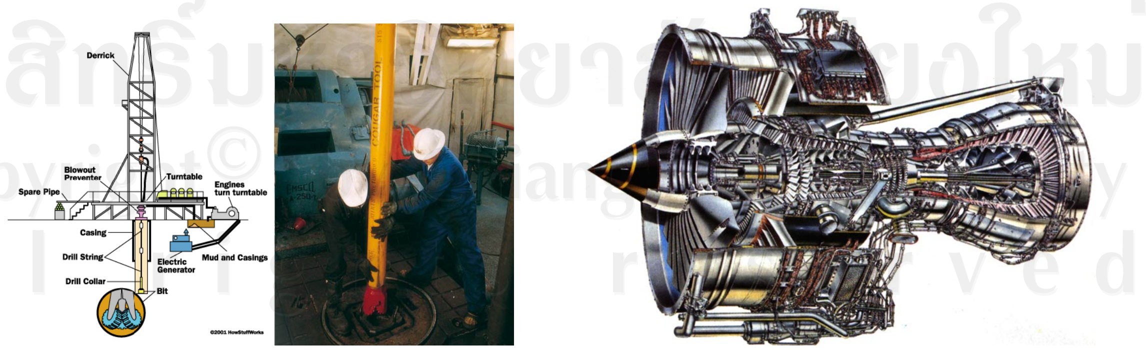
CHAPTER 1

Introduction

1.1 Rotating Machinery

Rotating machines generally consist of two main parts which are a rotating part, referred to as the rotor, and a stationary part on which the rotor is mounted. Transmission shafts, parts of reciprocating machines that have only rotational motion and other rotating machine elements can be considered as rotors. The parts of a machine that do not rotate are usually referred to as the stator. The study of the dynamic behaviour of rotor-stator systems and the vibration that arise in them is referred to as "rotordynamics".

Rotating machines are used in a wide range of applications in manufacturing and other industry, particularly for power transmission. Many of these machines are turbomachines which perform some form of power conversion on a working fluid. The energy loss due to the leakage of the working fluid reduces the efficiency of such machines. In recent decades, efforts have been made to minimizing the leakage of fluid by reducing the sizes of gaps/clearances between the rotor and the stator. On the other hand, increases in size and power of machines and machine operation in high speed range are often needed. With these requirements, the vibration of the machine and possibility of contact interaction



(a) Drilling rig in petroleum industry (b) Rolls Royce Trent 900 Aircraft engine

Figure 1.1: Examples of rotating machines

across clearances between the rotor and stator becomes a major issue in the design and operation of such machines.

1.2 Vibration in Rotating Machinery

Vibration in rotating machines can be divided into natural vibrations (or self-excited vibration) and forced vibrations. The natural vibrations are caused by some initial conditions with no further exciting force and they depend on rotor's properties which are the natural frequencies, the damping ratio and the mode shape of the rotor system. The forced vibrations are the vibrations caused by an external time-varying forces (e.g. unbalance forces, cutting forces in machine tools, fluid forces in turbo machines), or moments or vibration disturbance (e.g. vibration of foundation). The most importance source of excitation in rotating machines is the unbalance forces. The imbalance of a rotor could be caused by imperfect manufacturing or misalignment. The unbalance force is a major cause of vibration in rotating machines [1, 2]. A flexural rotor always develops large vibration at a critical speed and this may lead to interaction between the rotor and it's surround. This rub interaction is a main cause of rotating machine vibrational instability, which can cause self-destruction of the machine. Such instability takes the form of a self-excited vibration that grows in severity over time until a steady limit-cycle response is attained or else failure occurs.

1.3 Rotor-stator Interaction

Interaction between the rotor and it's surround may occur under normal operation such as during speed-up or coast-down through a critical speed. The rotor vibration amplitude tends to a maximum when the operating speed approaches the critical speed. This cause of contact can be predicted and prevented in the design process or subsequently by balancing of the rotor. The conventional auxiliary bearings (also called back-up bearings or retainer bearings) can be used to limit a large amplitude vibration of the rotor and prevent rotor-stator contact. However, the dynamic stability of the rotor system becomes a more complex issue. The rotor-stator interaction also may arise under abnormal operation such as the stator motion, base vibration, external disturbance and due to thermal effects. These situations cannot be easily predicted or tackled in the design process.

Rotor-stator interaction is a nonlinear dynamic behaviour. Therefore, different response behaviours of the system can be caused by the same excitation. Typical motion patterns that may arise in a rotor system after a contact are

1) **Synchronous Full Annular Rub:** While the rotor runs in a forward whirl, imbalance excites the rotor orbit and the rotor has a permanent contact with interior surface of the stator. The bending stress of the shaft is constant. Similar behaviour can occur due to non-linear stiffness of components that support or surround the rotor. These may cause a response branch for vibration where critical vibration (resonance) occurs over a wide interval of rotational speed. Such non-linear vibrational states can occur even when residual excitation of the rotor, e.g. due to unbalance, is relatively low. A motion pattern involving synchronous whirl is probably least dangerous and a rotor may encounter full annual rub for a short time without damage.

2) **Partial Rub:** The rotor has multiple impacts in a revolution. The contact occurs in short period of time and gives a high impact force. The motion can be periodic, non-periodic or chaotic. It should be remarked here that this type of behaviour is not the main focus of this thesis. Partial rub is found more often than a synchronous full annual rub during the operation of rotating machine. Partial rub is covered by many other studies [3, 4, 5] and is usually associated with asymmetry (misalignment) of the rotor-stator assembly or its loading.

3) **Backwards Whirl:** If rub persists, friction forces can drive the rotor whirl vibration in an opposite direction to rotation if friction is sufficiently high. Fully developed backwards whirl can involve high frequency vibration with very large contact forces and is therefore a serious malfunction that can lead to structural failure.

1.4 Background

1.4.1 Rotor-stator interaction dynamics

As previous explained, contact between a rotor and it's surrounding can cause the motion of the rotor-stator system to become dynamically unstable. This instability can be predicted by understanding of the contact dynamics. An early study by Johnson [6] modeled a vertical shaft supported by a clearance bearing and two zero-clearance bearings. It was found that the amplitude response curves can be different according to whether the speed is increasing or decreasing. This type of hysteresis, or bistable response behaviour, can be considered as one form of instability.

Friction effects have an important role in contact behaviour and may drive a rotor to undergo backward whirl. Such behaviour, though uncommon, has been recently reported as occurring for an advanced liquid hydrogen (ALH) turbopump [7] and windpower anemometers [8]. An early theoretical study by Black [9] modeled rotor-stator contact to predict possible whirl behaviour. Dry friction backward whirl was also produced in experiments within a ball bearing. Black's conclusion were that

1) For a synchronous (forward) whirl involving a multi-mode rotor-stator system, the conditions for rub interaction can be described by a straightforward vector relationship involving the dynamic properties of the component systems.

2) Backward whirl for a flexible rotor and a flexible clearance bearing is possible only when shaft speed is within the range from the rotor/stator natural frequency to the next (higher) natural frequency of the combined rotor-stator system.

3) The feasibility of backward whirl with rolling or with slip can be determined by calculation of the required friction angle (ϕ). Generally, a plot of required angle versus frequency will be of the "U" shape. Thus, the bottom of the "U" shape, which depends upon the degree of rotor and stator damping, determines a minimum angle of friction for possibility of backward whirl.

4) The effect of mass unbalance in a backward whirl regime can be considerable as rotor speed is approaching the backward whirl speed. In this condition, pronounced beating occurs between forced synchronous vibration and the backward whirl. With sufficient mass unbalance, the direction of whirl is reversed and subsequent increase in rotor speed is accompanied by synchronous whirling of the combined rotor-stator system.

Childs and Bhattacharya [10] revisits Black's original approaches and further considered the possibility of multiple rotor modes. It was thus demonstrated that Black's conclusions remain generally valid. Bartha [11] undertook a theoretical and experimental study of dry friction backward whirl and gave recommendations on avoiding backward whirl. Muszynska [12] considered a non-linear mathematical model giving two exact stationary solutions ("synchronous full annular rub" and "dry whip") and investigated their stability. Muszynska concluded that high stator stiffness, high surface friction coefficient, low rotor stiffness and damping can make the rotor synchronous vibration unstable and with rotor speed higher than the rotor first natural frequency lead to backward whirl.

There are many research studies focusing on characteristics and solutions of a model for interaction between rotor and stator. Childs [13] analyses and explains both hypotheses and experimental results that circumferential stiffness variations induced by rubbing over a portion of a rotor's orbit can lead to parametric excitation of half-speed whirl at a rotor's natural frequency. Fumagalli et al. [14] simulated the dynamics and loads of a rigid rotor in a simple rigid bearing with clearance considering cylindrical and conical motion at high speeds. Fumagalli and Schweitzer [15] experimented with the contact interaction of a spinning rotor within a retainer bearing. Von Groll and Ewins [16] used the harmonic balance method to calculate the periodic response and stability of non-linear rotor-stator contact under periodic excitation.

Although stability investigation methods for rotor-stator contact problems have been widely studied, there is still a need for new methods that can be usefully applied in the design of rotating machines to prevent instability in the rotordynamic behaviour.

1.4.2 Active vibration control in rotor systems

Vibration suppression of rotating machines is an important problem. During the operation of a high speed rotor, several lateral bending modes of vibration can be excited. There is a possibility of excessive vibration and a passive vibration control is not always a viable mean to suppress vibration of all these modes. Due to the rapid development of actuation technology and digital signal processing over several decades, many active control strategies have been devised in order to reduce lateral vibration of a rotor and minimize the risk of unwanted rotor-stator interaction [17, 18, 19, 20, 21]. However, disturbance forces, base motion or shock vibration are other potential causes of contact and some active vibration control strategies are usually designed by assuming a linear dynamic response of the rotor system. Such designs can result in unacceptable vibration, or loss of stability when rotor-stator interaction occurs, which may cause damage to the machine.

Previous works on active control of vibration in non-linear rotordynamic systems cover various situations. Inoue et al. [24] considered the unbalance compensation in a single disk rotor with nonlinear supports. Cole [25] designed model-free controllers for robust synchronous control during rub based on a geometric analysis of circular whirl with rub. Ginzinger and Ulbrich considered the problem of controlling rotor vibration during different phases of interaction with an active controlled auxiliary bearing. It was shown that motion of an auxiliary bearing can be synchronized with the rotor orbit in order to minimize contact force while orbit size maintaining acceptable [26, 27]. In other work, destabilizing nonlinear effects have been accounted for in controller designs through linear approximations [28, 29, 30, 31].

1.5 Objectives of the Thesis

When a machine rotor is running through critical speeds or large disturbances act on the rotor, such as sudden increases in unbalance or base vibration, rubbing between the rotor and the surrounding stator can be possible. This may lead the machine to malfunction or cause damage. If rub behaviour can be predicted accurately then this failure may be prevented. Furthermore, it will open up possibilities for the design of active vibration control systems that can assure that if rub occurs, severe or unstable vibration response behaviour is avoided. Previously [6, 9, 10, 12, 25], simple models based on frequency response characteristics have been used to predict when continuous rub can occur in both forward and backward whirl cases but they have an accurate prediction only with a single contact plane and an assumed whirl solution is required. Therefore these methods can be used to check only for particular forms of solution and it cannot ensure that any whirl response will converge to this particular solution. Knowledge of both local and global stability of a system response is thus important, particularly if non-linear effects is present and the dynamic characteristic of the system are to be altered using feedback control. The Lyapunov stability condition based on state space model approach will be considered in this research because it is able to treat non-linear elements and verify whether global stability of a system is achieve or preserved under active control. Also, the assumption of particular whirl solution is not necessary. Another advantage of state space methods is the ability to deal with multiple non-linear elements and thus multiple potential contact planes. Lyapunov stability analysis also extends in a natural way to controller design problems, and this will be further considered within this thesis.

The overall aims of this research are

1) To develop methods to investigate vibrational stability for non-linear rotor-stator contact based on state space modeling.

2) To develop control strategies which can be implemented on a real machine for preventing vibrational instabilities arising from rotor-stator contact.

1.6 Thesis Outline

The remainder of thesis begins with a description of a basic rotor-stator dynamic model in Chapter 2. In this chapter, the contact between rotor and stator is assumed to be possible only in one plane. Basic predictions of bi-stable and unstable response behaviour involving forward whirl and backward whirl are considered.

The test rig and experimental setup are described in Chapter 3. The key components such as rotor, stator, power electronics and sensors are explained in detail. An active magnetic bearing (AMB) is used to apply the control forces to the rotor and to simulate other forms of disturbances. The direct measurement of rotor-stator interaction forces will be considered as a means to achieve stabilizing control approach in this research and so a novel design of force sensing device is explained and the calibration procedure outlined. The instrumentation and data acquisition equipment are also described.

In Chapter 4, a mathematical model of the test rig dynamics is developed. The peak-picking method is used to identify parameters for the rotor model. The parameters for the stator model are obtained by free vibration response measurements. A full non-linear state space model is derived by the combination of a linear model of rotor-stator structure dynamics and a nonlinear interaction model.

Chapter 5 presents two stability prediction approaches. The Nyquist stability approach based on the frequency response function of the rotor system and an alternative Lyapunov based approach. The Lyapunov approach is based on the state space model of the rotor system. This provides a stability condition in the form of linear matrix inequalities (LMIs) which can be further used in a controller design procedure. In this chapter, a sufficient condition for global stability is obtained considering systems prone to both forward whirl and backward whirl response with rub.

Chapter 6 develops the controller design procedure based on a Lyapunov approach. The control method is termed "dynamic force feedback control" because it requires a direct measurement of the rotor-stator interaction forces. A globally stabilizing controller gain can be obtained by using standard numerical routines for solving LMIs.

Simulation and experimental results involving both forward and backward whirl instability cases are presented in Chapter 7. The control performance is first evaluated by simulation. The implementation of the control on the experimental system is explained and the experimental results shown and discussed.

The conclusions of this research are presented in Chapter 8 and the achievements of the work summarized. The suggestions for further work and possibilities for industrial application are also described.