한국소음진동공학회 1996년도 춘계학술대회논문집 pp. 361-369.

Validation of the Strain Pattern Analysis(SPA) Measuring Technique

헬리콥터 Blade 의 모드해석에 적용된 응력패턴해석 계측기법의 타당성

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ABSTRACT

The accurate prediction of modal parameters of a rotating blade is an important requirement in the assessment of the dynamics of a helicopter rotor. Indeed, predictions of flight loads and stability are normally dependent on initially predicting the undamped mode shapes. A measuring technique, known as Strain Pattern Analysis (SPA), appears to be the most successful technique for measuring the mode shapes of rotating blades. This method was developed to be used on actual aircraft so no attempt was made to measure rotating mode shapes directly in order to validate the SPA method.

This report summarizes results from experimental investigations which were carried out to validate the SPA method for the prediction of aerodynamically damped modes of a rotating blade. A series of modal tests were carried out on two rotor models in which the non-rotating, undamped and aerodynamically damped rotating modes were measured directly (strain and displacement patterns). It is shown that the SPA method to be very successful in itself but there are a number of limitations in validating this technique. To provide data which could be used to confidently validate theoretical prediction codes, existing limitations should be addressed.

Introduction:

The structural dynamics of flexible rotors is of fundamental concern to helicopter and wind turbine manufacturers. To design light and cost efficient flexible blades, a heavy reliance has to be placed on computer codes which attempt to predict the behavior of the rotor in the presence of turbulent winds. But validation of these codes is very difficult to obtain. Usually, some of the system matrices are periodic in nature, which excludes a straightforward normal mode analysis and requires a time step approach. To reduce the number of degrees of freedom required for the time step analysis, it is convenient to use the undamped normal modes derived from constant system matrices. In horizontal axis rotors it is the structure stiffness matrix which is periodic, because of gravity, and for the vertical axis machines it is the aerodynamic matrix. Understanding the modal parameters of a rotating blade becomes more complicated in that the frequencies, damping and mode shapes are a function of rotational speed and may vary over a wide range. Consequently, determining the modal characteristics of a rotor blade in static conditions is not sufficient; one must know how parameters have shifted at the proposed method was developed at Royal Aircraft Establishment(Farnborough)[1,2] known as Strain Pattern Analysis. The method involves measuring non-rotating mode shapes together with the strain distribution, or strain patterns along the rotor blade; to each modal shape there is a corresponding strain pattern. The rotating mode shapes are then determined by resolving the rotating strain distributions in terms of the non-rotating strain and displacement patterns. The only available comparison that could be made with the shapes obtained by the SPA method has been theoretical predictions. To validate both the existing methods of measurement and the theoretical prediction codes, the following case studies were conducted to measure the rotating mode shapes directly.

Apparatus and Procedures:

Test Rig (I): Two rotor blades, each of 600mm span were mounted on a shaft to rotate in a horizontal plane. The blades were mounted as cantilevers from a circular hub on the shaft. Stiff radial arms above each blade supported non-contacting displacement transducers that measured the flap-wise bending movements of the blade. Suitable displacement transducers were used which had a degree of analogue signal amplification in the transducer head. One blade was instrumented with strain gauge bridges sensing flap at two stations between the root and tip, with small amplifiers mounted on the rotor shaft. An electro-dynamic shaker mounted co-axially with the shaft, below the blades, was used to excite the blades in flap via two piezo-ceramic force gauges. The blade system was mounted on operating speed of the rotor.

At the moment, theoretical estimates used for the rotating modes are based on measurement of the non-rotating modes. An improvement upon this a test tower having an electrical variable speed drive and equipped with a slip-ring assembly, and a system of speed measurement with continuous display. Figure 1 shows the experimental arrangement for this set up.

Test Rig (II): A vertically mounted hollow steel shaft supported via two bearing by a four legged frame, was used to carry all the rotating components. The 1.5 meter diameter blades were instrumented with strain gauges, non-contacting displacement transducers and two force transducers from which the measured signals were transmitted to the stationary frame via slip-rings. The rotor blade assembly was belt driven by a 15hp DC motor with a variable speed controller. For safety, the framework was encircled by two layers of archery netting curtains. Two rotor blades each of 1105mm length, untwisted and with uniform chord of 75mm were mounted on the shaft to rotate in a horizontal plane. These were mounted as cantilevers from the hub on the shaft. An attempt was made to achieve dynamic similarity between the rotor blade and an existing full scale blade by choosing a cross-sectional design, blade length and rotational speed that gave, theoretically, similar ratios between the flapwise bending mode frequencies and similar degrees of centrifugal stiffening as are present in the behavior of the Westland Lynx helicopter rotor blade; this was found to be achievable without including any additional flexibility or hinge at the root. A reasonable correspondence was also achieved for the torsional and lead-lay modes.

Figure 2 shows the experimental arrangement for this set up.

Rotor Blades: The blades comprised a shallow, light alloy, I-beam embedded in self-skinning foam providing an NACA0021 aerofoil external shape for the blade aerodynamic requirements. The I-beam comprised two wide flanges bonded together via a channel section web allowing its position to be adjusted to give any desired shear centre location. Figure 3 shows chord-wise cross section of the rotor blade.

Blade Excitation Method: Excitation of the blades was provided by an electro-dynamic shaker mounted co-axially with the shaft, above the blades. The link between the shaker and each blade was made through a stiff, light carbon fibre strut and two vertical flexible rods (to allow for any mis-alignments). A piezo-ceramic force transducer was inserted between each rod and blade and was attached to the blade by means of a light alloy clamp.

Displacement Sensor: Non-contacting displacement transducers (using eddy current induced by low energy RF transmissions) were chosen as being the most suitable. This system comprised a probe and a signal conditioning unit. Local displacement was measured by mounting the probe normal to the chord line of the blade, in the flap direction. Thin steel targets were placed at each measuring station. Specially designed slip-ring channels were used to conduct the output signals to the signal conditioning units.

Strain Sensor: To measure the required blade strain patterns, electrical resistance type strain gauge in conjunction with a suitable miniature type amplifier were used. Each blade was instrumented with twelve strain gauge bridges. Four strain gauges were placed at each station in the spanwise direction, one at each of the edges of the I-beam flanges. By altering the bridge connections of these gauges they could be used to measure, selectively, either flapwise or lead-lag bending strains and, additionally, torsion-bending strains. Exchangeable printed circuit boards were used to select the desired strain mode. All strain signal conditioning was mounted on the rotating shaft.

Signal Processing and Analysis:
A minicomputer-based analyzer was used to carry out Fourier analysis and other tasks using software programs. The capabilities of the system were, computation of power spectral density, cross spectral density, transfer function, coherence and correlation functions.

Data Processing Program: The excitation signals was generated by the computer in digital form and then stored in the memory of the Buffer DAC (8 bit amplitude resolution for the 2048 points). Using an internal 10 MHz clock, the excitation signal was output in one burst at a clock-controlled rate or in repeated bursts and then directed to the shaker to excite the structure. The output of the buffer DAC was fed through a low pass filter to smooth the stepped signal and then its amplitude controlled by a multiplying DAC if necessary.

A general data processing program was used and modified for the specific needs of the experiment. The program allowed test of a structure to be carried out using swept sine or stepped sine frequency excitation. The measurement of responses were recorded in the time domain and the results were then processed by a Fourier transformation program for frequency domain analysis. The excitation signals were generated from the buffer DAC using a clock synchronized with that performing the ADC function. This meant that an integer number of cycles of data could be acquired for the single frequency sine and thus leakage in the subsequent Fourier transformation was avoided.

For single sine frequency excitation, each data signal was resolved by multiplying with an internal digitally generated reference array (equivalent to the sinusoidal array in the Buffer DAC) and one in quadrature with it. The second harmonic introduced by multiplication was removed by averaging over an integer number of cycles (8, 16, 32, 64, ...). In order to account for the transient associated with frequency stepping at each frequency increment, the resolver output values were used to indicate convergence by repeating acquisition until the steady state had been reached and an acceptable accuracy obtained.

Data Analysis Program: The signals that were processed were time histories and some useful information extraction procedures may be performed whilst in the time domain. However, very often, transform methods are employed since they offer simplifications both in theory and physical interpretation. The most common method for determining modal characteristics of a system is to excite the structure with a sinusoidal input at a particular frequency. Then, allowing the response to reach the steady-state condition, the amplitude ratio of the output to input and phase shift of output relative to input is measured. By repeating this procedure for other frequencies in the band of interest, a complete picture of the amplitude and phase of the system frequency response function can be obtained.

Initially, swept sine excitation was used for measuring the blade resonant frequencies. The Fourier transform was used to determine power spectral density, cross spectral density and frequency response function (H1). From the real and imaginary components of H1, the frequency response function amplitude and phase were determined. The peaks in the amplitude indicated the system natural frequencies. To avoid leakage, the sweep only occupied part of the data window in order to allow the structure response to decay before the end of the window.

Having established the blade resonant frequencies, single sine excitation was used to excite the blade at each measured resonant frequency. Then the measured response signal was correlated with respect to the reference excitation signal via a resolver averaging process in order to yield the frequency response function amplitude for a particular measuring station. By this means the results of several tests (at a resonant frequency) were collated in order to obtain a mode shape. This approach provided a good signal-to-noise ratio.

Measurements and Results:

When the initial tests to measure the non-rotating blade mode shapes commenced, it was soon observed that;

1. since the blade tip deflection was restricted to ± 1mm, by the displacement transducer, a limit on the excitation force level existed.
2. when the blade was excited at the force level limit, the strain gauge responses were very low especially for the first flap-wise bending mode.
3. at a pre-set computer controlled excitation voltage level the actual excitation force measured at the blade varied significantly from one resonant frequency to the other due to the drop out offset.

From the above observations it was concluded that, in order to obtain sufficient output voltage from both the displacement transducer and the strain bridge, the resonant frequency of a mode should be excited at two different force levels. However, depending on the level of the blade non-linearity, the transfer or frequency response function could vary with the input force level and become distorted. As a result, the extracted modal parameters could vary significantly from one force level to another. The change in the transfer function curve is caused by the stiffness and/or damping non-linearities.

The first flap-wise bending mode was excited between 19 to 21 Hz frequency range. The input was adjusted to achieve three constant force levels of 0.5, 1.0 and 1.5 N. The transfer function amplitude and phase spectra from each force level were obtained and compared together. As shown in Figure 4, the transfer functions displayed reduction in the response amplitude at a fixed resonant frequency (i.e. damping non-linearity). With the knowledge of the type of blade non-linearity it was permitted to excite the blades at different force levels in order to obtain the resonant frequencies.

Identification of Flap-wise Bending Resonant Frequencies: In order to excite pure bending modes, the excitation force was applied through the blade chord-wise shear centre position. The blades were then excited between 10 and 500 Hz frequency range. The resonant frequency search was carried out at the rotational speed of 550 RPM (still air condition) in addition to the non-rotating condition. The frequency response functions, obtained from the above, showed that only the first six flap-wise bending modes could be excited. Figure 5-8 show sample time and frequency domain data.

Blade Mode Shapes: To obtain a mode shape, the blades were excited at a resonant frequency using steady-state sine excitation. The blade responses were measured and correlated with the reference excitation signal to yield the response magnitude and phase for a particular transducer position. The above procedure was carried out for the blade (non-rotating, rotating with aerodynamic shroud and rotating in still air conditions) and repeated for each span-wise station (12 positions) to obtain a mode shape.

Displacement Patterns: In measuring the displacement patterns of the blade, similar groupings to those used for strain were made. The twelve measuring stations were divided into six groups to cover the blade length using available displacement transducers.

During the non-rotating test some difficulties were experienced in measuring the first bending mode shape of the blade in the outboard region. This was due to the high blade flexibility which gave large tip displacement. By using an excitation force level of 0.25% of total amplifier power it was possible to obtain a reasonable first flap-wise bending displacement pattern. For the higher modes the above difficulty did not exist.

With the blade rotating, randomly excited vibration of the blade produced tip deflections well outside the displacement sensor range. This excitation was thought to be caused by aerodynamic forces and so an external aerofoil shroud was made which enclosed the blade and support arm. The shroud had 240mm chord with a 50% thickness to chord ratio. Unfortunately, the shroud did not reduce the amplitude of the randomly excited oscillations to a level at which the intended excited mode shape could be measured, as had previously been possible with stiffer blades [3-6]. As a results, only a set of non-rotating displacement patterns could be measured on the second rotor model. Measurements using test rig (I) provided three sets of displacement patterns (non-rotating, rotating with aerodynamic shroud and rotating in still air conditions) to be used for the SPA predictions. Figure 9 shows two sets of displacement patterns obtained from non-rotating and rotating tests.

Strain Patterns: Due to the limited number of anti-aliasing filters it was possible to capture only six data channels at a time. One channel was dedicated to record the input force and the five remaining channels for the blade response. To obtain a strain pattern, the twelve measuring stations were divided into three overlapping groups each containing five strain sensors. With the blade rotating, due to the noisy strain response, eight overlapping groups were considered. In doing so the accuracy in measuring the blade response at certain stations was increased. Strain pattern measurements resulted in two sets of data (non-rotating and rotating in still air conditions) for the case of test rig II and a complete set for the first test rig. Figure 10 shows two sets of strain patterns obtained from non-rotating and rotating tests.

SPA Predictions:

Having established the required strain and displacement patterns in both non-rotating and rotating conditions, the SPA method was applied. A linear combination of the non-rotating strain patterns to the rotating blade strain patterns was evaluated using least-squares error fitting procedure. The displacement pattern of the rotating blade was then derived from the same linear combination of the non-rotating displacement patterns. Similarly, the undamped rotating strain and displacement responses, in the still air condition, together with the aerodynamically damped strain readings were used to predict the aerodynamically damped displacements.

The SPA predictions for three flap-wise bending modes of the blade is shown in Figures 11-13. These predictions were compared with the directly measured displacement patterns and theoretical estimates (analysis based on virtual work principle). In general, there is a good agreement between the above data sets except for the first bending mode where there was a large tip deflection. Furthermore, due to physical restrictions in direct measurement of the aerodynamically damped displacement patterns, the SPA predictions could not be validated for this category.

Conclusions:

Strain Pattern Analysis is the most practical and cost effective measuring technique for predicting the modes of rotating blades. Present investigation has shown that accurate SPA predictions could be made if the technique is applied to stiff rotors. Confirmation of the SPA technique for the prediction of aerodynamically damped modes was possible since blade displacement response could be successfully extracted from the rotational environment. Using stiff rotors, SPA predictions can be extended to cover aerodynamically damped modes of a rotor subjected to an on-coming wind. However, an extremely fast data acquisition process is required to measure instantaneous displacement of a rotor blade at any azimuth station throughout a complete cycle of rotation in the presence of an air flow.

In the case of flexible rotor blades, considerable difficulties exist in measuring the rotating blade displacement response. Recent survey has shown that sensors based on optical methods(laser holography, optical fiber and laser type photoelectric sensors) are difficult to incorporate in the rotating environment. Further work is felt to be useful if a suitable displacement sensor which could cope with the large tip deflections becomes available.

Acknowledgments:

The experimental study on test rig (II) was carried out under Science and Engineering Research Council (SERC-UK), contract number. GR/F 03059.

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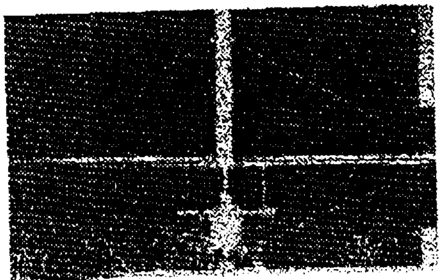

Figure 1- Test rig I.

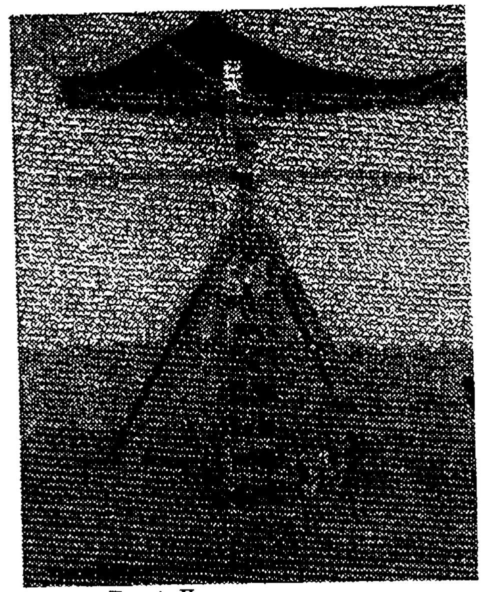

Figure 2- Test rig II.

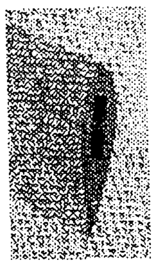

Figure 3- Cross section of the rotor blade.

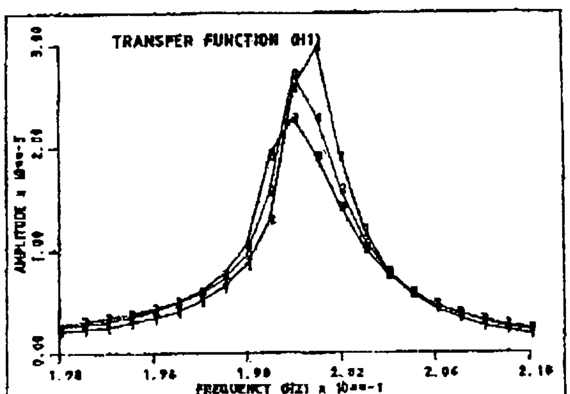

Figure 4- The blade non-linear characteristics for the 1[st] flap-wise bending mode.

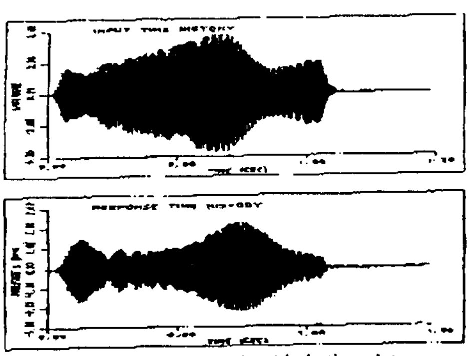

Figure 5- Sample non-rotating blade time data.

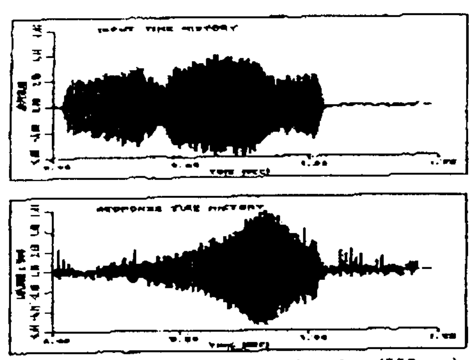

Figure 6- Sample rotating blade time data (550rpm).

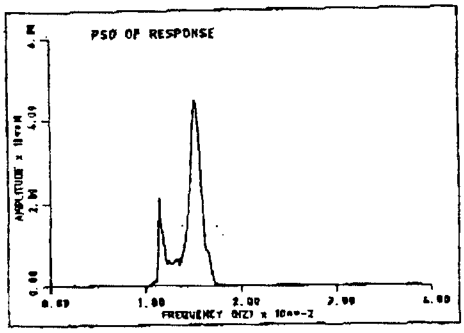

Figure 7- Sample non-rotating blade PSD data.

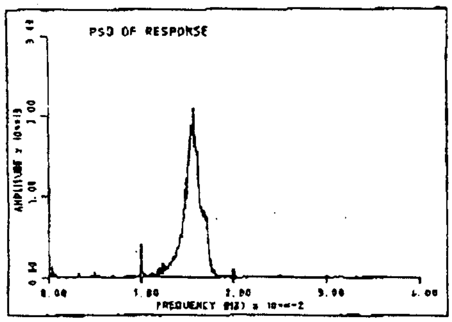

Figure 8- Sample rotating blade PSD data (550 rpm).

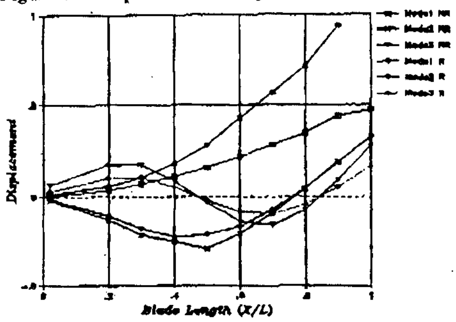

Figure 9- Two sets of displacement patterns (non-rotating and rotating conditions).

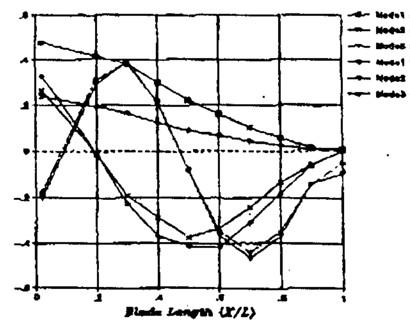

Figure 10- Two sets of strain patterns (non-rotating and rotating conditions).

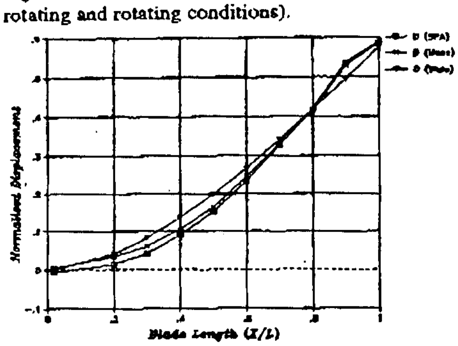

Figure 11- *

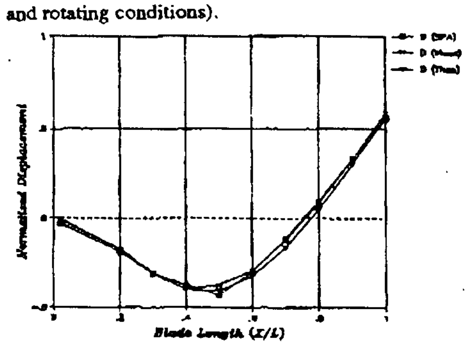

Figure 12- *

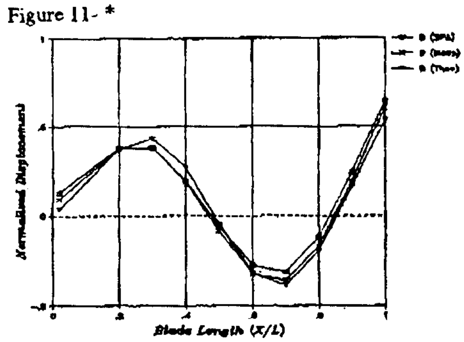

Figure 13- *

* Comparisons between the SPA predicted, directly measured and theoretically calculated displacements. Test case: 1st, 2nd, 3rd flap-wise bending modes in still air condition at 550rpm.