physical structural quantities from the system characteristics or realization parameters (A, B, C, D). This is because

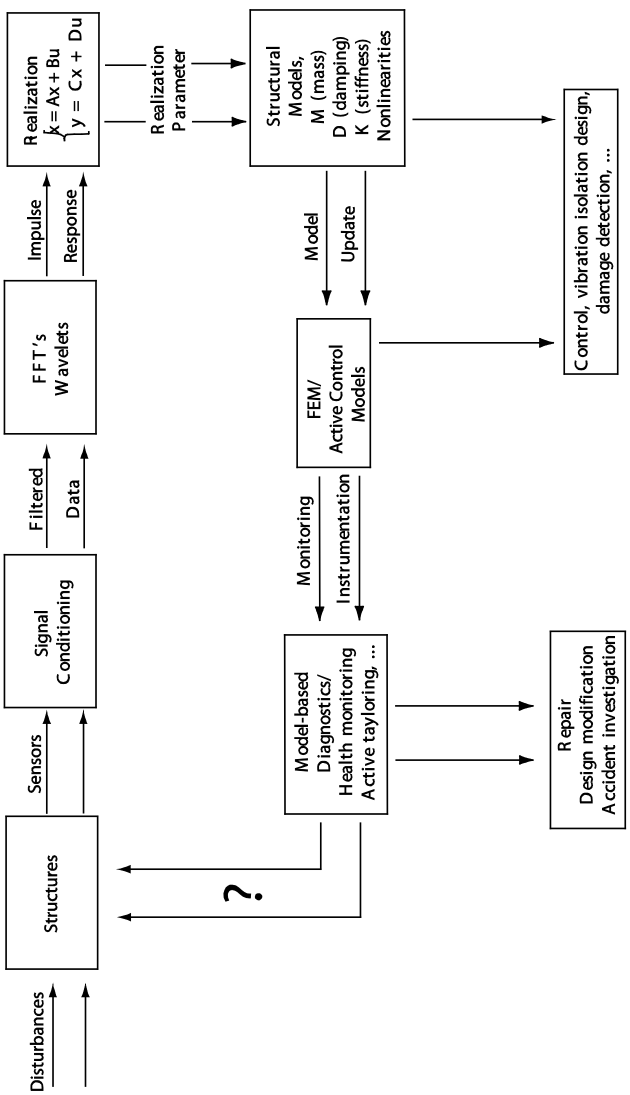

Fig. 1: System Identification

realization of characteristics still consist of abstract mathematical models, not necessarily in terms of the desired structural quantities, hence we obtain as follows,

Given realization parameters: A, B, C, and D

Determine either

1) Modal quantities: modes(ω) and mode shapes (φ)

2) Physical matrices: mass (M), stiffness(K) and damping (D)

Finite element model updating, active controls and health monitoring are the beneficiaries of the preceding four activities.

4. FAILURE ANALYSIS

From the analysis and literature review of the previous work done some of the procedure that where followed during failure analysis are

- Design and methodology for failure
- Analysis of frame work and failure prediction tools
- Testing and validation

Limiting modes

- The three limiting failure modes encountered in the wing primary structure are
- Composite laminate failure – this prevails in most or all of the carbon prepeg tube structure (Analogous limiting case to material failure in isotropic materials).
- Euler bucking failure – This is said to prevail in long slunder sections without a high EI, primarily the ¾" rear spars.
- Torsional buckling failure – This is said to prevail in the main spar, which has the potential to undergo high torque.

Laminate Failure Analysis

Failure Theories for Composite Lamina

Failure theories for composite lamina can be classified into three distinct groups as follows

- **Non-interactive or limit theories**: The failure modes are predicted by comparing individual stresses or strains respective to their ultimate stresses and strains. However, such theories do not account for interplay between different stress components. Examples of such theories are maximum stress criteria, and maximum strain criteria.
- **Interactive theories**: These theories are said to be a step further than limit theories, and also account for interaction between various stress/strain components. Examples of such theories are those of Tsai-Wu and Tsai-Hill. They are able to predict overall failure, but cannot predict the exact failure mode.
- **Failure mode based theories**: These theories provide separate criteria for failure of matrix, fiber and interface. Examples of such theories are those of Puck and Hashim-Rotem.

Capped Spar Sections

Composite laminate failure modes were addressed primarily because of the availability of proven analytic methods. Professor J. Hansen covered laminate failure analysis extensively in his advanced composites at UTIAS, and the course slides proved a valuable resource for adapting and implementing such a scheme for the HPO. The detailing of this analysis will begin with capped spar sections, as the different (e.g. unidirectional and axial) layers of the laminate require a more complex analysis than for a purely axially-wrapped non-capped tube.

The tubular laminates used in the HPO fall within the assumptions of classical plate theory, because of their extremely-thin characteristic in one dimension compared with the other materials. Those lack of significant stresses which are perpendicular to the plane of the laminate, and the proximity of the in-plane strains to zero. Working from this theory, the constitutive relation for a laminate can be assembled in the form

$$\begin{bmatrix} N \\ M \end{bmatrix} = \begin{bmatrix} A & B \\ B & D \end{bmatrix} \begin{bmatrix} \varepsilon^\circ \\ \kappa \end{bmatrix}$$

where

- **N** is the force resultant vector through the laminate;
- **M** is the moment resultant vector through the laminate;
- **A** is the membrane "stretching" stiffness matrix;
- **B** is the membrane bending/stretching coupling matrix;
- **D** is the flexural rigidity matrix;
- **ε°** is the vector of the membrane strains;
- **κ** is the vector of the membrane curvatures.

In the situation where there are no flexural stresses on the laminate (which is also said to be the case prevailing, even though in the gross structure there may be flexure), the assumption can be made such that no bending or curvature are present in the laminate. Therefore, the situation simplifies in this form,

$$[N] = [A]\,[\varepsilon^\circ]$$

The form of the stress resultant and plate strain vectors are

$$N^T = \lfloor N_x \; N_y \; N_{xy} \rfloor$$

$$\varepsilon^{\circ T} = \lfloor \varepsilon^\circ_x \; \varepsilon^\circ_y \; \gamma^\circ_{xy} \rfloor$$

where x and y refer to the longitudinal and lateral dimensions of the laminate, and xy refers to shear. Each stress resultant component is obtained by taking the average of the stress over the thickness of the lamina. The membrane stiffness matrix "A" is determined for a laminate comprised of n uniform-thickness layers using the summation as follows below

$$A = \sum_{i=1}^{n} (\overline{Q})_i t_i$$

Where "$\overline{Q}$" is the transformed orthotropic elastic constants matrix

Capped Tube Sections

The laminate failure analysis for capped tube are as follows. First, "A" is assembled using the contributions from the cap and the tube laminate, taken at the point of maximum thickness in the cap (which is the point of maximum compressive stress due to bending, that being the point furthest away from the centroid). The structural stresses are then taken from the FEM, and combined such that the total maximum compressive force and tensile forces are found. As for a beam, the bending stress will act as a compression on one capped surface and a tension on the other capped surface. These are combined with any axial force to compute the aforementioned maximums in the spar. The force resultant vector "N" is then determined using the maximum stresses. The constitutive relation given for "A" and "N", is then inverted to find the plate strain vector, ε°, which is uniform through out the laminate. The "$\overline{Q}$" vector is then taken for each of the spar and cap laminates. The stress strain relation is as follows.

$$\sigma = \overline{Q}\varepsilon^\circ$$

Note that the unidirectional caps already have their material axis aligned with the longitudinal axis of the structure, and hence the computed laminate stresses can be directly compared against the material strengths. However, the stresses in the tube laminate are displaced off of the material axis by the wrap angle "θ" of the tube. The stresses thus need to be rotated using the transformation

$$\sigma_{material} = T\sigma_{structure}$$

where "T" is the transformation matrix given for computing the orthotropic elastic constants. Now, the tube material stresses can also be directly compared with their determined material strengths. It is important that the compressive and tensile stresses both shall be compared with their corresponding material strengths, as these are said to be different for composite materials. The failure proximity of the spar was then computed via the maximum stress conditions. The conditions is that in no direction the stress in the material will not exceed the failure stress in that axis given the imposed safety factor. The method of maximum stress conditions is in common use in industry. Thus it is possible to determine whether a laminate structure is sufficiently strong in each direction, and by what margin.

Non-capped Sections

The process for a non-capped section is identical to that of capped section, except for two simplifications.

1. The assembly of "A" does not include any unidirectional cap layers;
2. The stresses need to be only calculated for the rotated tube laminate.

Euler Buckling Analysis

This has been implemented in the case of rear spar. Euler buckling is induced when an axial compressive load on a member, that may not be sufficient to cause a material failure in that member. This causes an instability and forces the member to bend and deform in a premature collapse. The equation for determining the Euler buckling load "P" in a slender structural member is

$$P = \frac{\pi^2 E1}{kl^2}$$

where "l" is the unsupported length of the column and "k" is the effective length factor, a measure of the stabilizing nature of the structure's end supports. In the case of the rear spar, the unsupported length is taken as that between the ends of each in-plane truss section. As for the end conditions, because the joint of each rear spar section is only stabilized against out -of-plane motion by the frame, and not solidly against rotation, the end condition is treated as a pin (for which k = 1). In reality this is a conservative choice such that in actuality the spar is somewhat stabilized by the intervening ribs and the rear spar is somewhat restrained at the end of each frame by its inherent bending stiffness.

The rear spar was checked for the possibility of Euler bucking failure by computing the in-plane drag loads on the wing. During thrust, the rear spar is under tension and hence buckling is not a concern. The in-plane truss was modeled similarly to a beam structure, the compressive force in the rear spar was determined using the maximum compressive stress that was found for the wing structure in the in-plane direction. Given the load in the wing, the proximity to failure of the rear spar in Euler buckling could be easily determined.

Torsional Buckling Analysis

As mentioned, consultation with Juan Cruz, who was the structures specialist of M.I.T.'s Daedalus HPA project gave appreciation for the buckling failure modes to be encountered.

The torsional buckling exhibited in these tubular sections is again an instability problem, where the shear stresses in the thin tube wall cause a snap-through and collapse under load. The failure is analogous to the twisting of an aluminum pop can, when the thin sidewall buckles and allows the can to be crushed. This mode is most frequently encountered in thin shell structures. Rocket bodies are fabricated in this fashion. Therefore, to be able to predict their failure modes under torsion, axial, and bending induced thin-wall buckling, NASA has contracted several studies. Two of these were "NASA Contractor Report 912: Shell Analysis Manual", recommended by Professor Hansen and "NASA Space Vehicle Design Criteria (Structures) SP-8007: Buckling of Thin-Walled Circular Cylinders", recommended by Juan Cruz. SP-8007 was coded by Cruz for this purpose in the Daedalus project, and was intended for use with orthotropic materials as also used here; and Cruz noted that the results were very consistent for the composite carbon fibre tubes used in that project. SP-8007 gives two crucial formulas: first, a formula for the critical torque that will cause a thin-walled cylinder to buckle; and second, a formula that determines the applicability of the buckling analysis. The formula for the critical buckling torque is given as

$$\tau_{cr} = (BF)\bar{D}_y^{\frac{5}{8}} \left(\frac{\bar{E}_x \bar{E}_y - \bar{E}_{xy}^2}{\bar{E}_y} \right)^{\frac{3}{8}} \frac{R^{\frac{5}{4}}}{l^{\frac{1}{2}}}$$

Where,

- x - Longitudinal axis of the tube laminate
- y - Circumferential axis of the tube laminate
- xy - Shear axis
- BF is the "Buckling Factor", an empirically – adjusted coefficient;
- $\bar{D}_y = \frac{E_y t^3_{tube}}{12(1-v_{12}^2)}$;
- $\bar{E}_x = \frac{E_y * t_{tube}}{1-v_{12}^2}$;
- $\bar{E}_y = \frac{E_y * t_{tube}}{1-v_{12}^2}$;
- $\bar{E}_{xy} = \frac{\upsilon_{12} \frac{(Ex+Ey)}{2} t_{tube}}{1-v_{12}^2}$;
- L is the unsupported length of tube.

The shear flexibility coefficient of tube.

$$R = \frac{\pi^2 D}{\iota^2 D_q}$$

with D the wall flexural stiffness per unit width

$$D = \frac{\left(\frac{E_x + E_y}{2}\right) t^3_{tube}}{12\,(1-v_{12}^2)};$$

and D_q, the transverse shear – stiffness parameter,

$$D_q = \frac{G_{xy} h^2}{h - t_{tube}};$$

where "h" is the distance between the exterior and interior face plies. The formula for the thickness ratio of the tube is

$$TR_{tube} = \left(\frac{\bar{D}_y}{\bar{D}_x} \right)^{\frac{5}{6}} \left(\frac{\bar{E}_x \bar{E}_y - \bar{E}_{xy}^2}{12 \bar{E}_y \bar{D}_x} \right)^{\frac{1}{2}} \frac{l^2}{r_{tube}}$$

where

TR = Thickness ratio

$$\overline{D}_x = \frac{E_x t_{tube}^3}{12(1-v_{12}^2)}$$

and for the buckling analysis to be accurate, it is suggested that TR be roughly ≥ 500.

5. CONCLUSION

Thus the various Engineering problems along with their classification are studied. A comparison of engineering analysis and system identification along with structural modeling are done.

The failure analysis of the Unmanned Aerial Vehicle (UAV) has been carried out which are summarised below;

- A process to evaluate the stresses and proximity to failure of a tubular composite.
- Laminate under combined bending, axial, and torsional loads;
- A corrected formula to predict the buckling of slender tubular compression members;
- A satisfactory prediction tool for the torsional buckling of thin-walled tubular structures of non-uniform cross-section fabricated with an orthotropic composite.

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