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The Dynamic Characteristics of a High Pressure Turbine Stage in a Transient Wind Tunnel

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ABSTRACT

A new transient facility for the study of time averaged and unsteady aerodynamics and heat transfer in a high pressure turbine was recently commissioned. During the facility design a high priority was placed on ease of access to the turbine blading to facilitate the development of blade mounted instrumentation. The turbine disc was cantilevered on a shaft by a thin annular link, with the shaft passing back through the disc to a single row and matched pair of relatively closely spaced bearings. The bearings were originally designed for use in a marine gas turbine. Due to the facility's novel mode of operation the bearings were working well beyond some of their original design limits, primarily due to the high turbine acceleration and the high speed/low load condition at the end of facility operation.

During the facility design a programme of work was undertaken to predict its dynamic performance. This was continued during commissioning to measure actual facility performance. In this paper the predicted dynamic performance of the disc/shaft assembly and its bearing system are presented. The programme of work undertaken during the facility commissioning phase to ascertain actual dynamic performance is described, and the results discussed. Finally, the technique developed to field balance the rotating assembly following changes to blade instrumentation is described.

1 INTRODUCTION

The facility described in this paper was conceived originally as a low cost alternative to the conventional full scale testing of engine hardware. This allows the fluid flow phenomena associated with the aerodynamics of, and heat transfer to, turbine blading to be studied.

Conventional turbine test facilities have, in many cases, been built to operate continuously at full–scale engine operating conditions of mass flow, temperature and pressure. The problems associated with extracting detailed measurements from such facilities become steadily greater as the turbine inlet temperature is increased, resulting in their role becoming one of design validation. To make the detailed measurements required for the derivation, development and validation of theoretical

design methods, a simpler test facility than those used for full–scale engine tests is more appropriate.

The Rotor Facility was conceived by Ainsworth et al [1988] as a short duration turbine test facility containing a full stage 62% size high pressure turbine stage with an operating point typical of modern high by–pass ratio gas turbines. The turbine is a shroudless design designated the B22 turbine. This turbine was chosen as the rotor mid–height section had been the subject of much work in cascade, reported by Jones [1988]. The turbine operating point is given in Table 1. The short duration air supply for the working section is an Isentropic Light Piston Tunnel (ILPT), described by Jones et al [1973]. Turbine and ILPT geometry are given in Table 2.

The ILPT concept is of a free piston compressor in which the working fluid is compressed isentropically over a period of approximately one second to the required temperature and pressure, then passed through a working section under steady state conditions for a period of order 0.25 seconds. The ILPT mode of operation is described in detail by Schultz et al [1977].

The testing of a model turbine stage in a transient flow facility presented an interesting design challenge. Although from an instrumentation viewpoint the flow severity of the Rotor Facility was reduced, the mechanical complexity of a fully rotating turbine stage was retained. As a result, rotor dynamics played an important part in the Rotor Facility design and successful operation. This paper presents a detailed analysis of the facility's dynamic characteristics.

2 DESIGN CONCEPTS

The Rotor Facility comprises a working section incorporating the turbine stage and a short duration air supply (an ILPT), Figure 1. Operation of an ILPT is complex and care was taken during working section design to ensure that simplicity of operation was a principal design constraint in order to avoid facility operation becoming prohibitively difficult.

The realised design, Figure 2, incorporates a number of novel features which have proven successful during facility operation. The primary objective of the Rotor Facility was to enable the
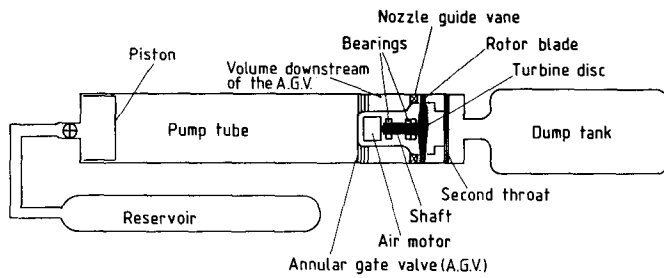
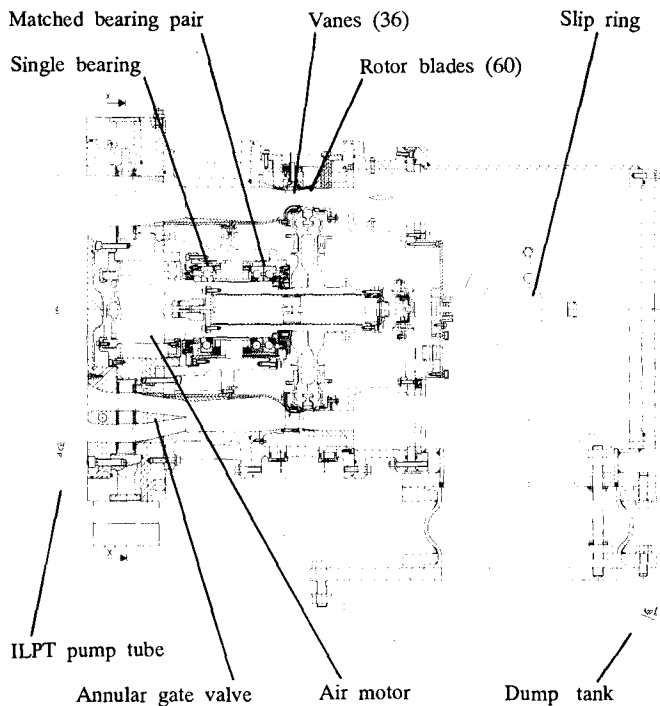**Figure 1** **Layout of the Rotor Facility transient turbine test facility.**

Figure 2 **General arrangement of the Rotor Facility transient turbine test facility working section.**

Table 2	
Turbine and ILPT Geometry	
Vane Mid–Height Exit Angle	70°
Blade Mid–Height Exit Angle	–63.13°
Vane Mid–Height Axial Chord	0.0312 m
Vane Throat Area	0.0185322 m²
Number of Nozzle Guide Vanes	36
Number of Turbine Blades	60
Nozzle Guide Vane Hub/Tip Ratio	0.846
Turbine Blade Hub/Tip Ratio	0.846
Nozzle Guide Vane Aspect Ratio	1.348
Turbine Blade Aspect Ratio	1.741
ILPT Tube Volume	1.604 m³

flow–field in the rotating frame to be studied. In order to facilitate this it was recognised that much effort would be directed at developing suitable blade mounted instrumentation to facilitate the measurement of mean and unsteady pressure and heat transfer. Ease of access to blade mounted instrumentation to enable the regular changes of instrumentation required for its development was, therefore, considered crucial. Access was provided to the turbine blading by choosing a single sided bearing system. This arrangement enabled the entire bladed disc to be removed or re–mounted on its shaft in less than an hour.

The design of a facility with no brake results in the turbine accelerating during facility operation. This was not, however, a problem as over one revolution of the turbine speed change is small enough to be neglected. The Rotor Facility data acquisition system sampled at 500 kHz, giving 7 ms (3.5 kBytes) of information at each measurement point, with a blade incidence variation due to speed change of $\pm$ 0.3°. The effects of this on Mach number are considered in Ainsworth et al [1988], and are essentially negligible. During data acquisition one rotor blade is influenced by 35 nozzle guide vane wakes at design speed. Whilst ensemble averaging over a number of revolutions on one run is not possible, the concentration on amplifying signals in the

rotating frame (Ainsworth et al [1989]) means that signal–to–noise ratios are in practice 50 and the effects of rotor/stator interaction are clearly seen both on mean and unsteady heat transfer (Hilditch & Ainsworth [1990]) and on mean and unsteady pressure measurement (Dietz [1990] and Ainsworth et al [1991]).

2.1 FACILITY DESCRIPTIONS

A full description of the working section (Figure 2) is given in Sheard & Ainsworth [1992] and is briefly described here. The

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choice of an overhung single sided bearing system drove the design of the secondary features. A motor was required to spin the turbine to a pre-run speed close to design speed prior to facility operation. To enable the turbine disc to be removed the motor had to be fitted upstream of it. Space was restricted upstream of the turbine disc so an air motor was chosen as it was small compared to an . equivalent power electric motor. As air motors are purely mechanically devices they have a second advantage as they generate no electrical interference. The air inlet and exhaust ducting for an air motor are substantial which lead to the choice of a multi-spoke Annular Gate Valve (AGV) to separate the ILPT pump tube and working section, the spokes of the AGV being used to duct air in and out of the air motor.

2.2 MODE OF OPERATION

The facility's mode of operation consisted of a pre-run set up procedure, facility operation and finally ILPT operation, described fully by Sheard & Ainsworth [1992]. The pre-run set up procedure commenced by closing the AGV and evacuating the working section and dump tank. The initial pump tube pressure and high pressure reservoir pressure were then set. Facility operation started by accelerating the turbine to a speed of approximately 6500 rpm using an air motor. The air supply to the air motor was subsequently cut, causing the turbine to decelerate under the action of bearing losses. ILPT operation commenced when it was 'fired' (the term used to describe the sequence of events automatically following after pressing the 'fire' button), the valve between high pressure reservoir and pump tube opened, the consequent air flow driving the piston down the pump tube until the required tube temperature and pressure upstream of the piston were reached. The AGV was then opened to allow air into the evacuated working section and the ILPT low speed data acquisition system to be triggered (Oldfield et al [1978]). As the power developed in the turbine was not absorbed by a compressor or brake, the turbine accelerated until the piston reached the end of the pump tube and mass flow through the working section stopped. The initial turbine speed was chosen to ensure the turbine passed through its design speed during ILPT run time.

3 THE BEARING SYSTEM

The bearing configuration was chosen for easy access to blade mounted instrumentation, not mechanical stiffness. It was recognised that the chosen design, being an overhung system, would have a low first natural frequency. The bearing system also had to operate satisfactorily with the extraordinary load conditions imposed on them. During a typical run, the rotating system accelerated at a rate of 1600 rad/sec^2 for 0.2 sec, with an axial gas force on the turbine producing an axial gas load on the bearings. At the end of the run, this load was suddenly removed, leaving a high rotational speed (9800 rpm) with only the pre-load on the bearings. The mode of operation was novel enough for there to be no published information available on actual bearing performance under similar loading. This led to the realisation that a careful analysis and prediction of bearing performance was required, backed up by measurements of actual bearing performance during facility commissioning. The two main causes for concern were the possible onset of 'skidding' caused by the high acceleration (deviation from pure rolling leading to severe frictional heating of the bearings and their subsequent rapid failure) and the occurrence of 'truncation' due to high speed/low load operation (where the ball overides the bearing inner raceway shoulder, again causing rapid wear). During facility commisssioning the bearing cages were fitted with a small steel wire with magnetic pickups being located close to them to provide a once per revolution signal as the cages passed. Turbine shaft speed was also measured which enabled measured bearing cage to shaft speed ratios to be derived.

To avoid truncation at the high speed/low load condition, a high
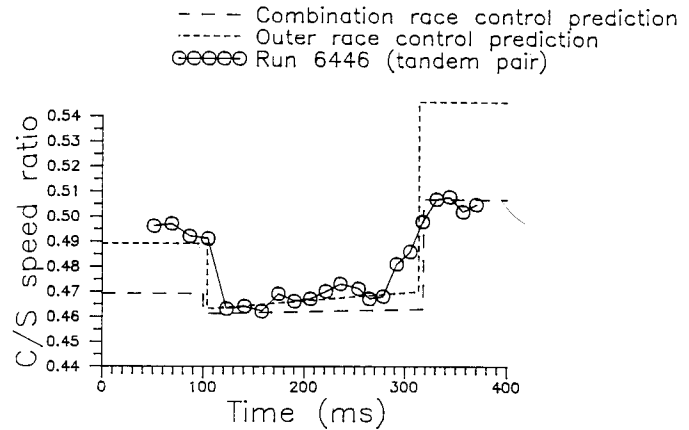
Figure 3 Cage/shaft speed ratio measured for the upstream bearing of the matched pair during ILPT Run 6446 and comparison with predictions.
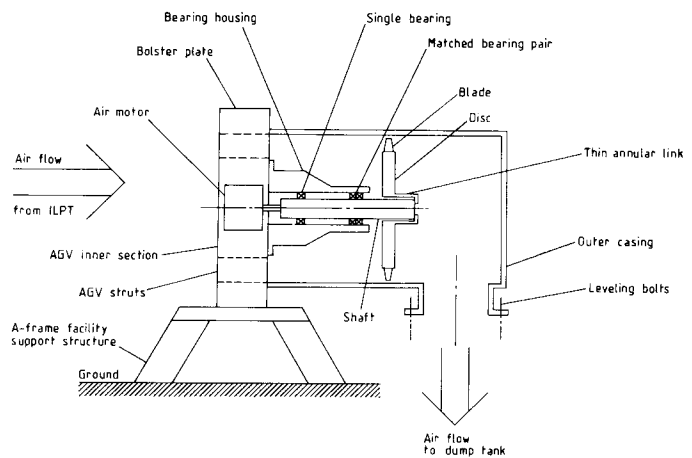

Figure 4 A schematic illustration of the Rotor Facility working section components between the turbine disc and ground.

pre-load (10 kN) was selected for the chosen bearing configuration. Using a standard technique (Stimpson [1989]), predictions of the bearing cage/shaft speed ratio and inner race contact angle were made over the operating envelope. These are compared with measurements made on one of the bearings from the matched pair bearings in Figure 3 and indicated that the actual bearing cage/shaft speed ratio remained close to its predicted (no skid/no truncation) values.

4 ROTOR DYNAMICS

The choice of bearing configuration with the turbine disc overhung was expected to result in at least one natural frequency that would be directly excited at some point over the facility speed range. There were other aspects of facility design recognised as having the potential to introduce additional natural frequencies. The bearing housing was cantilevered off the inner section of the AGV supported by the valve spokes, Figure 4. The air motor was also supported here and was connected to the rotor shaft via a rigid drive coupling. The AGV was mounted within a bolster plate which was in turn mounted on an A frame fixed to the floor. The piston tube, supported by overhead rollers, was bolted to one side of the bolster plate and the working section outer casing was bolted to the other side. A
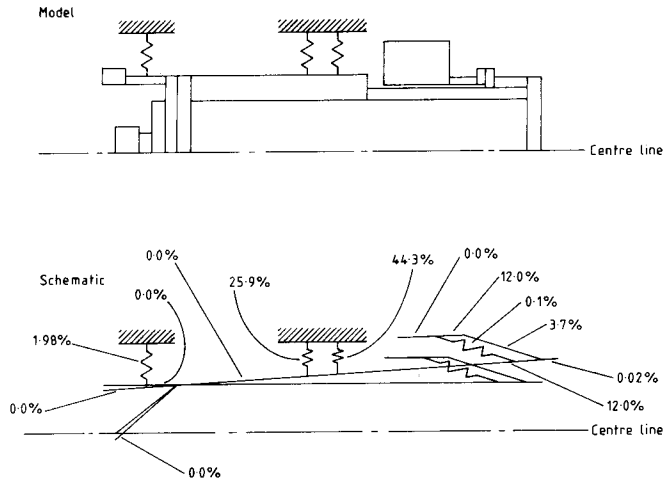
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Figure 5 The beam model used by Beynon [1988] to predict natural frequency and mode shapes of the working section rotating assembly and predicted first mode strain enery breakdown.
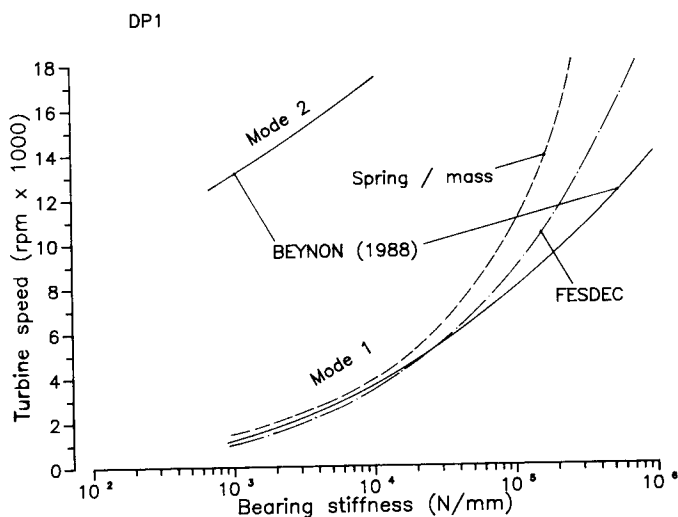

Figure 6 Predicted natural frequencies of the working section rotating assembly.

flexible tube linked the outer casing to the dump tank secured with four levelling bolts. A complicated dynamic response might be expected from such a design.

4.1 NATURAL FREQUENCY PREDICTIONS

A dynamic analysis of the working section rotating system was performed by Beynon [1988]. The turbine was modelled by beam elements of varying size relating to the various annulus sizes along the shaft. The disc was modelled as a rigid element of zero density attached to the shaft with its mass and moment of inertia applied, Figure 5. The predicted first and second natural frequencies were plotted, Figure 6, for a range of shaft support stiffnesses.
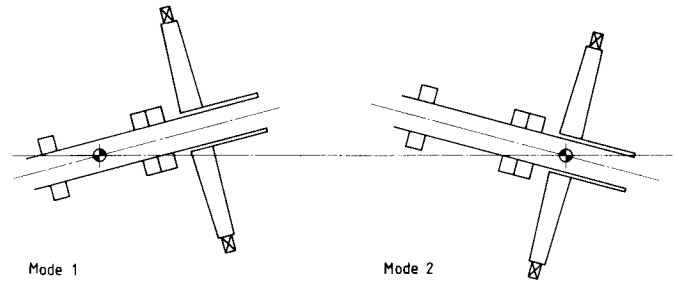

Figure 7 Predicted mode shapes of the working section rotating assembly, from Beynon [1988].

The beam element program also predicted the mode shapes associated with each natural frequency, Figure 7. The first mode was of the most concern as out of balance forces in the plane of the turbine disc (arising due to modifications to blade instrumentation) would excite it directly. The strain energy breakdown through the beam/spring model, Figure 5, indicated that the majority of strain energy was in the matched bearing pair, with only 12% in the thin annular link (defined in Figure 4). This prediction was performed using an estimated bearing stiffness of 20 kN/mm which corresponds to a first mode natural frequency at 75 Hz, 4500 rpm, Figure 6. The actual bearing stiffness was calculated by Dietz [1990], who measured the natural frequency of the shaft alone and the disc/shaft assembly. From a knowledge of natural frequencies and the disc and shaft respective masses, Dietz calculated bearing stiffness to be 14 kN/mm which corresponds to a first mode natural frequency of 67 Hz, 4000 rpm. The error in estimating bearing stiffness was considered small enough to have a negligible effect on strain energy distribution.

In addition to the prediction of Beynon the first mode shape was predicted using a finite element dynamic analysis package, FESDEC, described by Moir [1987]. The disc/shaft assembly was modelled geometrically accurately, Figure 8, using axi–symmetric harmonic elements. Bearing stiffness was varied over the same range as Beynon and the first natural frequency calculated, Figure 6. The results obtained using FESDEC are in close agreement with those of Beynon. The prediction from a single degree of freedom analysis are also included and agree well at lower bearing stiffnesses when the assumption that the entire disc/shaft assembly vibrates as a rigid body is valid. At higher bearing stiffnesses the single degree of freedom analysis departs from those of Beynon and obtained using FESDEC, indicating some shaft flexibility above bearing stiffnesses of 10 kN/mm.

At the start of facility operation the turbine is spun to 6500 rpm and the AGV opened allowing air into the working section. The axial load is applied to the turbine as a step impulse and therefore an axi–symmetric natural frequency and mode shape analysis was performed using the FESDEC model. The predicted mode shape, Figure 9, could be excited by a step axi–symmetric impulse however the predicted natural frequency, 286 Hz, was an order of magnitude higher than the opening time of the AGV and therefore it was neglected.

4.2 NATURAL FREQUENCY MEASUREMENTS

Early spinning trials showed that the facility did have a complex dynamic characteristic. This is illustrated by a typical vibration trace, Figs 10a, b and c, measured with a velocity transducer mounted on the bearing housing as the rotor speed was increased. The aim of the dynamic analysis was to explain the causes of the resonances and if possible, to modify the facility to obtain a better characteristic with less sensitivity to
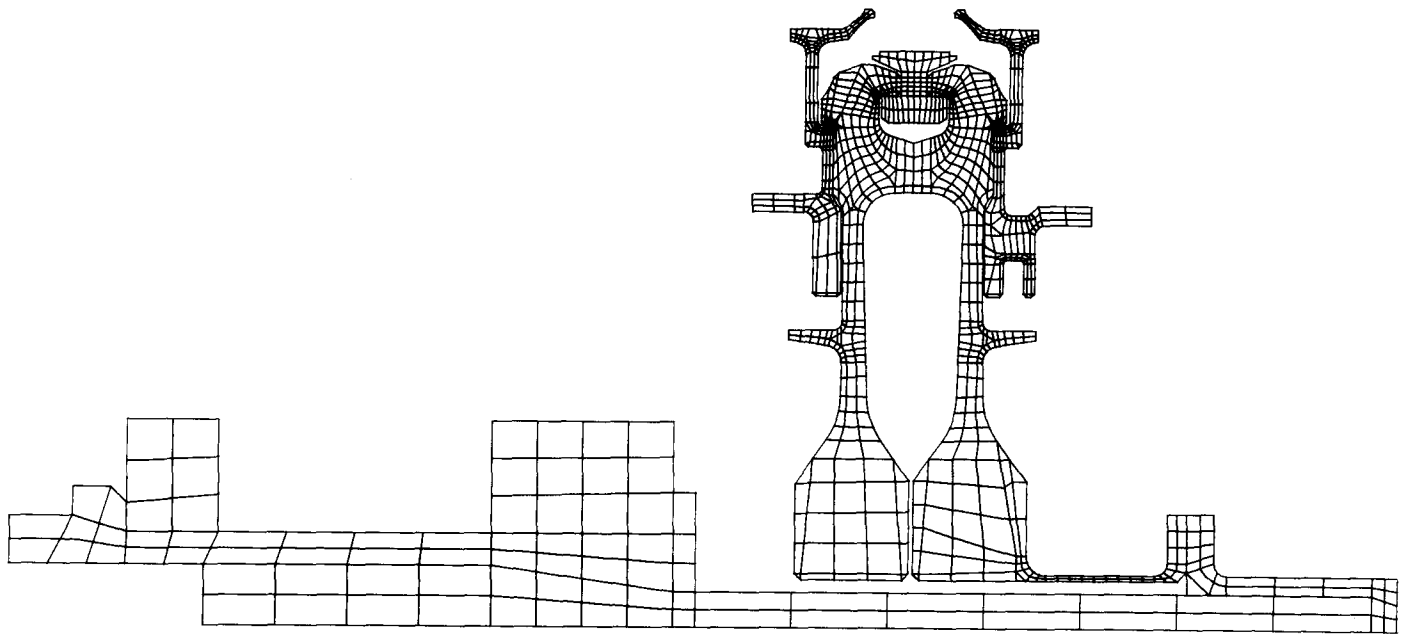
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Figure 8   The axi-symmetric FESDEC mesh used for the analysis of the rotating assembly model.
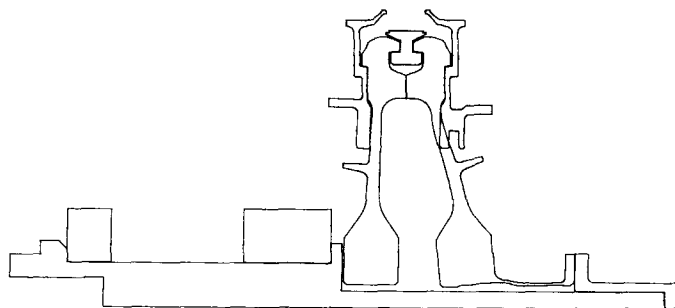

Figure 9   The first axi-symmetric mode shape of the rotating assembly predicted using FESDEC.

imbalance.   It was recognised that a balancing technique was required to reduce the imbalance consistently and efficiently to a level where the resonances were not excited, Figure 10c.

The natural frequencies of a system may be measured using a 'bump' or impulse response test.   If the system is impacted with sufficient energy all its natural frequencies will be excited.

A system with a finite number of degrees of freedom will have a response made up of the same number of natural frequencies. These natural frequencies may be more easily distinguished in the frequency domain.   A Fourier transform of the impulse function gives unity amplitude at all frequencies and the Fourier transform of the system's impulse response gives the system frequency response, of course showing the system's natural frequencies.   This procedure is known as a bump test.

A number of bump tests were performed on the facility to establish the dominant natural frequencies.   The impulse was

provided by a rubber hammer and the resulting transient response measured with a velocity transducer attached to the point under investigation.   By moving the velocity transducer to different locations and striking various parts of the facility in both the horizontal and vertical directions a file of response curves was built up giving a detailed map of the natural frequencies of the facility.   The transducer's output was captured using a Dynamic Waveform Analyser.   The analyser sampled and digitized the signal, storing it for further processing.   An autocorrelation was then performed on the data to separate the periodic signal from the noise.   This procedure effectively multiplies the signal by progessively time shifted versions of itself, emphasising the periodic components and reducing the non-periodic parts.   A Fast Fourier Transform was then performed on the correlation giving a power spectrum showing the relative power in each frequency component.

The bearing housing transducer signal was recorded and its power spectrum calculated, Figure 11, when the disc was struck in the vertical direction.   A uniform window was used in the power spectrum processing.   Dominant frequencies may be clearly seen at approximately 30 Hz, 65 Hz and 70 Hz.   The response was measured using the same transducer but excited by striking the bolster plate, Figure 12.   The 30 Hz frequency and a 75 Hz frequency are visible.   This suggests that these frequencies are related to a vibration mode involving both the bearing housing and the supporting structure.   This was confirmed by a measurement taken with a transducer magnetically fixed to the end of the shaft in the vertical direction with the disc being struck in this direction, Figure 13.   The 65 Hz frequency was dominant here, suggesting it is a mode involving the disc/shaft and bearing housing independent of the supporting structure.

A number of tests were also made with the disc removed.   These tests were aimed at distinguishing the disc/shaft resonances.   It was expected that the resonances would move to higher frequencies with the disc removed as the vibrating mass has been reduced. The bearing housing transducer was used with the shaft being
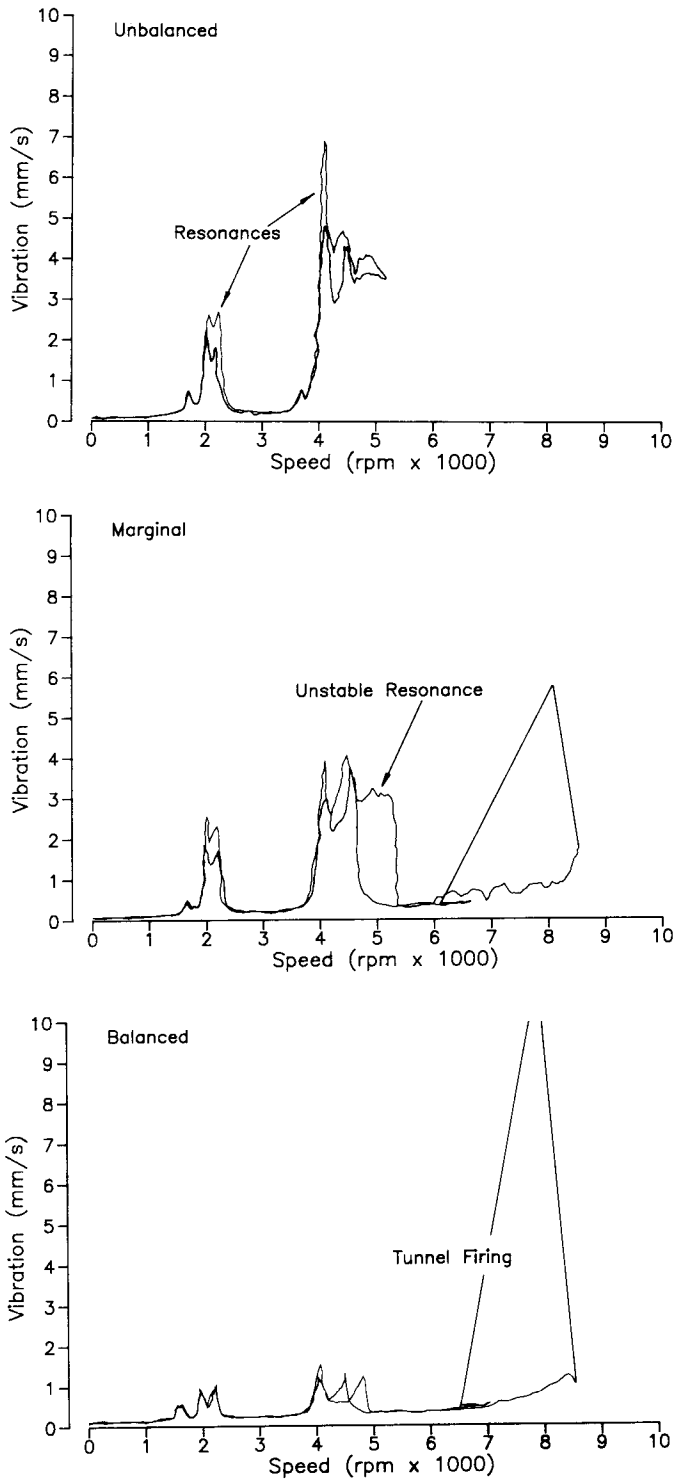
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Figure 10 RMS vibration levels of the rotating assembly for (a) unbalanced, (b) marginal and (c) balanced conditions.
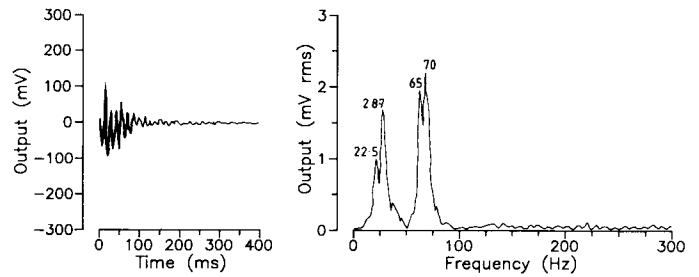

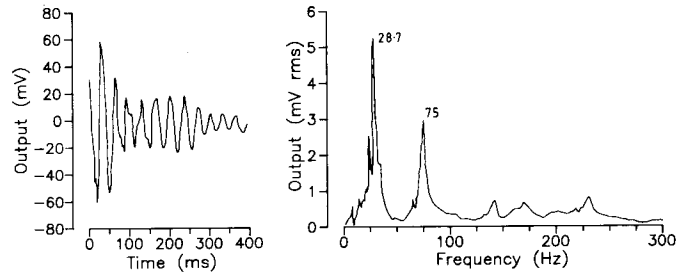Figure 11 Bump test. Bearing housing transducer, struck at disc base.

Figure 12 Bump test. Bearing housing transducer, struck at bolster plate.
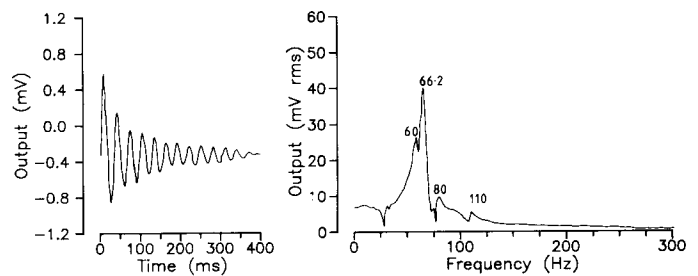

Figure 13 Bump test. Transducer on shaft, struck at disc.

resonances. This can be clearly seen in the shaft only test, Figure 15. The 30 Hz frequency has moved to 20 Hz and the 70 Hz frequency has moved to 60 Hz. The higher frequencies remained unchanged. The natural frequencies of the facility identified are summarised in Table 3.

4.3 FACILITY DYNAMICS

Two main tools were used to analyse the rotor's dynamic characteristic; a control computer used to record the rms velocity signal from the bearing housing transducer each run and a portable balancing machine. The portable balancing machine was able perform tracking measurements at selected multiples of the synchronous frequency. The amplitude of the selected signal component was measured as well as its phase with respect to a reference signal. The signal from a once per revolution optical detector was used as the reference signal.

The portable balancing machine was used to record the output from the bearing housing transducer for synchronous and twice synchronous tracking measurements during a run-up, Figure 16. The first resonance only just visible at 1800 rpm, is the 30 Hz natural frequency of the bearing housing and support. The next resonance, exhibiting a double peak at approximately 2000 and

struck in the vertical direction, Figure 14. The 30 Hz and 70 Hz frequencies remained unchanged but the 65 Hz frequency moved to a double resonance with peaks at 160 Hz and 170 Hz. This confirmed that the 65 Hz resonance was a disc/shaft related mode.

Of the many other tests made, one more deserves reporting. The levelling bolts were found to have a large effect on some of the

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2200 rpm is actually a supersynchronous resonance, vibrating at twice the rotational speed at a frequency equal to the first disc/shaft related natural frequency. It is best explained by the 'Gravity Critical' phenomena, fully described by Vance [1988]. The next resonance, a double peak at 4200 and 4500 rpm is primarily due to the main disc/shaft natural frequency of 60 – 70 Hz. There is also a support resonance at 75 Hz introducing complications. According to Vance, double peaked resonances or 'split criticals' often occur when the rotor response is being strongly influenced by the machine foundation or housing and this seems to be the case here. The vibration mode shape of this resonance will be close to that predicted by the beam model of Beynon but complicated by the support resonance. Resonances of speeds up to 4500 rpm may be explained by the natural frequency measurements and the predictions made using simple models. The responses are stable and repeatable and varied as expected when tests were performed with changes in support stiffness and damping. Above 4500 rpm however, a resonance which was highly unstable and nonlinear was encountered which cannot be explained by any of the simple predictions and measurements.

This unstable nonlinear resonance was sensitive to disc imbalance. At low levels of imbalance the resonance was not excited, Figure 10c, whilst at marginal levels it dropped out suddenly at varying speeds around 5000 rpm, Figure 10b, and at high imbalance levels the resonance could not be driven through, Figure 10a. The instability was closely related to the use of the air motor, as it was not excited during the rotor run–down. Sometimes when the air motor was cut the rotor immediately dropped out of the resonance. The unstable component at the bearing housing is supersynchronous at twice the rotational frequency while measurements on the bolster plate and outer casing showed a large synchronous vibration. There seemed to be a complex interaction between the imbalance and the bearing housing and support/outer casing in this region. Attempts were made to move and decouple any natural frequency effects which could be contributing to this resonance. This was done by altering the stiffness and damping in the supporting structure. A number of modifications were tried with the levelling bolts having the largest effect. Removing these bolts significantly lowered the support natural frequencies. The response measured

with the levelling bolts removed, Figure 17, indicated that although some of the lower resonances have been moved, the unstable resonance has remained unaffected.

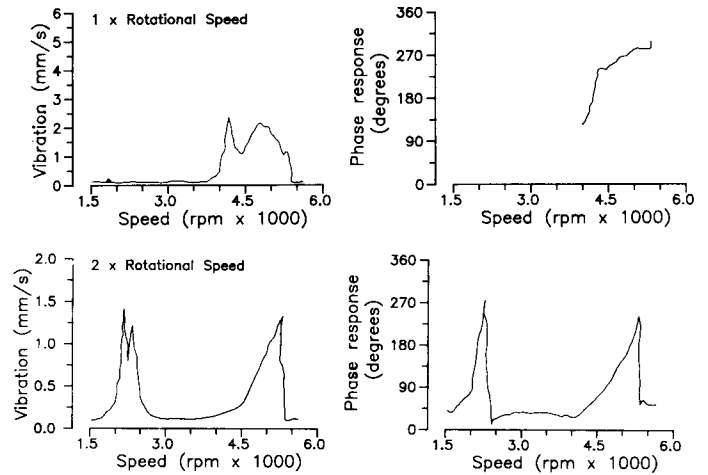| Table 3		
Oxford Rotor Facility Working Section Modes of Vibration		
Vibrating Components	Frequency	
	Hz	rpm
Bearing Housing	29	1750
Facility Support Structure	33	2000
Dump Tank	40	2400
Rotating Assembly	66	4000
Facility Support Structure	75	4500
Facility Support Structure	100	6600

Figure 16 Bearing housing vibration measurements for synchronous and twice synchronous tracking measurements during run up.
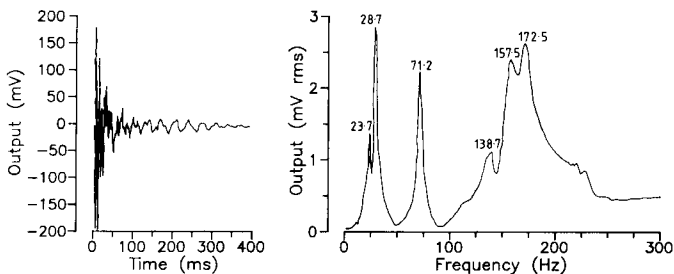

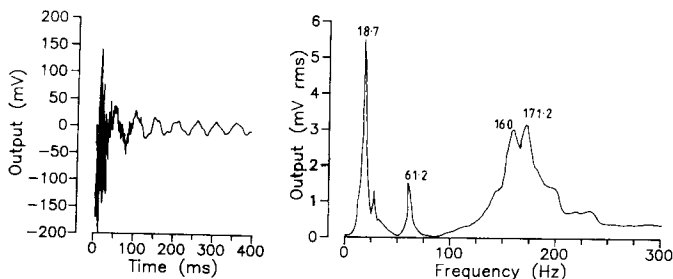Figure 14 Bump test. Bearing housing transducer, struck on shaft with disc removed.

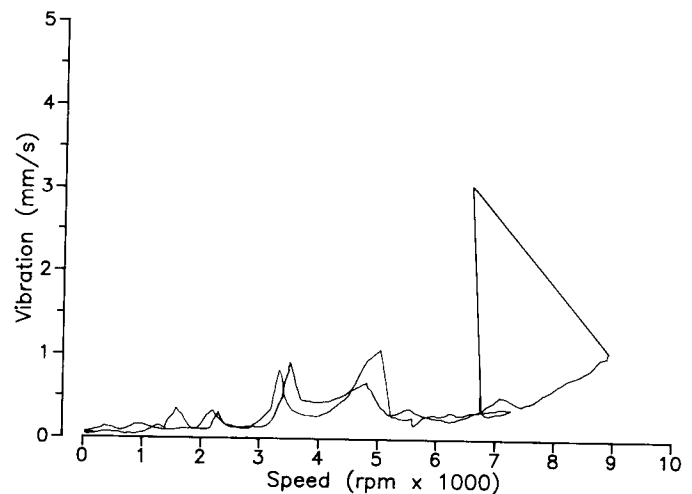Figure 15 Bump test. Bearing housing transducer, struck on shaft with no levelling bolts.

Figure 17 Rotating assembly run up trace with no levelling bolts.

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Rubbing of the blade tips or the labyrinth seals was another possible cause of the resonance. Once rubbing had been initiated by an imbalance level at a critical speed, an unstable vibration would continue at higher speeds under the driving energy of the air motor until the air motor was cut or the instability suddenly disappeared. Such behaviour had been seen by Beynon on a non-linear test rig. This possibility was investigated by gradually stripping the rotor of blades and seals and measuring its response during run-up after each modification. Although signs of rubbing were found in the seals, the unstable resonance could still be excited when the rotor had been completely stripped and there was no possibility of any rubbing. It was concluded that the rubbing was an effect rather than a cause of the unstable resonance.

It became apparent that any further dynamic investigation was impractical as it would involve major modifications to the facility. The development of an effective field balancing technique therefore became the most practical course of action.

5 FIELD BALANCING

Field balancing is the technique of balancing fully assembled rotors under operating conditions. Even a rotor which has been fully balanced in a dedicated balancing machine will often need further correction when installed due to variations in its mode shapes introduced by the flexibility of its bearings and supports. The support resonances and other masking vibrations often make the field balancing task more difficult than machine balancing. Although the rotating assembly was machine balanced before assembly, it was recognised that field balancing would often be necessary because of the frequent disc weight variations introduced by instrumentation changes. A very low level of balance was required because of the sensitive nature of the facility's dynamic response. To achieve this, an effective and efficient field balancing technique was required.

5.1 REQUIREMENTS FOR EFFECTIVE BALANCING

The simplest balancing method is single plane balancing. Single plane balancing involves measuring the vibration vector (magnitude and phase) at a certain speed with the disc in the unbalanced condition and with a trial weight added at a known position in the balancing plane. The required balance mass can be determined from the systems response to the trial mass. The single plane balancing method is only effective for certain types of rotors. The requirements are:

a) Linearity. The frequency response must be linear, ie doubling the trial mass should double its corresponding vibration vector. Any spurious vibration components which are measured in addition to the imbalance component detract from this linearity.

b) Couple Imbalance. If the principle axis of inertia is inclined to the axis of rotation a rotating couple will result. This condition is known as dynamic imbalance and may occur in rotors which are statically balanced where the centre of gravity is on the axis of rotation. Two balance planes are needed to reduce this couple imbalance. For narrow rotors the couple imbalance is insiginificant and one balancing plane will suffice.

c) Flexible Rotors. If the rotor is too flexible, having a number of resonances in the operating range, the correction weights will have different effects at different speeds as the vibration mode shape changes. The single plane method only balances the rotor at one speed and an extra balance plane is needed for each speed at which the rotor is to be balanced. If the rotor is rigid, the mode shape will be constant at all speeds (there is only one resonance), and a single plane balance will balance the rotor for all speeds.

d) Repeatability. The chosen balancing speed and the system configuration must be identical for each set of balance runs. Repeatability in a resonance region is poor due to the high rate of change of vibration level. Variations in disc/shaft alignment or alignment of the shaft in the bearings also affects the repeatability. Growing support looseness can be another contributing factor.

5.2 EXPERIENCE WITH THE ROTOR

The rotor design had only allowed for one trim balancing plane in the hope that a single plane method would suffice. The degree to which each of the above requirements is met in the Rotor Facility will now be considered.

The signal from the bearing housing transducer was used for balancing. The portable balancing machine was used to measure the rotational speed and the magnitude and phase of the synchronous vibration component, filtering out the unwanted frequency components.

Although the rotor was effectively narrow, the correction plane was on one side of the disc. As a result, any imbalance introduced in the plane of the blades could never be fully corrected by the balance weight without introducing a couple imbalance. This effect was minimized by individually weighing each blade and calculating a blade distribution which would not introduce any further imbalance. With this technique very low levels of imbalance were possible.

The flexibility of the rotor, evident in the number of resonances encountered in the run-up trace, posed a problem for the single plane technique. In the regions above and below the major resonances the rotor is relatively insensitive to imbalance with a small vibration signal occurring in these regions. The sensitivity of the portable balancing machine did allow balancing in these regions but although the vibration level could always be improved at the speed selected for balancing (eg from 0.38 mm/s to 0.07 mm/s at 3000 rpm), the resonances could only be reduced to a certain level after which no further improvement could be gained. It was therefore necessary to attempt to balance in the resonance region. After some trial and error it was determined that balancing at a speed of 4650 rpm gave the best results, reducing the resonances below this speed and also reducing imbalance influence on the unstable resonances above.

6 VIBRATION LIMITS

Wide ranges of operating 'levels can be experienced during operation of the facility and particularly while the rotor was being balanced. Some guidelines and limits were necessary to prevent damage to the facility and to ensure operator safety. When defining vibration limits for a rotating machine there are two distinct criteria which need to be considered. One set of criterion refers to vibration severity, a measurement of the greatest rms value of vibration velocity measured on the machine. The second set refers to the balance quality and is expressed as permissible residual imbalance. The two are complimentary since for a given rotor, a particular imbalance will give a corresponding vibration severity. The residual imbalance only reflects the balance state of the machine and does not reflect deterioration in any of the machine components. It should only be used as a criteria during balancing. The vibration severity criteria may be used as a guideline during machine operation.

6.1 VIBRATION SEVERITY

The International Standard ISO 2372 [1974] specifies limit values for the vibration severity of machines. The standard gives limits for six classes of machine. The limits for the class which applies to the Rotor Facility are given in Table 4.

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Table 4	
ISO Standard 2372, Mechanical Vibration of Machines	
Balance Quality	Vibration Level
Precision quality	< 1.8 mm/s
Commercially acceptable	1.8 to 4.5 mm/s
In need of attention	4.5 to 11.2 mm/s
In need of immediate attention	11.2 to 28.0 mm/s

Table 5	
Vibration Limits of Rotating Machinery from LIFSHITS et al [1986]	
Balance Quality	Vibration Level
No faults	< 1.6 mm/s
Acceptable	1.6 to 3.2 mm/s
Marginal	3.2 to 6.3 mm/s
Failure probable	6.3 to 12.7 mm/s
Danger of immediate failure	> 12.7 mm/s

In addition to the ISO standards and the similar VDI guidelines there have been a number of papers written which propose vibration severity limits based on statistical and concensus evaluations of vibration and failure data from many operating units. Lifshits et al [1986] is one such paper. The limits he proposed, with the correction factors applicable to the rotor facility applied are given in Table 5.

These proposed severity limits point to three main operating regions. Vibration velocities less than 4 mm/s are acceptable for normal operation. A marginal area exists from 4 to 10 mm/s where the facility can be run but should be monitored closely and the problem rectified at the next opportunity. Operation of the facility with vibration levels above 10 mm/s should be avoided. These limits agree with experience in the Rotor Facility to date. At levels above 10 mm/s the noise and the vibration transmitted through the floor are enough to convince any operator that continued running will be likely to damage his health and that of the facility. At levels less than 4 mm/s the vibration seems satisfactory and levels less than 1 mm/s are hardly noticeable. Current experience with the rotor is that vibration levels of less than 4 mm/s are routinely achieved.

6.2 PERMISSIBLE RESIDUAL IMBALANCE

The Standard ISO 1940 [1973] 'Balance Quality of Rotating Rigid Bodies' gives permissible residual imbalance as a function of the maximum service speed for various groups of representative rotors. The standard uses specific residual imbalance (e) as its representative parameter. It is the residual imbalance divided by the rotor's total mass and is equivalent to the displacement of the rotor's centre of gravity from its axis of rotation. Practical experience with rotors showed that for rotors of the same type, the product of specific residual imbalance with speed was constant. This product was used to define quality grades for different types of rotors. The quality grade is based on the upper limit for the specific residual imbalance for each rotor type in millimetres and the rotor speed in rad/s. A quality grade of 6.5 was recommended for the rotor facility by King [1988]. At 4650 rpm this gives a specific residual imbalance of

15 um. For the 59 kg rotor disc this is equivalent to an imbalance of 885 gmm or a correction weight of 6 grams at 144 mm. Experience while balancing the rotor has shown that a balance weight of 5 grams at 144 mm is enough to throw a well balanced rotor out of balance, which agrees well with the permissible residual imbalance guidelines.

7 SUMMARY AND CONCLUSIONS

The facility described in this paper is a logical progression from two–dimensional cascades of aerofoils to a three dimensional working section incorporating a full stage turbine. This enables a more accurate simulation of the turbine stage flow, by including the effects of rotation and three dimensional geometry.

In moving from a stationary to a rotating experiment the facility dynamic performance became an important consideration. Three discrete system natural frequencies were evident during run up of the turbine to 4500 rpm corresponding with expectations from static frequency response tests and predictions. Above 4500 rpm a non–linear and unstable resonance was evident and although minor modifications were made, no significant improvement to this particular resonance was achieved.

The complex dynamic performance of the facility has required careful study, the conclusion of which is a set of guidelines for running and balancing. In practice, by systematic means of reducing the system imbalance, the peak vibration velocity encountered during facility operation was routinely well within the category of 'precision quality' as defined in international standards for this type of machine. The designers' original goals have, therefore, been realised as the facility may be run while making the numerous changes to blade mounted instrumentation required for instrumentation development.

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