VIBRATION PREDICTION AND QUALITY CONTROL
OF A CENTRIFUGAL IMPELLER

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ABSTRACT
A multi-disciplinary approach was used to investigate and control the vibration of a 1.7 MW gas turbine impeller. Three techniques were used. First, determination of natural frequencies by Spectral Analysis of an impulse on the impeller. Second, Electronic Speckle-pattern Interferometry (ESPI) to assertain the modal shape of each natural frequency. Finally, 3-dimensional finite element analysis (3DFEA) of the whole impeller.

The analyses were synthesized to give an accurate determination of modal shapes and frequencies at test conditions and validate the 3DFEA. Effect of modifications could be predicted with confidence by the 3DFEA.

A single parameter was found which linked dimension to resonant frequency in disc mode vibration. A test was developed which gave a predictive technique for vibration quality control in a production environment.

NOMENCLATURE
A, B, C, D Empirical constants
f Frequency Hz
n Number of nodal diameters

INTRODUCTION
Development of a new engine usually reveals several features of construction which must be controlled if reliability is to be maintained. The European Gas Turbines (EGT) Hurricane is no exception.

The Hurricane is a 1.7 MW single shaft engine and has been described by Sanders et al (1991). It is equipped with a single stage radial compressor of 454mm dia. The nominal thickness of the impeller tip is 5.6mm. The vane thickness to tip thickness ratio is between 1:4.5 to 1:2.2 at the impeller outside diameter. The impeller of this compressor is of complex form (Fig. 1) with many potential modes of vibration, from the body, the full vanes and the splitter vanes and cross coupled vibration.

There was also concern that the frequencies of the disc modes would be sensitive to tip thickness. Thus an investigation, both experimental and theoretical, was undertaken. The objective was to find methods of predicting vibrational frequencies of each impeller in service from some measurement, either dimensional or otherwise, taken during the manufacturing process. A secondary objective was to find a way to assess the effect of any detail modification to the impeller on the vibrational frequencies.

IMPELLER FREQUENCY AND MODE TESTS
A series of frequency and mode tests were carried out on the impellers to enable:

(1) Construction of Campbell diagrams of the natural frequencies of the impeller backplate and vanes to detect likely problems with critical frequencies.

(11) Validation of the 3DFEA by comparison of test frequencies and mode shapes with the theoretical cold non-rotating case.
Tests were conducted to characterize the natural frequencies and associated mode shapes of the impeller backplate, full vanes and splitter vanes. The tests for the three features were conducted separately, but a note was made when modes appeared to coincide or were very close. For the tests, the impeller was mounted horizontally on a test stand, supported on its journal bearings.

**IMPULSE TEST**

The natural frequencies of free vibration were excited by impulse from a modal hammer which was selected for having an energy band covering the frequency range required for the test. The frequency range studied for these tests (0 - 14500 Hz), was that considered likely to be excited by engine order interaction with aerodynamics of the vanes and diffusers, and of the differences between them. The running range was taken as 110% of nominal with a 10% margin above that. The resonances were detected by placing a precision sound intensity meter in close proximity and normal to the feature being tested, with the output fed to an FFT frequency analyzer.

Peak-hold pseudo-averaging was used to build up a full spectral response as different modes were excited with the hammer. The zoom facility being used subsequently to gain more accurate frequency resolution of each resonant mode.

Initially, a survey was made of the fundamental frequencies of all full vanes and all splitter vanes to allow an assessment of scatter for each. Then a sample of each, with a mean response, was selected and tested over the full frequency range with the modal hammer. The survey of all vanes and splitter vanes allowed detection of variations of frequency due to tolerance build-up or machining errors, by deviation from a normal distribution, with the possibility of correction by frequency tuning.

**ESPI TEST**

For each of the frequencies detected for the impeller backplate and vanes, the modal pattern was obtained by the technique of Electronic Speckle Pattern Interferometry (ESPI) using a RETRA 1000 from Conspec, Norway. This is a industrialized instrument that can be used anywhere, given a reasonably stable surface. The impeller was prepared by covering the relevant surfaces with retro-reflective tape for this part of the test. This caused some of the natural frequencies to be pulled down by as much 6%, compared to the impulse test results. This was not enough to get the different modes confused.

For this method the impeller had to be excited sinusoidally with a discrete frequency, which was achieved by attaching a piezo-electric crystal to the back of the particular feature under test and driving it from a digital frequency synthesizer.

On a VDU, the RETRA gave a real-time full-field picture of the vibrating impeller, with out-of-plane displacement fringe contours superimposed for the particular mode being excited. The brightest fringes, the zero order, represent lines of zero displacement, or node lines, with the contrast of the fringes decreasing with increasing deflection, giving an image that should be read like a contour map. The fringes do not indicate the relative phase of different anti-modes but this can be obtained by phase modulation of the reference beam. Also, by heterodyning the modulation unit in the reference beam path, the displacement fringe field can be animated, giving a clear picture of the mode characteristic.

The modal fringes were recorded by photographing the VDU image. The speckle patterns which are used as a carrier for the interference fringe information were attenuated by a speckle noise reduction technique. The photographs were made with multiple exposures, with the speckle field being changed for each exposure, by illuminating the impeller from a slightly different angle. The result was a smoothing out of the speckle field whilst retaining the fringe information.

Future development of the ESPI technique will involve digitization of the fringe field and use of software to improve fringe resolution and to quantify the displacements.

Plotting the natural frequencies of the impeller backplate modes, against the number of nodal diameters (Fig. 2), showed a steady progression of the first two sets of circumferential/diametral modes. Modes that had been missed, due to a weak response, were pin-pointed and found by increasing the power and applying the excitation to the anticipated position of an anti-node of that mode.

For the backplate frequencies with one central circumferential node, 24 diametral modes were identified (Fig. 3). With two circumferential nodes, 9 diametral mode frequencies were identified. (Fig. 4). For the full vanes, 22 natural frequencies were identified (Fig. 5) and a further 12 identified for the splitter vanes. Mixed backplate and vane modes were found with the ESPI method. (Fig. 6). All the natural frequencies were then corrected for engine operating conditions and plotted out on a Campbell diagram to highlight possible critical frequencies.
FINITE ELEMENT MODEL

In order to predict the natural frequencies and relevant mode shapes of the Hurricane impeller, 3D FEA was required. This was dictated by the back sweep of the impeller vanes which gave a 3-dimensional vane geometry.

The 3D FEA was performed using the MELISSA FE program for the solution and the SDRC I-DEAS pre- and post-processors for building the FE model and for examining the results.

The MELISSA vibration analysis program uses the inverse power method to calculate the natural frequencies and mode shapes of free undamped vibration. In addition, static condensation to reduce the mass and stiffness matrices is used as a standard technique. This requires the specification of master (dynamic) degrees of freedom in the model which are a subset of the total number of degrees of freedom. The master degrees are chosen so as to characterise the deformation of the structure in the various modes of interest. Then the program condenses down the system of equations to those associated with the master degrees of freedom using the Guyan reduction method.

The only material properties incorporated in this analysis were Young's modulus of elasticity, Poisson's ratio and the density. The material was assumed to be homogeneous, isotropic with linear elastic behaviour. Finally, only cylindrical polar co-ordinates were used.

Several FE models were created and tested in order to gain an understanding of the way that different mesh densities, element types, restraints and master degrees of freedom behave in terms of predicted natural frequencies and mode shapes.

Impulse frequency test and modal identification by ESPI were used to verify the 3D FEA model that would provide the necessary accuracy without exceeding the available computing capacity.

The full vane and splitter vane vibrations were estimated by modelling a segment one thirteenth of the impeller. The segment carried one vane and one splitter, comprised 1968 fifteen and twenty node wedge and brick elements, with 71 master degrees of freedom.

Coupled disc-vane frequencies could have been estimated by expanding the model to a full impeller. This would have made the model too large for current computing capacity.

Future work will include the prediction of the cross coupled vibrations using the segment model and the cyclic symmetry capability available in MELISSA.

For the disk frequencies and modes a coarse 3D FEA model of the entire impeller was found to provide reasonably accurate results.

This model consisted of 8164 nodes, each having three translatory degrees of freedom, and 4992 3-D solid, isoparametric, linear elements, 3835 of these were 8-noded brick elements and the rest were 6-noded wedge elements (Fig. 1). The linear elements provide less computational complexity but also generally less accuracy than the parabolic element. The model included 650 master degrees of freedom. Located on two radial-circumferential planes were 104 nodes restrained for motion in axial, radial and circumferential directions. Half the nodes were on the outer circumference of the front face of the impeller nose, the rest of the nodes were on the outer circumference of a plane sectioning the shafting at the rear of the impeller.

THICKNESS - FREQUENCY PARAMETER

The results of vibration tests and the finite element analysis showed a clear consistent relationship between frequency of disk mode vibration and the number of diametral nodes (Fig. 2). The test results included measurements from an impeller on which the disk tip thickness had been reduced in a series of cut-backs; the thickness ranged from 4.95 to 6mm. Frequency measurements were recorded at each change in thickness.

Examination of the results from all tests showed that the exponential relationship:

$$ f = A\cdot\exp(B\cdot n) \quad (1) $$

gave a correlation over the 8 to 16 diametral mode range. A typical value of correlation coefficient was 0.999. Fig. 2, shows a typical test result together with the frequencies as calculated by the 3D FEA.

Values of A were all within a range of 0.4%. B did not display the same consistency. However, in the test sequence on the series of cut-backs, a relationship to the variation in tip thickness was found.

$$ B = C\cdot t + D \quad (2) $$

The correlation coefficient being 0.982. The relationship is shown graphically in Fig. 7.

The two relationships are anticipated in the theory of vibration of circular plates (Southwell 1920). The existence of these relationships in a real disk type component are of value for the following reasons:

(a) The size and tolerance of the tip thickness of the impeller could be set with reference to one of the important design criteria, the avoidance of resonant frequencies,

(b) The performance of each impeller in service, for the case of disk mode vibration, be related to a single dimension rather than a range of frequency test results.
A SIMPLE TEST

The consequence of (a) and (b) is that quality control procedures may be simplified without endangering product integrity. It was initially anticipated that to ensure product integrity that each impeller would be tested by an impulse test and ESPI. The impulse test would indicate the true value of the frequency and ESPI the modal shape. The combination of the two tests would have been needed to determine, with confidence, the values of the fundamental and most damaging vibrations. The ESPI test was time consuming but the test for disk mode vibration can now be dispensed with. The relationships (1) and (2) give the link between frequency and mode.

REFERENCES


Fig. 1 Finite Element Model of Impeller.

Fig. 2 Frequency from Impulse Test and 3D FEA.

Fig. 3 Backplate mode at 2292Hz.
One circumferencial node.
Two diametral nodes.
Fig. 4 Backplate mode at 8714Hz.
Two circumferential nodes.
Six diametral nodes.

Fig. 5 Full vane at 820Hz.
First flexure

Fig. 6 Mixed vane & backplate mode at 6265Hz.

Fig. 7 Variation of Exponent B with Thickness

Tip thickness, t. mm
Exponent B

4.95 5.46 5.57 6