

SYNTHESIS REPORT

FOR PUBLICATION

CONTRACT No: BRE2-CT92-0222

PROJECT No: - 5463 (ROSTADYN)

TITLE: Modelling of Rotor/Stator Interaction Dynamics

PROJECT

CO-ORDINATOR: Rolls-Royce

PARTNERS:

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STARTING DATE: 1 / 6 / 93

DURATION: 45 MONTHS



PROJECT FUNDED BY THE EUROPEAN
COMMISSION UNDER THE BRIT/ EURAM
PROGRAMME

30/4/97

Modelling of Rotor/Stator Interaction Dynamics (ROSTADYN)

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Abstract

This paper presents the findings of a research programme which examined several different but related aspects of vibration and stability in the rotors and stators of high-speed rotating machinery. The project was primarily concerned with the detrimental effects that would result from attempts to reduce the rotor-stator radial clearance (which would be desirable in the interests of efficiency), including the initiation of coupled rub- and fluid-induced instabilities of the rotor (type a: rigid cross section) and other flexible rotor or stator components (type b: non-rigid cross sections).

The objectives of the research were to develop end-user predictive design methods which had been validated by tests on laboratory rigs and evidence from industrial cases. These objectives were achieved.

In the course of this work the following instabilities were demonstrated on Laboratory rigs:

- . Rotor reverse whirl instability due to stator rubs (a)
- Rotor instability due to internal hysteresis (a)
- Rotor and casing coupled instability due to wave-speed coincidence (b)
- . Rotor or stator seal element instability due to fluid coupling (b)

1. INTRODUCTION

This work was concerned with the mechanical interactions of high-speed rotors with stationary casings and seal lands. It applies in principle to a wide range of fluid/ thermodynamic machines or to any machines containing a high-speed rotor, e.g. steam and gas turbomachinery, turbopumps, turbocompressors, electrical alternators.

The efficiency of many high-performance rotating machines is strongly dependent on the running clearances between the rotor and its casing. While there are very large economic benefits associated with reduced clearance, (particularly to the electrical utilities), there are potentially significant risks for durability, mechanical integrity and even personal safety.

The results of this programme are applicable to a wide range of rotating machines. **Many** structural instabilities and operating failures in rotating machines are not properly understood. The intention of the programme was to demonstrate and verify proposed rotor/ stator interaction mechanisms and to provide predictive design methods and criteria so that smaller running clearances can be used in future high-performance machines without risk of mechanical failure.

Instabilities of either the whole rotor system or the immediately-adjacent dynamically-flexible components may occur due to coupling through the working fluid or physical running contact. The instabilities are expected to be characterised by forward and/or reverse rotor whirl and near-coincidence of forward and rearward traveling waves.

There are currently no dynamic stability criteria for some types of component which run with close clearances and where there are periodic in-service problems which require redesign, usually adding weight and manufacturing complexity (cost). Similarly, there are other potential rotor instability problems where stiffness (weight) is added to the structure on the basis of simple but invalidated linearised predictive models, leading to conservative designs. This approach is no longer acceptable to manufacturers due to the high level of competition and the pressure to improve performance and minimise cost and weight. The project focused on the ability to make an accurate description of the different forms of dynamic interactions which occur between the rotating and the adjacent stationary components in these machines, many of which can cause damaging vibrations.

The 4 industrial partners in the project agreed on 5 related areas of technical need and, as a result, the objectives for these 5 different subjects were defined as follows:

Specific Objectives

The project sought to bring together a "number of related rotor/ stator interaction phenomena which, although different from each other in detail, share a number of common features and which are amenable' to a common design approach. The specific objectives were to provide enhanced design tools, as follows:

Objective 1 Structural Internal Rotor Damping

Analysis of validated structural dynamics models of rotating components, including realistic representation of inherent damping in rotating machines.

To develop and validate a structural damping model and establish the role of internal-/external damping ratio and cross coupling on machine

stability as a basis for predicting the destabilizing effects of rotor/ casing dynamic interactions.

Objective 2 *Rotor Stability, Fluid Clearance Forces*

Prediction method for rotor dynamics instability caused by fluid coupling.

To develop a simulation model and a corresponding computer routine for the prediction of rotor dynamic force coefficients for gas labyrinth seals and other seals (when the two elements are moving as rigid surfaces relative to each other).

Objective 3 *Labyrinth Seal Stability*

Prediction method for occurrence of unstable vibrations in labyrinth seals.

To develop a simulation model and a corresponding computer routine for the prediction of the stability behaviour of vibrating stator/ rotor seals with a turbulent gas flow in between.

Objective 4 *Rotor Whirl Characteristics due to Light Casing Rubs*

Prediction method for rub-induced unstable whirl and vibration of rotors.

To demonstrate and determine the conditions under which severe rotor vibrations occur due to rubs at blade tips, seals and bearings (hydrodynamics).

To develop a predictive analytical model applicable to partner machinery and validate this by experiments on dynamically representative models.

Objective 5 *Flexible Bladed Disc-Casing Severe Interaction*

Prediction method to avoid unstable coupled vibrations of flexible rotors and stators.

To demonstrate the phenomenon and verify the mechanism, establishing the controlling parameters criteria and a validated analytical model which can be applied to the design assessment of lightweight high speed machinery.

The work was carried out by the industrial partners and 4 universities and is reported below under the above headings.

2. TECHNICAL DESCRIPTION

2.1 Structural Internal Rotor Damping . . .

History is sprinkled with cases of unexplained vibrations and excessive deflection of rotors in high-speed machines. It is believed that some of these are responsible for unexplained catastrophic power plant failures:

It would be tedious to list all cases reported in the literature, but the fact is that publications relative to internal damping can be found dating back to the 1920s, driven by unexplained catastrophic failures. This and subsequent work showed the possibility of internal-friction induced instability.

Attention was rapidly paid to spline couplings, because the friction in those components was identified early on as a potential source of unstable behaviour. The generalisation to all kinds of damping came in the 1970s, with Nelson's publication (ASME 1976).

Today, the state-of-the-art seems somewhat paradoxical: although internal damping in rotordynamics is commonly underlined as being a potentially-destabilizing effect, little is recommended to avoid troubles. Studies of spline coupling damping effects were re-activated in the USA in the late 1980s due to an unstable behaviour in the space-shuttle engines but, since then, little attention has been directed to theoretical investigation.

The existing theoretical stability analysis, including only viscous damping effects, predicts that as soon as there is internal damping, an absolute limit of stability appears, depending on the first bending mode frequency of the rotor. This analysis neither predicts what kind of motion might happen after this limit has been crossed, nor what the effect of a more realistic damping model would cause on that prediction.

The participation of rotating terms, of second-order at low speed, becomes predominant in the stability map at high speeds. The situation can be worse in the supercritical range because, in that case, the equilibrium of the rotor is such that the damping reaction forces can add energy to the rotor, destabilizing the rotating motion.

All these trends are identified in published literature, but no synthesis of the various aspects has been achieved up to now. Even the well-known Professor Childs does not go beyond simplified models in his most recent book (1). So the state-of-the-art on this specific rotordynamics task is that fragments of knowledge do exist, but there is no effort to pull it all together in order to identify the criticality of internal damping in a general overview.

The intention of the work covered the following three aspects:

- to quantify the amount of internal damping available in a turbomachinery rotor, and to identify the main sources of friction,
- to create a tool capable of simulating the onset speed of instability of real rotors,
- to find guidelines for engineers in order to avoid the unstable behaviour, or to increase the onset speed of instability.

2.2 Rotor Stability, Fluid-Clearance Forces

The influence of large labyrinth seals on the rotor dynamic behaviour of rotating machinery can be expressed by dynamic coefficients. Different methods are applied in this project in order to calculate the dynamic coefficients.

One method is the bulk flow method, on which base a three volume model was applied. The main idea of this model is to describe the flow in a labyrinth seal and to split the sealing region into three typical control volumes, I to III. The control volumes I and II represent the jet flow beneath the cavity and the seal fin, respectively, while control volume III accounts for the vortex flow within the cavity.

For each of these volumes the momentum, continuity and energy equations are established.

A more accurate way to calculate the dynamic coefficients is to use the finite difference method, where two- and three-dimensional versions are applied. This method describes the compressible, turbulent time dependent and three-dimensional flow in a labyrinth seal by the Navier-Stokes equations in conjunction with a turbulence model. Additionally, equations for mass and energy conservation and an equation of state are required. A perturbation analysis is performed yielding zero- and first-order equations. For the first-order solution, assumptions are made for the behaviour of the fluid in a circumferential direction.

The two-dimensional method has the restriction that it is only valid for look-through Labyrinths. For the three-dimensional method no assumptions for the circumferential direction are made. The first-order variables are calculated at each grid point of a three-dimensional mesh. The time derivatives are removed by introducing a rotating co-ordinate system,

Thomas/ Alford Forces arise when a bladed rotor is deflected from the central position leading to variable clearances around the circumference, In this case, the fluid flow through the bladed area varies with the varying clearance. Consequently, the loss in the clearance increases and the tangential force at this position decreases. The resulting force of the two opposite forces acts perpendicular to the direction of deflection. For small deflections this can be expressed as cross-coupling stiffness coefficients.

The importance of these forces and their prediction was reviewed in this project,

2.3 Labyrinth Seal Stability

Labyrinth seals are commonly used in rotating machinery, but in lightweight aero gas turbines instabilities may arise in either” the rotating or stationary members, leading to high-cycle fatigue failures. Most of the published analysis originated over two decades ago and even the most recent analysis (3) is not given in an explicit fashion or even validated by experiment. The approach taken in this project was to develop an analytical stability model and to develop an experimental rig to validate the predictions.

2.3.1 Theoretical Model and Software :

The gas is described by the momentum. equations in axial and circumferential directions, the continuity equation and the energy equation. A perturbation analysis is performed yielding the dynamic pressure forces due to vibration motion of the structure. The structure is modelled by means of the finite element technique using NASTRAN in order to be convenient for the user and for the availability of a powerful element library. In order to solve the eigenvalue problem, the fluid routine is coupled with NASTRAN. Stability is estimated by examining the real part of the eigenvalue solution. The main fluid routine is easy to handle and provides excellent convergence behaviour for all boundary conditions. The routine is validated using the results from the test rig.

2.3.2 Labyrinth Seal Test Rig

An experimental rig was developed. This has the ability to vary the flow, pressure drop and inlet swirl to the seal as well as being able to test seals of different structural and acoustic natural frequencies and seal clearance. It has the capability to arrange the test seals so that they can be either upstream-supported or downstream-supported. The rig is non-rotating to minimise the cost and ease of the experiment without compromising the legitimacy of the validation.

The rig is shown below in Figure 1.

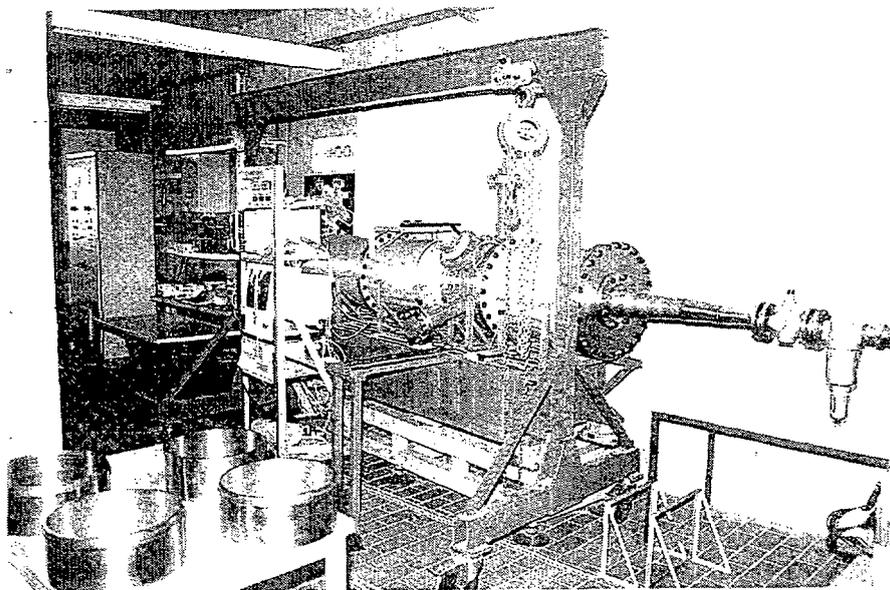


Figure 1 - Labyrinth Seal Test Rig

2.4 Rotor Whirl Characteristics due to Light Casing Rubs

General Introduction

Contact between a high-speed rotor and a stationary second body can arise in different types of machines. Reduction of the clearance between the rotor and the stator may lead to contact, rubbing or impacts, with severe implications for mechanical integrity. Instabilities may occur, either of the whole rotor casing system or of the dynamically-flexible components.

The broad aims of the work were to investigate by theory and experiment the interactions which occur when turbomachinery rotors make contact with stators, in the following industry applications:

- the general contact problem and studies of rubbing rotors supported by magnetic bearings,
- studies of reverse whirl in simulated Gas Turbine rotors supported by rolling element bearings, and
- studies of simulated Steam Turbine rotors supported in Hydrodynamic bearings.

Contact also occurs in systems where rotors are supported by active magnetic bearings, when the magnetic bearing fails. The rotor must then make contact with a

mechanical retainer bearing (4) which, obviously, has to be designed in such a way that it can withstand the dynamic loads.

The rotor dynamics literature on rotor/ stator contact interaction shows a wealth of models and rotor behaviour. The result of such a rub contact can be a reverse whirl of the rotor, as well as synchronous, subsynchronous and chaotic motions (5), (6), (7), or spirally increasing bending vibrations caused by rub induced hot spots on the rotor. An extensive literature survey is available, for example, in the review paper of Muszynska (9).

2.4.1 Experimental Rig

For each of the cases experimental rigs were produced and instrumented with configurations to represent the 3 industrial applications i.e. two bearing rigs for the first two applications and multiple (7) bearings for the latter.

For the magnetic bearing investigation a specific aspect was to investigate the rotor contact forces and duration occurring when a magnetic bearing failed and contact was made with the catcher bearing at speed up to 30000 rev/ min. Different catcher ring materials were investigated. The rotor simulating an aero gas turbine rotated at high speed and had the ability to excite forward or backward whirl in the rotor prior to making contact with a 'rigid' or 'more flexible' stator. Rotor/stator contact was initiated by increasing artificial excitation of the rotor at speed, until a rub occurred on a stator housing located mid-way between the bearings. A reduced scale steam turbine/generator rotor rig was constructed with a representative arrangement able to make contact with the rotor at its scaled running speed.

2.4.2 Theoretical Modelling

Analysis was carried out to model the theoretical contact conditions which were to be validated by the experimental work, for the magnetic bearing configuration. These results could then be utilised for all three applications.

Separate analyses were carried out using different analytical methods and approaches for each of the three cases (2).

2.5 Flexible Bladed Disc-Casing Severe Interaction

A United States air accident investigation report suggested that a particular aircraft engine failure in 1975 could have been caused by dynamic interaction between the fan blades and the engine casing, expelling a fan blade and causing severe damage to the airframe.

The vibration properties of axisymmetric flexible structures such as discs and rings may be described in terms of forward and backward traveling waves. If the mode shape and speed of a backward traveling wave 'on the rotor coincides with that of a forward traveling wave on the stator, and local contact then occurs, energy may be transferred from the rotor into the stator. Engine rotors have a high level of kinetic energy and so very high forces may be generated between the rotor and stator if this event occurs.

The intention of this part of the project was to investigate a possible travelling-wave-coincidence instability which could occur between an elastic rotor and an elastic stator in rotating machinery such as gas or steam turbines.

This task therefore proposed to build a demonstration test rig to simulate the possible interaction between a flexible rotor and a flexible symmetric stator, and to assess the instability mechanism by both an analytical and experimental investigation.

A test rig was designed and built comprising a test rotor of a thin flexible steel disc, rotated 'at controlled speed above a circular stator carrying 20 flexible flat steel blades parallel to the axis and tuned to the same frequency. This arrangement allows relatively low-speed testing, and minimum windage problems. The rotor can be run at a selected speed and the stator clearance adjusted to a low level before contact is initiated.

A numerical integration method was developed in parallel with the experimental work to predict the rotor and stator response due to interaction between blade tips and the rotor, including a contact model to determine the forces at the interactions.

3. RESULTS

3.1 Structural Internal Rotor Damping

Measurement of the internal damping was made on several specimen rotors. These were designed to allow the identification of the damping created by conventional rotor interfaces : shrink fits, bolted flanges, tie-bolts alone and curvic-couplings. A dedicated test-bed allowed the measurement in both rotating and non-rotating conditions, and the gyroscopic effect given by the rotor discs allowed the separation of the damping in forward and backward modes. The quantification of the amount of damping available in each joint type has been achieved, together with the amplitude-dependent non-linearity and the evolution of the apparent damping when rotating.

It was the purpose of the simulation phase to use a finite element modelling in order to simulate the experimental results. The finite element used has a classical beam formulation together with an expression for the internal damping in the rotating reference frame. This internal damping allows a viscous part and a hysteretic part to be included. A theoretical model linked the physical source of damping - the friction - to the coefficients necessary for a numerical simulation. Then the comparison of test-rig experiments and of one industrial case provided a demonstration of both qualitative and quantitative agreement with reality.

An attempt to find guidelines for engineers was made using a theoretical approach, based on a Fourier expansion of a set of differential equations representing a Kelvin-Voigt shaft stiffened with discs. With a method of linearisation, it was then possible to analyse the coefficients given by the Routh-Hurwitz criterion, and to identify trends useful for engineers. Because of the complication of the equations of motion, it was then necessary to make numerical simulations on simplified geometries which located the "onset speed of instability" of the rotors, together with the behaviour above this speed when the Hopf bifurcation has occurred. It was found that the

operation at unstable speed gives a limit cycle at such a high amplitude that it is of no use in a real engine. An “optimal value of internal damping” was also found, for which the onset speed of instability reaches a maximum value.

3.2 Rotor Stability, Fluid-Clearance Forces

Three different routines for the calculation of the dynamic coefficients are now available. The routines follow the theory described in the previous section and are validated with measured data provided by previous investigations.

With these routines, calculations of the dynamic coefficients for look-through seals of the industrial partners were performed. One partner provided four seals with different geometries and different flow conditions. The dynamic coefficients were calculated for more than fifty different operation points. The coefficients are used to complete finite element models of the engines. For another partner, the dynamic coefficients for seven different gas turbine seals were calculated. It is expected that in this case the influence on the overall behaviour is only weak, since the magnitude of results for these gas turbines seals is only small.

A finite element model of a complete steam turbine was built for another industrial partner who delivered the dynamic coefficients for the seven bearings of this machine. The dynamic coefficients of one seal of this machine, the balance piston, were calculated. The analysis of the engine with all bearings and this seal showed critical behaviour at about 35 Hz.

3.3 Labyrinth Seal Stability

3.3.1 Theoretical Results ‘

(a) Structure Modal Results

For the stability analysis of the rig structures, finite elements were built using shell elements. The comparison of the measured and the calculated natural frequencies of the modes for the different rotor and stator versions showed good agreement. While fitting the calculated to the measured values, it was noticed that modes with more than four nodal diameters can be adjusted by changing the Young’s modulus and those with a low number of nodal diameters by changing the radius where the structure is fixed-to-the- housing. The deviation of the calculated values to the measured values is less than five percent.

(b) Structure Stability Results

In order to produce diagrams comparable to Abbott’s (3) diagrams, calculations were performed where the natural frequency of the structure is changed by changing the Young’s modulus of the material. The acoustical frequency of the interfin cavity remains constant. Both the damping provided by the fluid and the modal damping of the complete fluid-structure system are calculated for different pressure drops while the entrance pressure is six bar. The overall results show good agreement with Abbott’s diagrams. For very small pressure drops the complete system is stable, independent of the structure, whether upstream or downstream supported, or stiff or soft compared to the acoustical interfin natural frequency of the cavity. This

indicates that stability is not only a question of geometrical conditions but also of fluid boundary conditions.

Downstream support effects

When the structure is soft (low natural frequency) compared to the acoustical natural frequency of the interfin cavity, the fluid provides negative damping values. This means that the fluid is destabilizing on the complete system. For small pressure drops of about 0.1 bar the damping values are slightly negative, but for higher pressure drops, the damping becomes strongly negative.

The sign of the damping turns to positive values when the natural frequency of the structure is close to that of the acoustic frequency. The gas is then stabilizing. When the structure frequency becomes higher than the acoustic frequency, the damping has a maximum and for very stiff structures the damping decreases.

Upstream support effects

By changing the support side the behaviour of the structure changes completely. For soft structures, the fluid damping is positive, When the structure is stiffer than the acoustic frequency, the damping becomes negative and has a minimum. There is a common point for all pressure drops, where the sign changes from positive to negative values. This point does not move for higher pressure drops as is the case for downstream support.

3.3.2 Experimental Results

The response of the seal members for different flow conditions were measured and the frequency spectra analyses.

The stiff labyrinth seal structures are stable for a low pressure side support. The occurrence of very small excessive amplitudes can be explained by the resonance phenomenon in the fluid. The high pressure side supported stiff versions are unstable for certain working conditions; the stator gets in contact with the rotor. This is in line with the present design criteria.

For the soft (low natural frequency) rotor and stator versions both supported sides result in unstable behaviour. In the case of the soft low pressure side supported seal the 3 nodal diameter gets unstable at a supply pressure of 8 bar with a pressure drop of about 0.02 bar. For the high-pressure side supported soft seal the 8 nodal diameter becomes unstable at a supply pressure of 8 bar with a pressure drop of about 1.5 bar. The high pressure side supported configuration is more stable than the low pressure side. So there is a need for new design criteria, because the present method failed in this case.

If we compare the high pressure side supported soft with the stiff (high natural frequency) labyrinth seal configuration then stability is the reason for the high amplitudes occurring in both cases. The safe operation range of the soft structures is however larger than the stiff variant.

With the experimental results, present design criteria can be evaluated and its limits can be shown. Phenomena are observed that couldn't be related to the present criteria, Before there is a real aeroelastic instability problem the structural amplitudes are caused by forced response.

Some very interesting results have been generated from the large amount of data acquired with the different seal types, support conditions and fluid pressures.

3.4 Rotor Whirl Characteristics due to Light Casing Rubs

3.4.1 Experimental Results

The measurements of contact force investigated on the magnetic bearing rig provided some interesting results.

The impact coefficient of restitution, ε , can be derived by comparing the velocities of ring and rotor before and after impact, and this leads to values between $\varepsilon = 0.3$ and 0.8, depending on the material of the contact ring.

The whirl motions measured on the magnetically-supported rotor could be considered in 3 separate phases as follows:

Phase 1- Initial contact of the rotor and casing,

Phase 2- Whirl motion itself,

Phase 3- Kinematic rolling condition.

Violent vibrations were demonstrated to occur, with associated rapid deceleration of the rotor. Different characteristics were obtained with different catcher bearing materials.

Similarly, for the aero engine type rig, violent vibrations could be demonstrated which were similar in principle to those on the magnetic bearing rig, considering that in this case it was a flexible rotor.

It was expected that the initiation of contact by vibrating the rotor would yield quite different results according to the initial rotor motion. In fact, it was found that many of the *a priori* assumptions were incorrect. For example, except for a very short initial period, the outcome is independent of the vibration which precedes contact, and also the outcome at higher rotor speeds was not found to be more dramatic.

For the steam turbine rotor, provision had been made in designing this six-bearing rig to bring static components into contact in the plane of the last stator stage at the generator end of the LP turbine rotor. Light periodic contact applied using a bolt which was screwed in horizontally (the most probable point of contact) had very little influence on any of the bearings. Therefore much heavier rubs were imposed in all of the tests so that in each, contact with the rotor was maintained over more than half of the circumference. Even so, the changes in total amplitude were not large, whether the rubs were horizontal as described, or vertical (on top of the rotor) or both. All of the tests were carried out at a fixed high speed. It is believed that significant damping comes from the 6 hydrodynamic bearings.

3.4.2 Analytical Results

It was found from tests on the magnetic catcher bearings that different ring materials exhibited quantitatively different behaviour but, supported by theoretical work, important qualitative similarities were demonstrated and mathematical representations of the contact forces which are immediately applicable in rotor

dynamics codes were devised. The resulting equations of motion are, of course, non-linear and require some effort and care when making predictions.

Simulation of reverse whirl in the aero-engine rig used a Finite Element Model. This model was constructed using NASTRAN elements, which represented the basic configuration of the test rig. The vibration modes were calculated and shown to correspond with rig measurements. A non-linear transient dynamics analysis was then conducted using the same FE model but with the inclusion of additional features such as damping, excitation, snubbing rings, point dampers, etc. Using these features the experimental test conditions could be simulated directly.

Analytical models generated for the steam turbine configuration were in quantitative agreement with the observed motions.

3.5 Flexible Bladed Disc-Casing, Severe Interaction

The test rig performed successfully. A slight impact to the disc causes initial rotor/ stator contact, and if the rotor and stator wave speeds are closely matched, the interaction amplitude will increase rapidly, controlled only by limit stops placed on the rotor and stator. However, at non-critical speeds the impacts decay away. Safety was a major consideration of the design of the test rig, as the forces acting on the rotor are considerable during critical speed operation.

The theoretical time-marching predictions show the relative motion of the rotor and stator after an applied initial disturbance caused contact. The model results describe the increasing vibration amplitudes close to the critical speed, and has been used to investigate the effect of damping and mistuning on the stability of the system.

The model confirms that reduced symmetry in the stator reduces the strength of the traveling waves, and hence the risk of instability. Similarly, increasing the damping in the system also reduces the risk of instability.

4. CONCLUSIONS

4.1 Structural Internal Rotor Damping

- The measurements conducted on several 'representative test rotors under rotating and non-rotating conditions established the quantity of internal damping present in the various test rotors, (2). The procedure developed identified instability of the rotor without the test arrangement itself becoming unstable, and this measurement technique is believed to have wider application.
- The theoretical work identified the existence of an 'optimum value' of internal damping and a better understanding of the nature of the instability.
- By modelling the friction damping in the rotor using a dedicated finite element, it was possible to simulate accurately the damping characteristics observed in the test specimens.

4.2 Rotor Stability, Fluid-Clearance Forces

- Software was prepared to predict the 'linear dynamic rotor forces generated in labyrinth seals and bearings, from which dynamic coefficients can be derived.

- The fluid forces for gas turbine rotors are relatively small. However, on steam turbines, due to the high pressures and long seals especially the balance piston the forces may have a strong influence on rotor behaviour.

4.3 Labyrinth Seal Stability

- The predictive software was produced and validated by the experimental rig test and industrial gas turbine seal evidence.
- A dimensionless parametric study was conducted which gave some design pointers and criteria for preliminary stability assessment. It should be remembered that actual stability is dependent on the inherent structural damping of the seal element as well as the aerodynamic effects studied here.
- The rig -tests showed the importance of some of the major controlling parameters e.g. acoustic frequency relative to structural frequency and support side of the seal. The evidence generally supported the analysis of Abbott (3).

4.4 Rotor Whirl Characteristics due to Light Casing Rubs

- Contact between a rotor and a stator, as encountered in turbomachinery and in magnetic bearing suspensions, can lead to violent vibrations.
- Rub-induced reverse whirl has been demonstrated and characterised on a magnetic bearing rig representative of aero-engine rotor dynamic conditions.
- Theoretical methods have been developed which are in qualitative agreement with the test results, but which highly overestimate the severity of the response at each condition. This discrepancy is considered to be due to unaccounted non-linear loss mechanisms in the contact behaviour and in the stator supports.
- The measurements of the contact interaction between a steel rotor and its stator ring made of graphite show that, during the whirl motion, the whirl velocity increases until it locks onto the first elastic natural frequency of the rotor, rigidly supported at both ends, and that it usually increases only slightly from then on before, finally, it breaks down as a consequence of the energy dissipation.
- Light contact in steam turbine rotor/ stator systems is likely to be benign in all cases in which the stability limit is not close to rated speed.

4.5 Flexible Bladed Disc-Casing, Severe Interaction

- The theoretical analysis and rig demonstration show clearly that the hypothesised phenomenon does exist and should be considered at the design stage when assessing critical speeds, and reducing clearances in new engine designs.
- The analytical model is validated quantitatively by the experimental results.
- Stability can be increased by imposed non-axisymmetry on the stator, and - to some degree - by adding damping to the system.

5. ACKNOWLEDGEMENTS

The following acknowledgements are gratefully recognised:

- The authors wish to make it clear that the work reported here was carried out by many researchers at the participating universities and industrial organisations.
- To the European Commission for the funding of the ROSTADYN project (BRE2-CT92-0222, BE 5463) and in particular to M. Andrieu, P. Kruppa and J-P. Berhault (EURESYS) for their advice, help and guidance.
- To the Swiss Bundesamt für Bildung und Wissenschaft for financing the Swiss contribution to the project, and the Brazilian National Technology Council CNPq for supporting their team member Marco Fumagalli.
- . To the Swedish Government for their contribution to work at Volvo A.C.

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