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THE ANALYSIS AND RESOLUTION OF A FATIGUE FAILURE OF A LOW PRESSURE TURBINE BLADE CAUSED BY THE EXCITATION OF A BLADED DISC MODE OF VIBRATION

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ABSTRACT

A high cycle fatigue failure of a low pressure turbine blade was investigated. Strain gauge tests of a running engine indicated a high dynamic response of the blade at the nozzle passing frequency. This could be attributed to the excitation of a bladed disc mode of vibration. A Finite Element analysis of the low pressure turbine blades and discs, together with bench testing of the complete structure, confirmed the existence of a high frequency 2nd Nodal Diameter mode of vibration. The levels of dynamic strain determined through strain gauge tests were found to be sufficient enough to explain the failure at the given location.

Having understood the problem, the situation was resolved through the use of Finite Element analysis with a short term modification to the original blade aerofoil to prevent the mode from being excited.

An aero/mechanical re-design of both the low pressure turbine rotor and the stator was undertaken to resolve the problem by both retuning the blade to avoid high frequency excitation, and also by reducing the forcing effect of the nozzle passing frequency. The new design has been validated through strain gauge tests and endurance tests. A further improvement in performance was also obtained.

INTRODUCTION

The Hurricane is the smallest gas turbine in the EGT range at 1.6 MW electrical power output. 4 units have been in commercial operation in combined heat applications. The lead engine has now run in excess of 17000 hrs.

The complete unit is entirely cantilever supported off the driven epicyclic gearbox, via struts across the air intake, all services to the turbine bearing pass down these struts. The rotor has a design speed of 27245 rpm and consists of a centrifugal compressor and a two stage axial turbine, mounted on a sleeve bearing upstream of the compressor, and a tilting pad journal bearing upstream of the turbine. The single stage titanium impeller gives a pressure ratio of 9:1, with a mass flow of 7.5 kg/s. Compressor exit air is fed into a radial diffuser and then an axial straightener to reduce swirl from the compressor. The 1st turbine stage features a cooled, segmented nozzle ring and an uncooled, shroud less, single crystal (CMSX4) rotor blade. The second stage turbine uses a one piece nozzle ring and shroud less, forged Nimonic 105 rotor blade. Overall thermal efficiency of the engine is 25%. The combustion system has a single, reverse flow combustion can, feeding into a turbine inlet duct. The combustion can material is Nimonic 75 with a ceramic coating and the turbine inlet duct is made from Haynes 230 and uses effusion cooling. A single dual fuel injector is used with air atomisation on liquid.

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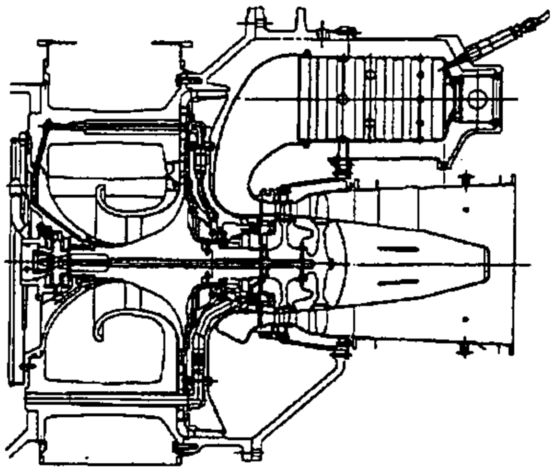

Figure 1 General Arrangement

ORIGINAL LP STAGE CONFIGURATION

The original Hurricane design incorporated 41 LP nozzles and 43 rotor blades.

Initial Blade Failure

A high cycle fatigue failure of the LP blade during development tests in Lincoln indicated a failure of the blade at the tip, 66% along the chord from the leading edge. The failure occurred in a low steady state stress region and the blade failure could be apportioned to high cycle fatigue by closer investigation of the crack propagation.

Strain Gauge Testing

The engine was strain gauged to investigate the above failure. Ceramic gauges were attached to the LP blade, a telemeter was attached to the LP disc and signals were recorded on tape.

The strain gauge test identified high strains which were considerably higher than EGT experience at high engine order frequencies. (See Figure 2). It was noted that the difference in blade and nozzle numbers (i.e. 41 and 43) could have excited a 2nd Nodal Diameter (2ND) mode of vibration (Reference 1) and this was the basis of the study outlined here. Analysis of the response traces indicated identical frequencies on different LP blades which seemed to confirm this belief. For other modes, blades responded at different frequencies to one another corresponding to blade alone or mistune bladed disc vibration. This variation in frequency is due to differences in blade to blade geometry, material properties, etc..

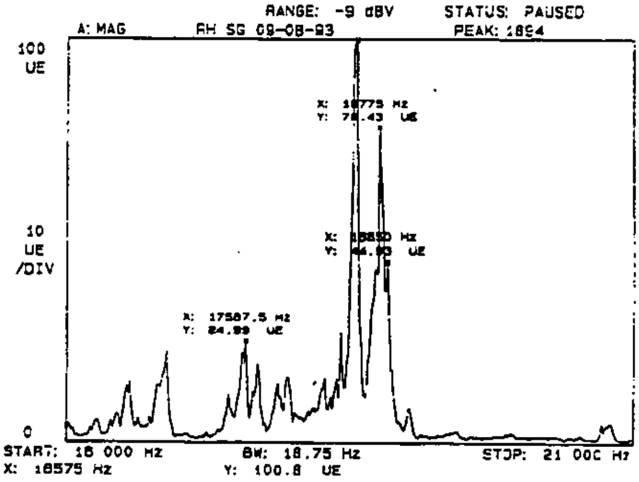

Figure 2 - Response at Nozzle Passing Frequency

Finite Element Analysis

A finite element analysis of the blade and disc was undertaken using a model of the blade and disc. Figure 3 shows the finite element model. A segment of the disc is modelled with the blade attached. Cyclic symmetry restraints were applied to both sides of the disc segment to predict the bladed disc nodal diameter frequencies.

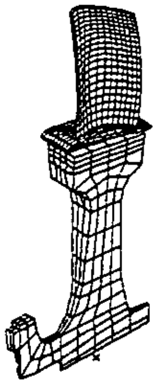

Figure 3 - Finite Element Model of Blade and Disc

Figure 4 shows the predicted frequencies from the analysis. This frequency plot shows frequency against nodal diameter. For low nodal diameters the bladed disc frequencies are dominated by the disc stiffness. As the nodal diameter increases the frequency approaches the blade alone frequency. The blade alone frequencies correlated well with bench test blade alone frequencies and so helped to validate the model used. It can be seen that many 2ND blade frequencies exist around the frequency of 18600 Hz and it was hard to determine which mode correlated to the one excited during engine tests.

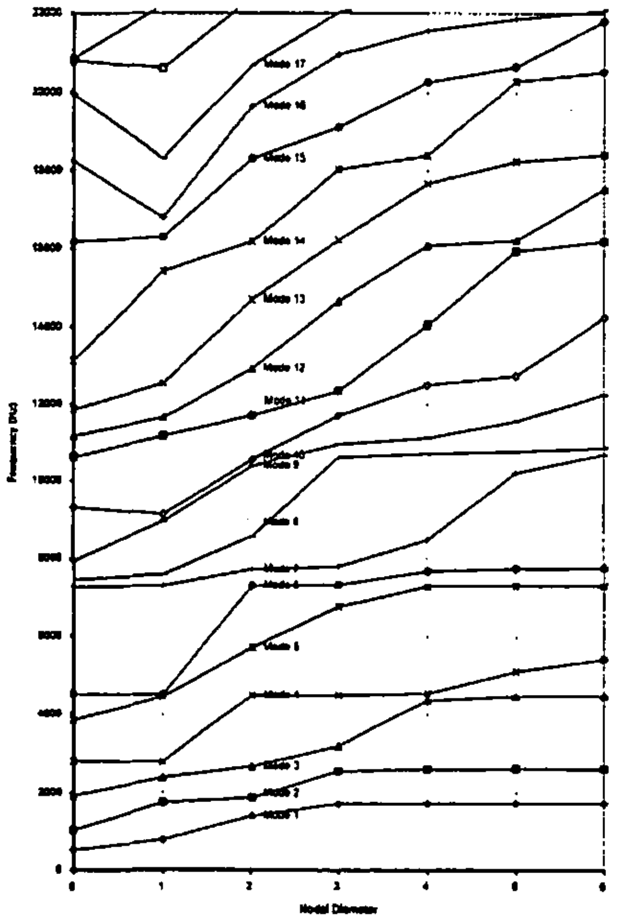

Figure 4 - Finite Element Model Frequencies

The modes shapes were expanded within the Finite Element Analysis package and 2ND modes 15 and 17 were identified with peak stresses around the failure location. .

Bench Testing

Bench tests on bladed disc assemblies are normally restricted as blade roots are 'loose' in their roots when no centrifugal loading is present. The Hurricane LP blade incorporated a single lobe root which locked up during running conditions and remained locked in position once shutdown had occurred.

Bench tests were used to validate the model using electronic speckle pattern interferometry. Low frequency bladed disc modes were identified (Figure 5 shows the first two 2ND modes), and compared favourably with FE results. A high mode was readily excited, and this frequency coincided (allowing for temperature correction) with the strain gauge frequency. The mode shape appeared to resemble a 2ND mode in general format, however definition was poor.

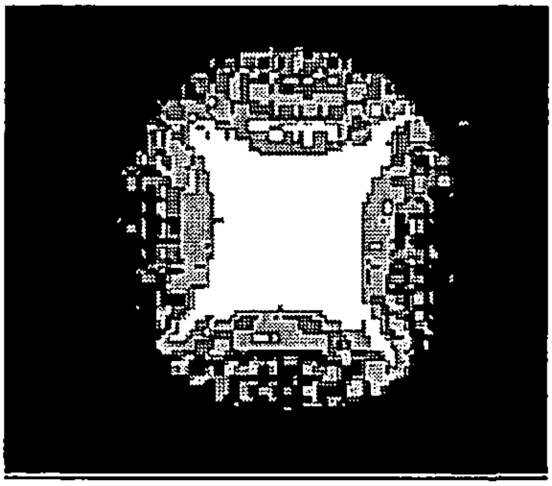

Figure 5(a) 2nd Nodal Diameter
1st Mode - 1216 Hz

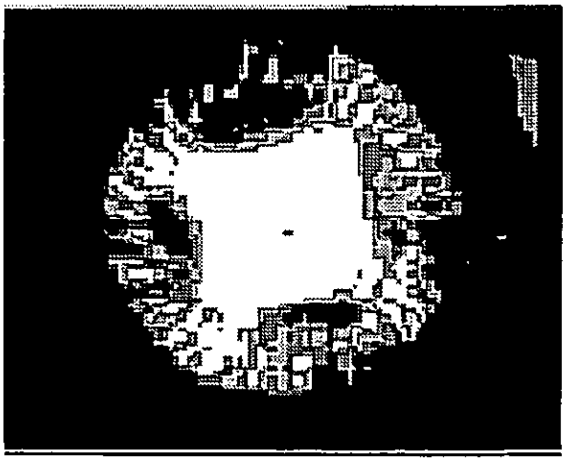

Figure 5(b) 2nd Nodal Diameter
2nd Mode - 1758 Hz

Figure 5 - Bench Test Results using Laser Holography

Blade Modification

As a short term measure the trailing edge of the aerofoil was shortened at the tip section. The blade was strain gauge tested and indicated that the vibration mode was no longer excited at the 41st engine order. A redesign of the blade was instigated to improve the performance of the blade. The results of the Finite element analysis were used to identify the number of nozzles (31) to be used in the subsequent design, so that the excitation of a 2nd Nodal Diameter mode could be avoided in future. Figure 6 shows that if the nozzle number is changed to 31 the 12th ND mode is excited which is on the flat part of the curve so that the bladed disc acts in a manner more akin to blade alone vibration.

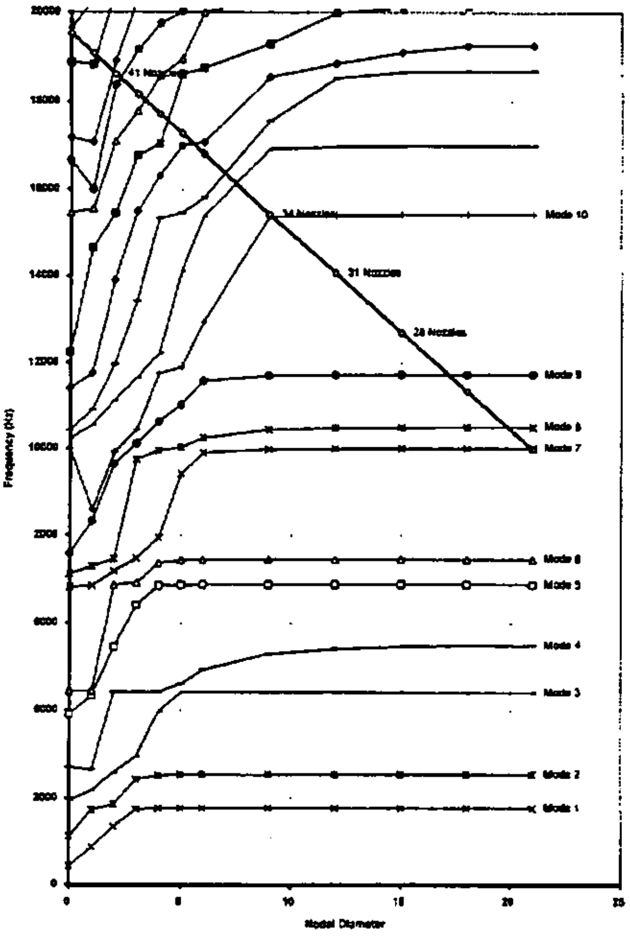

Figure 6 - Nozzle Number Selection

REDESIGNED BLADE

A re-engineering programme was established with a number of aims, one of which was to redesign the LP blade to recover power output by extending the trailing edge of the blade.

Mechanical Design

The mechanical redesign of the Hurricane LP turbine blade started with a series of preliminary aerofoil profiles stacked between an inner and outer gas path. The data was provided by an aerodynamisist working within the concurrent engineering team.

The task of the design engineer was to build a credible component around this geometry capable of withstanding the harsh environment, easily manufactured and within a specified budget. On top of this the engineer was restricted to delivering a design within an aggressive programme.

Design Features

The final solution included the following characteristics:

Optimised aerofoil stack - An aerofoils stack is the relative position of each aerodynamic profile in the axial and tangential direction. The combination of aerodynamic and centrifugal loads on the blade produce high stresses. These stresses can be reduced significantly by a technique of stack control called balancing. The Hurricane LP blade was balanced using FEA combined with a geometry optimiser. During blade stacking, profiles were allowed to move independently of each other in order to produce the optimum stress reduction. The final operation of a blade stacking job was to review the aerodynamic performance. Stacking is a change to the aerofoil and can have an adverse effect on performance. In some cases aerodynamic changes have to be made.

Thin platform - The platform forms the inner gas path wall. The leading edge was curved inwardly to produce a smooth gas entry to the aerofoil.

Short shank - In the interest of vibration response the shank was kept as short as possible. This provided a high first mode frequency.

Two lobe firtree fixing - A two lobe fixing was chosen over the original single lobe design to prevent lock up. Lock up of the blade in the disc is undesirable as it makes tip clearance control more difficult. Choosing the optimum number of lobes on a root fixing involves consideration of lobe loading, stress in the blade shank, available space in the disc, subsequent disc stressing criteria as well as manufacturing considerations. Fixing design is always a compromise. The fixing of the LP blade satisfied all criteria with the added bonus of satisfying the design requirements of the HP blade. Both stages of blade use the same firtree tooling.

Axial location tang and grooves - Axial location of the blade was achieved using a lip or tang at the leading edge of the firtree to prevent rearward movement and a peened strip located in grooves on the blade underside to prevent forward movement.

A model of the redesigned blade is shown in Figure 7.

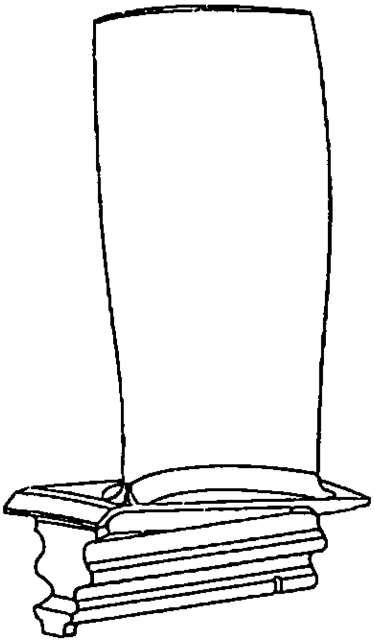

Figure 7 - Redesigned Blade

Blade Dynamics

Blade dynamic analysis was performed by using FEA methods to predict natural frequencies and mode shapes. A Finite Element model was generated direct from the aerodynamic profile of the blade using internal software. This enabled a quick turnaround of results so that the aerofoil could be optimised quickly with regard to both stress and vibration. FE correction factors were applied to the finite element results based on the results of earlier correlation between FE and engine test results to improve the accuracy of the prediction. A total of 12 iterations were undertaken within the design loop to achieve the required margins together with a reduction in overall blade stress. The typical time to produce frequency results from a retuned airfoil design was less than one day. Ansys 5.1 was used for this analysis.

The final blade design had a speed margin of over 7% from design running speed for the first blade mode (1st bending) from the 3rd and 4th. engine orders and the 9th mode (4th bending) from the 31st engine order (nozzle passing frequency). As will be explained later this relatively simple model produced frequency results that had close correlation to engine test data. The frequency results were used to produce a Campbell diagram for the blade

The Unigraphics part file was directly sent to the manufactures who machined the blade using a Leichti 5 axis milling machine. The blade geometry was checked by comparing measurements from a co-ordinate measuring machine directly to the blade tolerance limits.

Validation of the FEA results was achieved through experimental bench testing. A blade test fixture was designed to load the blade on the root flanks to simulate the engine centrifugal loading. This fixture had an integrated load cell to measure the thrust loads applied to the base of the blade. A series of repeatability studies on the fixture showed that for the first mode of vibration the fixture repeatability was better than 1%. Frequency measurements were made by exciting the blade with a miniature force hammer and measuring the blade velocity using a VPI sensor (Laser Doppler Vibrometer). Frequency measurements up to the tenth mode of vibration were carried out on a large sample of blades to obtain the blade to blade scatter. Mode shape measurements for the first ten modes of vibration were made by scanning the laser over the blade surface. The blade frequencies and mode shapes were compared to the results from the FEA. The correlation was excellent for the mode shapes and the frequencies, within 5 % for the first ten modes of vibration. Figure 8 shows the blade Campbell Diagram.

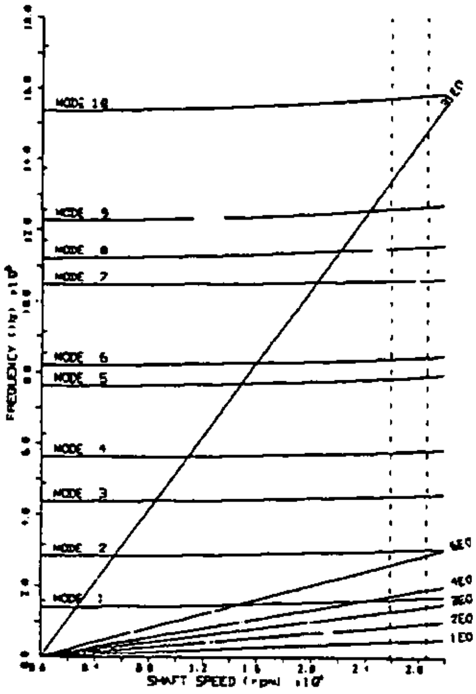

Figure 8 - Campbell Diagram

Optimum positions for blade strain gauges were determined from the FEA results, two positions were chosen in order to capture all of the first ten modes. Strain gauge factors for each of the gauge positions were determined from the FEA. A number of blades were selected for gauging which had maximum and minimum bench test frequencies for modes 1 and 9. Strain gauges were applied to these blades and covered with a ceramic coating to protect the gauges from the high temperatures. A telemetry transmitter was housed on the LP disc and used to send the signals to analogue tape storage. An engine test was carried out which contained speed sweeps at various load conditions. Strain gauge data was analysed by a Cranfield Data Systems analysis package, which provided multi channel data analysis capabilities at high sample rates and with high frequency resolution.

OISCUSSION

The strain gauge frequencies compared favourably with the temperature factored FE frequencies.

The Campbell Diagram of the redesigned blade (Figure 8) showed similar frequency distribution to the frequency distribution identified in the bladed disc analysis at high nodal diameters (Figure 6) despite significant change to the design (e.g. twin lobe root and modified aerofoil). This was because modes were not significantly affected by root modification (i.e. the modes of vibration were aerofoil dominated), the frequency separation between the 9th and 10th mode was maintained throughout the redesign process and, although changes in aerofoil section location and orientation were made, minimal changes were made to the aerodynamic profile.

The data showed acceptable speed margins from the 3rd, 4th and 31st engine orders. The highest strains were due to random excitation of the 1st mode. No significant vibration levels were identified at higher frequencies.

The maximum strain value (only the fundamental frequency showed any significant response) was factored to the highest stress location and an acceptable blade life determined from a Goodman diagram.

The design has improved performance and mechanical integrity from the original design. Modifications were made in other components of the engine (not outlined in this report) to further enhance the design.

CONCLUSIONS

1. This report highlighted how a low nodal diameter mode was identified during strain gauge tests by analysis of the signal and correlation of bench test data with Finite Element analysis results.

2. The above failure condition was designed out to provide an engine with significantly improved performance and improved life.

REFERENCES

1. D.J. Ewins "Structural Dynamic Characteristics of Bladed Assemblies", AGARD Manual on Aeroelasticity in Axial-Flow Turbomachines - Volume 2, Structural Dynamics and Aeroelasticity, US Dept of Com, NTIS, (1988).