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ERRATA

Page 3 : § 1.2 " Accuracy " instead of " Inaccuracy "

Page 7 ; § 4.4 ".... two columns of Figure 9 "
"When examining the shape of this mode (Figure 12) ... "
" (e.g a too high Young's modulus) "
instead of " (e.g too high a Young's modulus)

SYMBOLS

Page 11 :

δ_R Real component of element displacement's response vector

δ_I Imaginary component of element displacement's response vector

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STRATEGIES FOR DYNAMIC MODELIZATION OF A HELICOPTER STRUCTURE

Serge DURAND – Victor YANA

AEROSPATIALE
Helicopter Division
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THE CITY UNIVERSITY, LONDON, EC1V OHB, ENGLAND.

STRATEGIES FOR DYNAMIC MODELIZATION
OF A HELICOPTER STRUCTURE

Serge DURAND – Victor YANA

AEROSPATIALE, HELICOPTER DIVISION

ABSTRACT

The modelling of complex structures such as helicopter fuselage is a key point since it is intended to achieve the dynamic adaptation with respect to the main excitations, to study the rotor-structure couplings and to make forced response calculations.

An efficient dynamic model of a helicopter structure is not easy to define. This paper exposes a methodical approach based on three precepts : dividing difficulties, correlating the experience and simplifying the reality.

This methodical approach was successfully applied to the ECUREUIL helicopter tail section. It looks to be promising for a quicker analysis and solution of the structural dynamics problems.

INTRODUCTION

The reduction of vibrations on helicopters involves three different actions :

1. Dynamic optimization of the rotor blades
2. Installation of efficient anti-vibration systems
3. Dynamic optimization of the structure.

Constant improvements were already made by Aérospatiale on the first two points during the last decade.

It is now possible to obtain a good dynamic adaptation with respect to aerodynamic excitations.

As regards the rotor-structure interface, the various resonator passive suspensions such as the SARIB and its derivatives as well as the higher harmonic control look very promising.

The work performed on the third point did not lead to results as satisfactory as for the first two points.

It was therefore necessary to make a special effort to deal with the complex problem of the dynamic adaptation of structures.

Considering the differences obtained during the previous studies between the results from calculations and the results from tests, we re-examined our approach to the problem.

Before coming to the heart of the matter, let us remind which are the major problems encountered in structure dynamics.

1 – STRUCTURE DYNAMIC PROBLEMS

A helicopter rotor is a powerful source of vibrations.

The harmonic excitations generated by the rotors lead to mechanical vibrations which affect the comfort of the passengers and generate fatigue phenomena for mechanical parts. Moreover, these excitations may result in dangerous instabilities.

Being familiar with the dynamic characteristics of a rotorcraft structure is one of the keys to solving the vibration problems encountered on helicopters.

The main structural problems are :

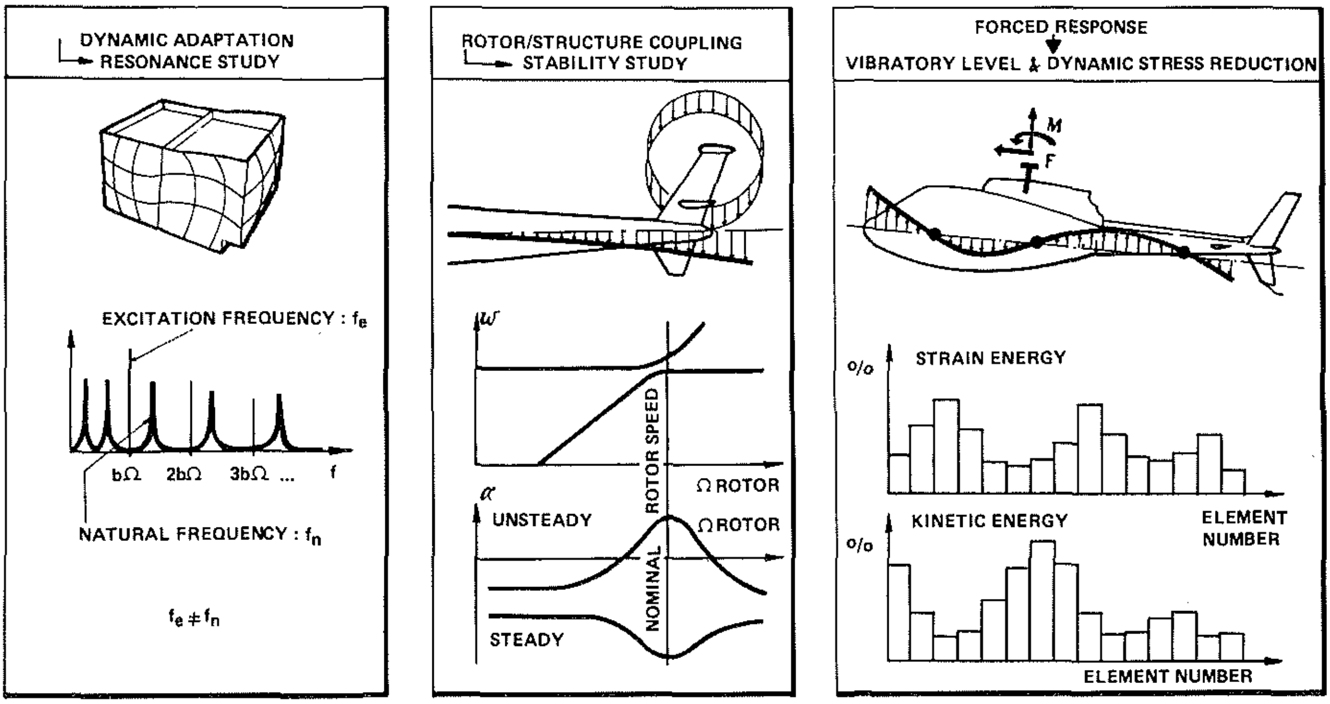


Fig. 1 : STRUCTURE DYNAMIC PROBLEMS

1.1 – DYNAMIC ADAPTATION

Firstly, DYNAMIC ADAPTATION of the structure as regards excitations generated by the main and tail rotors.

It is a RESONANCE STUDY.

Coincidence must be avoided between natural and excitation frequencies. If there is coincidence, then the mode shape must be examined to find out whether this mode is really excited.

On a helicopter, the dominant excitation frequencies are produced by the main rotor and the tail rotor. When applicable, the excitation frequencies due to certain weapon systems must also be added.

Dynamic adaptation not only concerns the overall aircraft in flight but also the different optional equipment installation configurations and also the hydroelastic problems associated with structural fuel tanks.

1.2 – ROTOR/STRUCTURE COUPLING

Secondly, knowledge of modal characteristics involves a *study of rotor/structure couplings.*

This study is very important because the couplings could result in dangerous instabilities.

For instance, the coupling between the first drag mode of the soft-in-plane tail rotor blades and a mode which affects the aircraft tail section.

In this instance, it is important to have a good model of the tail section together with a simplified model of the front section.

Inaccuracy of the predicted stability depends, among other things, on the quality of the modal computation.

1.3 – FORCED RESPONSE STUDY

The third main problem is the forced response study.

This computation is indispensable for dynamic optimization of the structure.

It is also indispensable for optimizing the setting of the suspension between the rotor and the structure (e.g. SARIB), particularly for excitations such as bending moments generating pitch movements and shear loads, which both have significant effects on the cabin vibration level.

In paragraph 5.2 it will be shown that by considering the location, the direction and the excitation frequency, histograms of the strain or kinetic energy distribution can be drawn up for each finite element.

The list of vibration problems encountered on a helicopter structure shows how important it is to have a good dynamic model of the structure. We hope that this will be achieved by using an appropriate strategy.

2 – STRUCTURE DYNAMIC COMPUTATION

The finite element method is currently used when designing a structure as complex as that of a helicopter, from a dynamic standpoint.

The principle of this method is to use a discrete displacement field with subregions and to reconstitute the complete region using interpolation functions.

This is a discrete approach to a continuum problem.

The code used by the Helicopter Division was developed by the University of Liege (Belgium) and is known as SAMCEF (*).

The stiffness matrix K is obtained from the strength of materials law. The mass matrix M is obtained from the solid mechanic's law. This matrix may be diagonal, a lumped mass matrix, if the mass is distributed on the grid nodes, or a consistent mass matrix if an interpolation function is used to distribute the mass.

The damping matrix D is obtained from the energy dissipation law ; damping can be introduced in different ways (proportional, modal, per element, per dash pot).

The three main computations covered in dynamics are modal analysis, harmonic analysis and transient analysis.

2.1 – MODAL ANALYSIS

This computation leads to the knowledge of the modal scheme : natural frequency, mode shape and generalized mass.

It consists in solving an eigenvalue problem in the form of :

$$[K]q = \omega^2 [M].q.$$

The originality of SAMCEF lies in the resolution method used (Ref. 1) : The Lanczos algorithm which works on the complete system outside central memory and eliminates the condensation stage, for a lower cost.

The method has been considerably improved to overcome the standard defects of the algorithm :

- non-detection of rigid modes
  reorthogonalization at each iteration
- skipping certain eigenvalues
  restart procedure of the starting random vector
- appearance of spurious solutions
  error bounds calculation

2.2 – HARMONIC ANALYSIS

Here it is a question of the forced response (i.e. after disappearance of the response due to the initial conditions) to a harmonic excitation calculated for a certain excitation frequency.

The fundamental dynamic balancing equation is expressed as :

$$-\omega^2[M]q+j.\,\omega[D]q+[K]q=F_0$$

where :

q : complex displacements vector
F : excitation force

The solution is obtained by modal superposition (Ref. 2).

2.3 – TRANSIENT ANALYSIS

This analysis concerns a response to an excitation known in time in a deterministic manner.

The fundamental equation is expressed as :

$$[M]\ddot{q}(t)+[D].\dot{q}(t)+[K]q(t)=F(t)$$

The solution is obtained by modal superposition or direct integration (Ref. 2).

3 - DYNAMIC MODELIZATION STRATEGY

An efficient dynamic model of the structure is not easy to define ; it is only with a certain amount of difficulty that a working knowledge of the dynamic characteristics of helicopter structures has been obtained over the past ten years.

These difficulties stem mainly from problems inherent to the modelization of junctions between the various substructures and their significant effect on modes as well as from problems inherent to the characteristics of the new materials used in the aeronautic field ; from problems inherent to the fluid/structure interaction in structural fuel tanks that take a significant amount of the overall helicopter volume, and this mainly on small aircraft ; and finally from problems inherent to modular design of small aircraft, e.g. bubble type cabin.

Difficulties encountered led us to adopt a systematic approach. Our strategy thus rests on three precepts :

(8) Système d'Analyse des Milieux Continus par Eléments Finis (Continuous medium analysis system using finite elements)

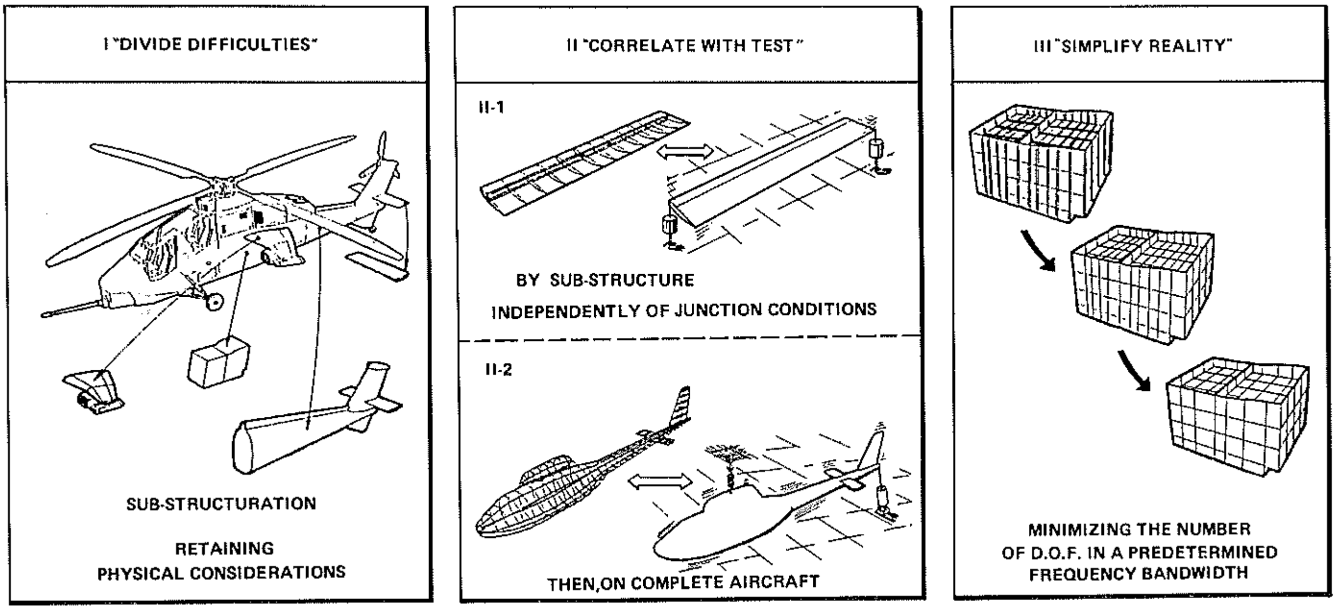


Fig. 2 : DYNAMIC MODELIZATION STRATEGY

3.1 – DIVIDE DIFFICULTIES

Given the modular character of a helicopter fuselage, it can be broken down into a certain number of substructures.

Dividing the difficulty consists in dealing with the following problems separately :

1. Is the substructure correctly modelized ?
2. Is the junction between different substructures correctly modelized ?

For this, experimental correlations are performed without reference to the boundary conditions then the links are identified.

3.2 – CORRELATE WITH TEST

Experimental correlations are indispensable to obtain a reliable finite elements dynamic model with which modifications can be suggested with a certain degree of confidence during the development phase.

Experimental correlations are performed substructure by substructure with a final correlation for the complete structure.

The model is thus validated for each substructure independently from the links between substructures.

3.3 – SIMPLIFY REALITY

In spite of the increased power of computers, computation times increase very quickly with the number of degrees of freedom, particularly in modal analysis when the resolution of the eingenvalue problem ($kq = \omega^2 Mq$) takes three to four times longer than the resolution of a linear system ($kq = f$) in standard static analysis.

The constant concern of dynamics analyst shoult be the minimization of the number of degrees of freedom (1) in a given frequency bandwidth or for a given number of modes.

The action consists, from a grid which is too fine (e.g. a grid designed for a static computation) in gradually reducing the number of degrees of freedom by simplifying the grid, until a significant natural frequency gap is obtained. This action is of course only effective in a given frequency bandwidth.

It may be applied quite easily with the help of the facilities provided by the GRATIS (2) automatic grid preprocessor developed at Aérospatiale's Marignane works.

The two-fold advantage of this precept is the reduction in computer costs but also the model utilization flexibility in particular for the forced response and parametric studies.

(1) For simplification purpose, degree of freedom is abbreviated as d.o.f.

(2) G.R.A.T.I.S. : Générateur Rapide Automatique par Technique Isoparamétrique (Isoparametric Technique Fast Automatic Generator)

4 – APPLICATION TO THE ECUREUIL HELICOPTER TAIL SECTION

These strategies were applied to the tail section of the ECUREUIL helicopter. This choice was dictated by the availability of the elements for tests and mainly by the fact that tail sections of helicopters excited both by the main and tail rotors are at the centre of many dynamic problems.

4.1 – SUBSTRUCTURATION

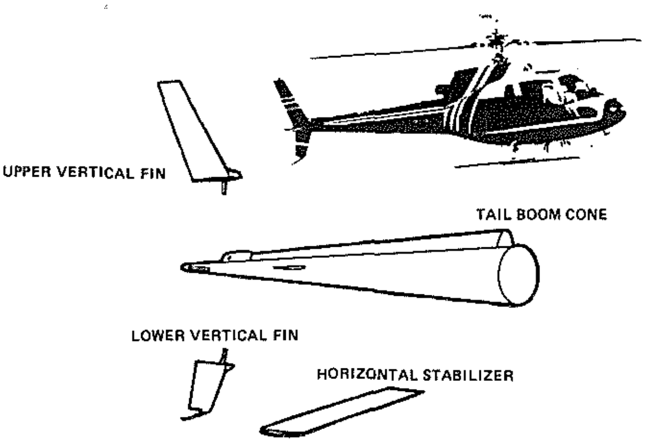


Fig. 3 : SUBSTRUCTURATION OF THE ECUREUIL HELICOPTER TAIL SECTION

The tail section was divided into the following substructures in accordance with our strategies :

- Horizontal stabilizer
- Lower fin
- Upper fin
- Tail cone

4.2 – EXPERIMENTAL CORRELATION FOR THE HORIZONTAL STABILIZER

For practical and test control reasons, the experimental correlation was performed under «free-free» boundary conditions.

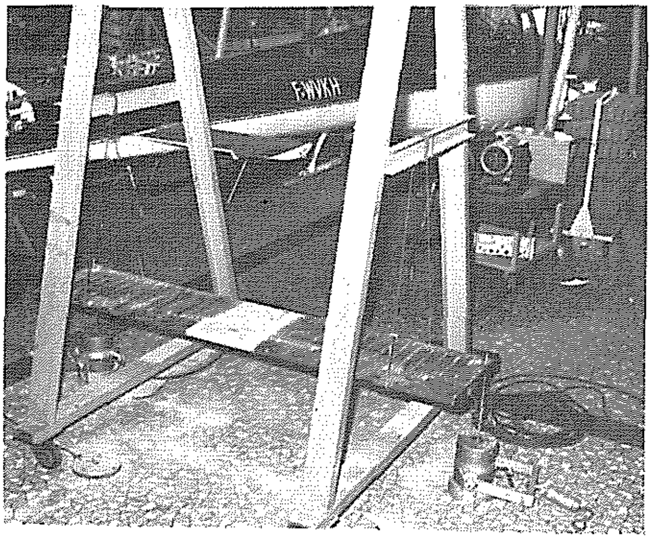


Fig. 4 : HORIZONTAL STABILIZER VIBRATION TEST

It was decided that we should limit ourselves to the first two natural modes.

Resetting was carried out by action on the Young's modulus E and on the pressure coefficient γ.

The correlation is excellent both in mode shape and natural frequency.

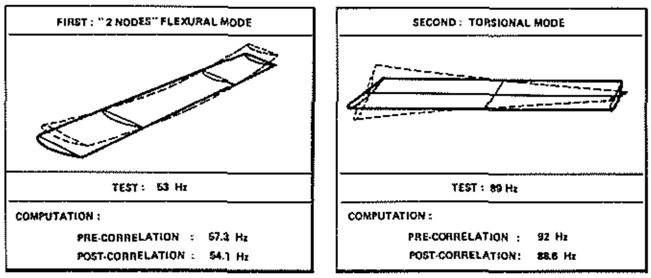


Fig. 5 : CORRELATION OF THE FIRST TWO MODES

The reset model will favour the action of minimizing the number of degrees of freedom with other boundary conditions.

4.3 – MINIMIZING THE NUMBER OF DEGREES OF FREEDOM

Minimization was achieved under the boundary conditions : «simply supported at tail boom junction points» nearest reality. It should be noted that minimization is valid only if it is achieved under realistic boundary conditions.

In fact, if minimization were achieved in «free-free» boundary conditions, the final model could have proved not to be satisfactory for the boundary conditions on aircraft. Choosing the boundary conditions is therefore essential, since according to the selection, a single area may contribute to either the kinetic energy or the strain energy.

The minimized model includes 772 d.o.f. for an original model of 1361 d.o.f.

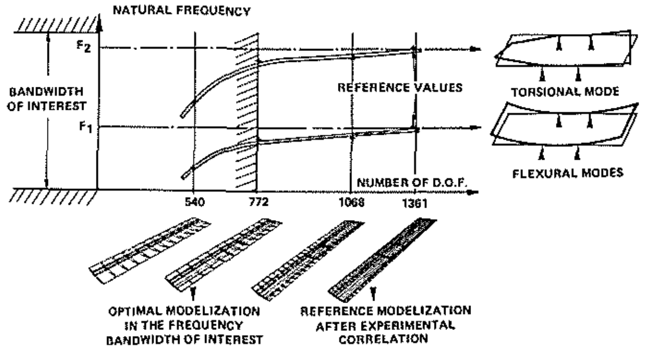


Fig. 6 : MINIMIZING THE NUMBER OF d.o.f.

4.4 – ASSEMBLY AND LINKS PROBLEMS

After experimentally correlating (precept 2) and minimizing each of the substructures (precept 3), they were assembled to obtain a complete model of the entire Astar tail structure.

In accordance with our strategy, a tail boom assembly modal identification test was conducted.

The correlation between the experimental and computed modes may be examined.

The results are given in the first two columns of Figure

As a rule, the correlation is not very good for the modes in general.

For example, the first vertical bending mode, computed at 10.3 Hz was found at 8.5 Hz by testing ; the third mode, computed at 18.5 Hz was found to be 16 Hz by testing.

When examining the shape of this mode (Figure 9) it is noted that there is a very large movement of the fins.

Given that assembly modes are poorly correlated whereas those of the substructure correlate very well, the problem can only be one of links. The difficulty has been divided (precept 1).

We have adopted a static approach to the problem of link modelizations, based on two complementary techniques :

1) Fine modelization of the link areas from which a simple equivalent modelization is derived.
2) Obtaining equivalent modelization from static tests.

This approach is adopted with a view to simplifying reality (precept 3).

We shall consider technique No. 2 using the example of the tail boom/wall link.

In reality there is a certain flexibility in the clamping achieved for the test.

In order to identify this fitting flexibility, which is as it were a link stiffness, a static test was conducted and the elastic lines were recorded using dial indicator gauges for a vertical, then a lateral load at the end of the cone.

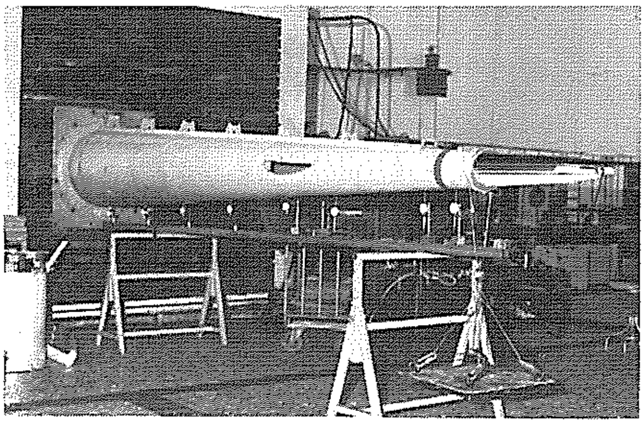


Fig. 7 : TAIL CONE STATIC TEST

The deflections measured were greater than those computed on the «as clamped» supposition.

When the measured deflection is subtracted from the computed deflection, the difference is linear.

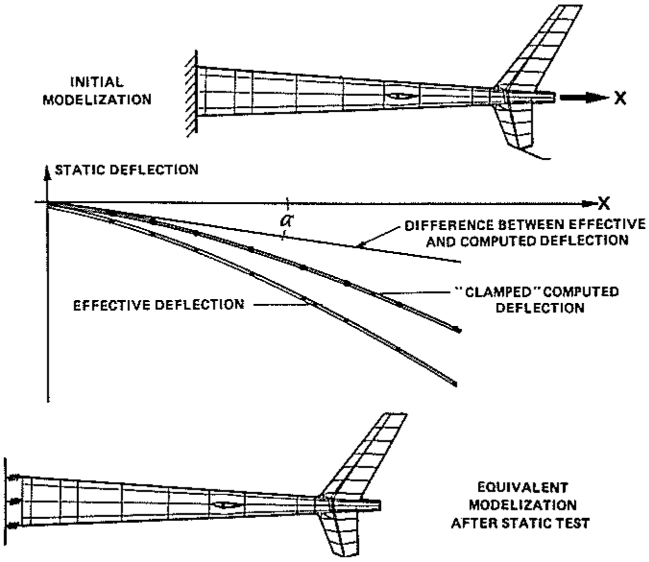


Fig. 8 : IDENTIFICATION OF WALL-CLAMPING FLEXIBILITY

The slope value is the same for the vertical load test as for the lateral load test.

The difference CANNOT stem from an overall modelization error (e.g. too high a Young's modulus) since the tail boom has already been subjected to an experimental correlation in modal analysis, under «free-free» boundary conditions.

It can ONLY be a question of fitting flexibility.

It can be modelized simply by a flexural spring at the cone/wall junction.

The junctioning conditions have also been modelized for the stabilizer/cone, fins/cone and skid/lower fin junctions.

OVERALL RESULTS

Once the link stiffness has been readjusted the experimental correlation is EXCELLENT in the selected frequency band (5 - 45 Hz), i.e. for the first eight modes, both in natural frequencies and mode shape.

NOTE : For quantification of the difference between measured and calculated mode shape through single scalar :

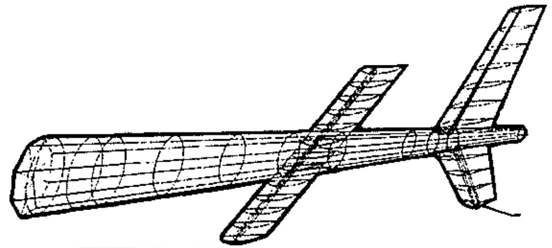
ABSOLUTE MEAN DEVIATION :

$$m_i = \frac{1}{n} \cdot \sum_{j=1}^{n} \sqrt{\left| q_{ij\ \text{measured}} - q_{ij\ \text{computed}} \right|}$$

STANDARD DEVIATION :

$$\sigma_i = \frac{1}{n} \cdot \sum_{j\quad 1}^{n} \sqrt{\left(q_{ij\ \text{measured}} - q_{ij\ \text{computed}} \right)^2}$$

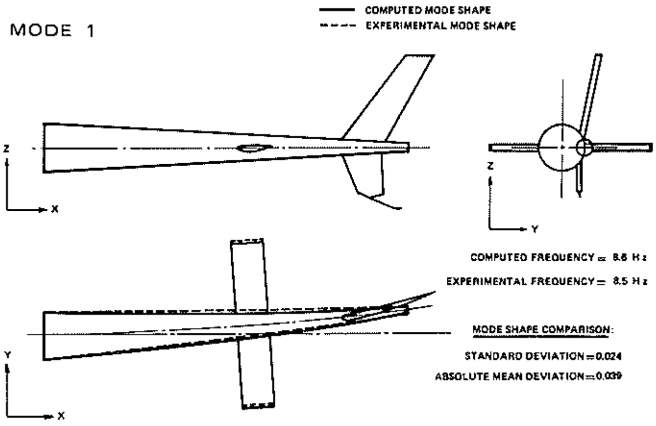
where q_{ij} is the $j^{\underline{th}}$ component of mode Number i.



	NATURAL FREQUENCY (H_z)				
MODE SHAPE N	EXPERIMENTAL	COMPUTED BEFORE JUNCTIONS CORRELATION	DEV.	COMPUTED AFTER JUNCTIONS CORRELATION	DEV.
1.	8.5	10.3	14 °/o	8.6	1 °/o
2.	9.6	11.7	18 °/o	9.5	1 °/o
3.	16.	18.5	14 °/o	17.	6 °/o
4.	25.1	26.2	4 °/o	25.1	0.4 °/o
5.	non identified	29.5	—	28.1	—
6.	29.6	32.7	9 °/o	29.7	0.3 °/o
7.	33.	37.7	12 °/o	33.1	0.3 °/o
8.	39.6	44.1 .	10.2 °/o	42.3	6.4 °/o

Fig. 9 : OVERALL RESULT BEFORE AND AFTER JUNCTION RESETTING

MODE 1

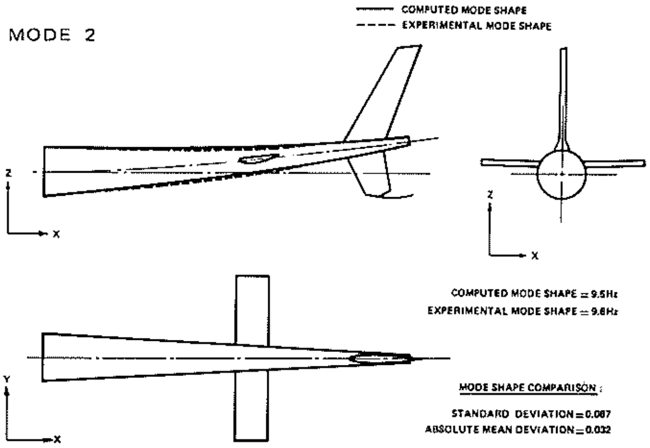


COMPUTED FREQUENCY = 8.6 Hz
EXPERIMENTAL FREQUENCY = 8.5 Hz

MODE SHAPE COMPARISON:
STANDARD DEVIATION = 0.024
ABSOLUTE MEAN DEVIATION = 0.039

Fig. 10

MODE 2

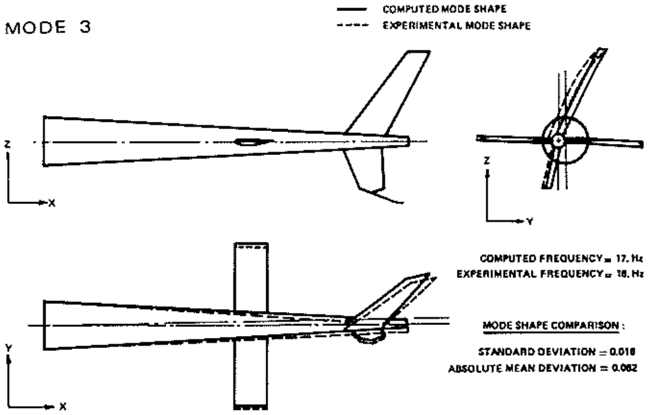


COMPUTED MODE SHAPE = 9.5Hz
EXPERIMENTAL MODE SHAPE = 9.6Hz

MODE SHAPE COMPARISON :
STANDARD DEVIATION = 0.087
ABSOLUTE MEAN DEVIATION = 0.032

Fig. 11

MODE 3

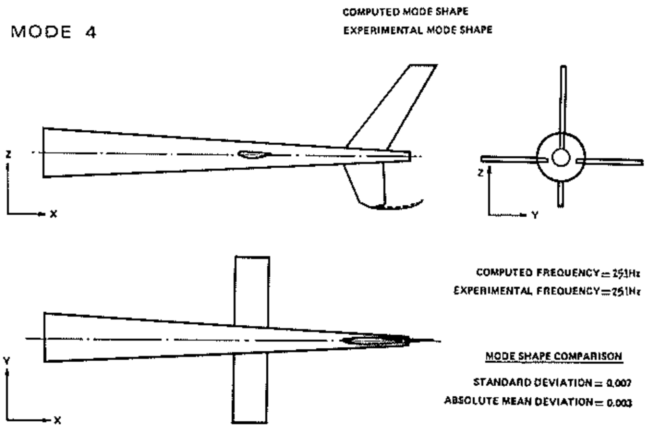


COMPUTED FREQUENCY = 17. Hz
EXPERIMENTAL FREQUENCY = 16. Hz

MODE SHAPE COMPARISON :
STANDARD DEVIATION = 0.018
ABSOLUTE MEAN DEVIATION = 0.082

Fig. 12

MODE 4

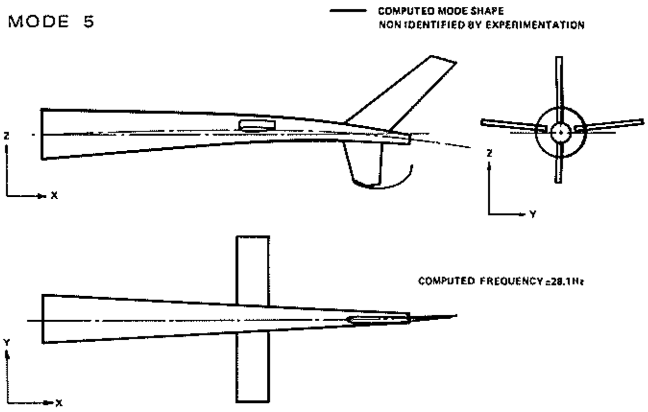


COMPUTED FREQUENCY = 25.1Hz
EXPERIMENTAL FREQUENCY = 25.1Hz

MODE SHAPE COMPARISON
STANDARD DEVIATION = 0.007
ABSOLUTE MEAN DEVIATION = 0.003

Fig. 13

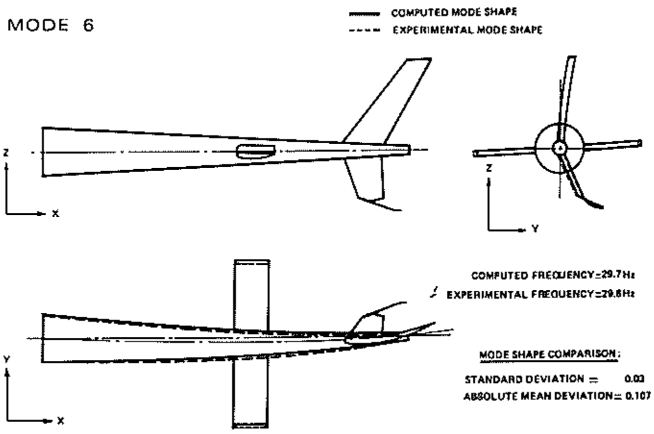
MODE 5



COMPUTED FREQUENCY = 28.1Hz

Fig. 14

MODE 6



COMPUTED FREQUENCY = 29.7Hz
EXPERIMENTAL FREQUENCY = 29.6Hz

MODE SHAPE COMPARISON :
STANDARD DEVIATION = 0.03
ABSOLUTE MEAN DEVIATION = 0.107

Fig. 15

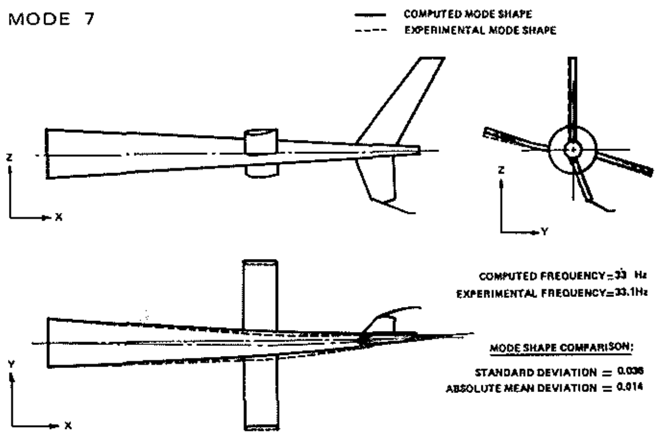


Fig.16

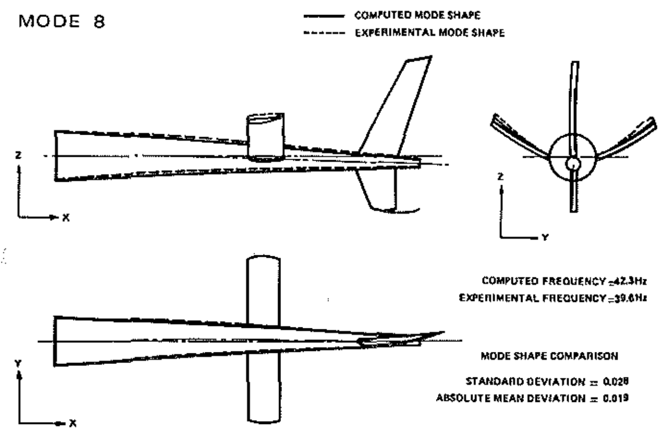


Fig. 17

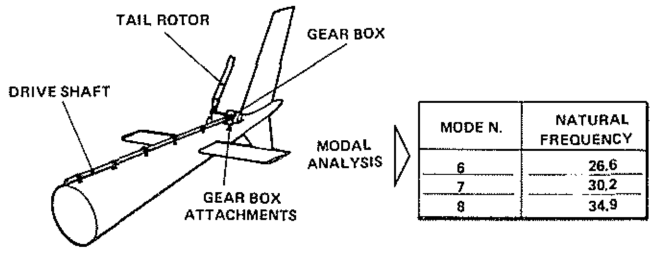
5 – VIBRATION PROBLEMS

5.1 – RESONANCE STUDY

We have a good model of the whole ECUREUIL tail section for a wide frequency bandwidth.

For convenience and test control purposes, the modal tests were conducted without the transmission assemblies.

By integrating tail rotor drive shaft, tail gearbox and tail rotor to the model, the dynamic adaptation of this section can be examined for excitation frequencies of 34 Hz (tail rotor unbalance) and 38.3 Hz (main rotor 6 Ω).



MODE N.	NATURAL FREQUENCY
6	26.6
7	30.2
8	34.9

Fig. 18 : RESONANCE STUDY OF THE ECUREUIL TAIL SECTION

It can be noted that there are resonance hazards on the tail rotor unbalance excitation.

Considering the density of modal base in this area, we studied the modification possibilities using a forced response calculation.

5.2 – FORCED RESPONSE STUDY

5.2.1 – Forced Response Strain and Kinetic Energy Approach

This approach aims at reducing overall structure vibration by taking into account the response participation of all modes for a particular load application at a particular frequency and during one period (Ref. 3 and 4).

It is assumed that the structural elements having the highest value of strain or kinetic energy are indicative of the best candidates for structural modification or mass displacement. This method was developed by SCIARRA of BOEING VERTOL.

The expression for the maximum damped forced response element STRAIN energy within a period is :

$$\boxed{(E_S)_{max.} = \frac{1}{4}\left\{\delta_R^T.K_e.\delta_R + \delta_I^T.K_e.\delta_I + \sqrt{(\delta_I^T.K_e.\delta_I - \delta_R^T.K_e.\delta_R)^2 + (\delta_R^T.K_e\,\delta_I + \delta_I^T.K_e.\delta_R)^2}\right\}}$$

The expression for the maximum damped forced response element KINETIC energy within a period is :

$$\boxed{(E_K)_{max.} = \frac{1}{4}\left\{\dot\delta_R^T.M_e.\dot\delta_R + \dot\delta_I^T.M_e.\dot\delta_I + \sqrt{(\dot\delta_I^T.M_e.\dot\delta_I - \dot\delta_R^T.M_e\,\dot\delta_R)^2 + (\dot\delta_R^T M_e\,\dot\delta_I + \dot\delta_I^T.M_e.\dot\delta_R)^2}\right\}}$$

Elements with the highest energies indicate those that are the most responsible for the structural dynamic amplification.

The forced response strain energy approach gives direct information at the specific excitation frequency of interest.

5.2.2 – Forced Response of the ECUREUIL Helicopter Tail Section

The method exposed above will be a valuable aid to study the problem regarding the tail section of the ECUREUIL helicopter as excited by the tail rotor unbalance.

The unbalance excitation is still present on the aircraft since balancing the ECUREUIL tail rotor is a particularly delicate operation.

The unbalance excitation is simulated at tail rotor by the exciting force forces and moments :

$$FY = FO \ . \ \sin\ (\Omega_{TR}\ t)$$

$$FZ = FO \ . \ \sin\ (\Omega_{TR}t - \pi/2)$$

with Ω_{TR} = Tail rotor circular frequency

Examining the histograms obtained by the «Forced Response Energy Method» shows that the elements of the horizontal stabilizer-tail boom junction area represent 7 % of strain energy.

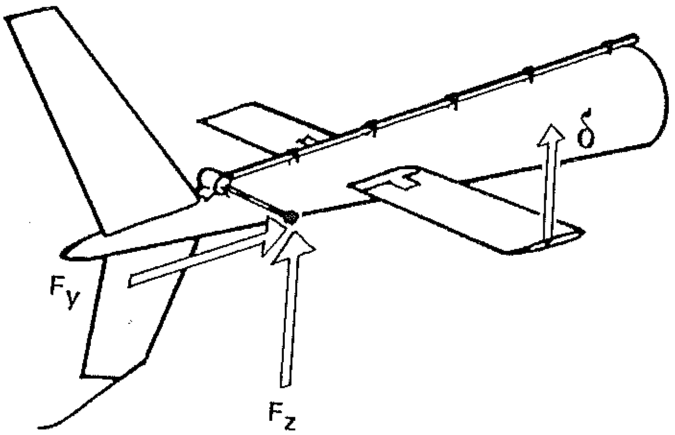


Fig. 19 : FORCED RESPONSE STUDY OF THE ECUREUIL TAIL SECTION

It seems appropriate to reinforce this area, which was achieved by replacing the material. The forced response calculation at iso-excitation shows improvements. The modification resulted in a 34 Hz vibration level as divided by 1.6 at the tip of horizontal stabilizer. The total strain energy was divided by 3.

The modification is embodied in production aircraft ; it allowed improving the fatigue strength of the ECUREUIL tail section.

CONCLUSION

The modelling of complex structures such as the helicopter fuselages is a KEY point since it is intended to achieve the dynamic adaptation with respect to the main excitations, to study the rotor-structure couplings and to make forced response calculations.

In the past, this type of modelling raised numerous difficulties.

It was necessary to deal with the problem methodically.

We therefore wanted to establish a method based on three precepts : dividing the difficulty, correlating the experience and simplifying the reality.

Softwares were also developed to ease the analysis in forced response mode.

We applied our approach to an example : the tail section of an ECUREUIL.

First, our modelling led us to a very good correlation between natural frequency modes as measured and calculated in a particularly rich mode base (nine modes).

Then, the quality of the mode base allowed us to make forced response calculations and to find by the «Forced Response Energy Method» a solution to a problem of unbalance excitation issued from the tail rotor.

The results look PROMISING.

They tempt us to apply our approach to the more ambitious problem of the dynamic optimization of a complete helicopter structure.

In this view, an appreciable structural weight saving can be envisaged (from 2 to 5 %) owing to - for example - either the elimination of structural reinforcements due to a poor dynamic adaptation or the elimination of cabin anti-vibrators then becoming of no use.

We can now envisage a quicker analysis and solution of the structural problems encountered during the development phase of a helicopter.

In this view, it should be noted that this general study of the ECUREUIL tail section sub-structures will be used for any similar aircraft for which the application is lighter since it takes advantage of the experience acquired.

The progress made in the dynamic optimization of structures owing to the strategy and the tools exposed above, associated with the recent improvement achieved both in the suspensions between the rotor and the structure and in the dynamic optimization of the blades, would result, in a near future, in a significant improvement of the vibratory level on helicopters.

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SYMBOLS

K	Stiffness matrix
M	Mass matrix
D	Damping matrix
ω	Circular frequency
q	Structural displacement response vector
F	Applied force
t	Time
R	Real component of element displacement's response vector
I	Imaginary component of element displacement's response vector
K_e	Element stiffness matrix
M_e	Element mass matrix
E_s	Strain energy
E_k	Kinetic energy
Superscripts	
T	Matrix transpose
.	Time derivation