Fault Diagnosis of Rotating Machinery

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ABSTRACT—This paper aims to provide a broad review of the state of the art in fault diagnosis techniques, with particular regard to rotating machinery. Fault diagnosis is a subject too wide ranging to allow a comprehensive coverage of all of the areas associated with this field to be undertaken, and it is not the authors' intention to do so. However, a general overview of the broader issues of fault diagnosis is provided, and several of the various methodologies are discussed. A detailed review of the subject of fault diagnosis in rotating machinery is then presented. Special treatment is given to the areas of mass unbalance, bowed shafts, and cracked shafts, these being among the most common rotor-dynamic faults. Vibration response measurements yield a great deal of information concerning any faults within a rotating machine, and many of the methods using this technique are reviewed.

Introduction

The objective of this paper is to provide the reader with an insight into recent developments in the field of fault diagnosis, with particular regard to rotating machines. The subject of fault diagnosis in rotating machinery is vast, including the diagnosis of items such as rotating shafts, gears, and pumps. The different types of faults that are observed in these areas and the methods of their diagnosis are accordingly great, including vibration analysis, model-based techniques, and statistical analysis. It is not the intention of the authors to attempt to provide a detailed coverage of all of these areas, since to do so would be beyond the scope of this paper. Nevertheless, it is intended that the reader should be left with an understanding of the wide range of topics involved, while detailed consideration is given to the subject of the fault diagnosis of rotating shafts.

The literature on the subject of fault diagnosis is vast and wide ranging, encompassing such areas as general surveys, general system modeling, and methods applied to the fault detection and isolation (FDI) of specific items of machinery, such as that found in land- and marine-based power plants and in aeroengines. Many kinds of FDI techniques can be applied in different situations, including both static and dynamic processes, where the same method can often be employed using different input and output parameters, depending on the system type.

The present day requirement for ever-increasing reliability in the field of rotor dynamics is more important than ever before and continues to grow constantly. Advances are continually being made in this area, due largely to the consistent demand from the power-generation and transportation industries. Because of progress made in engineering and materials science, rotating machinery is becoming both faster and lighter, as well as being required to run for longer periods of time. All of these factors mean that the detection, location, and analysis of faults play a vital role in the field of rotor dynamics.

The paper is divided into different sections, each dealing with various aspects of the subject: it begins with a summary of previous surveys and reviews of fault diagnosis, followed by a general overview of the numerous means of FDI modeling that exist today. Fault diagnosis in rotating machinery is then discussed in further detail, including general condition monitoring strategies. Special treatment is given to the subject of FDI by means of vibration analysis; an area including the common rotor-dynamic faults listed below.

One of the major areas of interest in the modern-day condition monitoring of rotating machinery is that of vibration. If a fault develops and goes undetected, then, at best, the problem will not be too serious and can be remedied quickly and cheaply; at worst, it may result in expensive damage and downtime, injury, or even loss of life. By measurement and analysis of the vibration of rotating machinery, it is possible to detect and locate important faults such as mass unbalance, shaft bow, and cracked shafts. Problems in rotating machinery may also be caused by degradation in the bearings; however, it is beyond the scope of this paper to report on the many different phenomena involved in bearing failure, and the reader is referred to the excellent coverage given by Neale (1995).

Previous Literature Reviews and Surveys

Several survey papers have been written in recent times on fault detection and isolation and on the general state of the art in rotor dynamics. At the heart of fault diagnosis lies the model-based approach, whereby as many variables and system parameters are taken into account as possible in order to construct a detailed mathematical model of the system under observation. Once the dynamic behavior of the system has been "adequately" modeled, it should, theoretically, be possible to detect faults via analysis of changes in input parameters to the model. Observer-based FDI is one of the most widely used model-based approaches. In

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this technique, knowledge observers—a control configuration akin to that of an analytical observer—generate signals known as residuals. Faults are detected when predetermined threshold levels assigned to each residual are reached. Willsky (1976) gave a comprehensive survey of design methods for FDI that, although designed for linear systems, may be carried over to nonlinear applications in many cases. Isermann (1984) reviewed the detection of process faults based on modeling and estimation methods, involving the estimation of unmeasurable process parameters and variables. A relatively new approach to system modeling, tending to become more and more popular, is the use of fuzzy logic and neural networks. Frank and Köppen-Seliger (1997) produced a paper outlining the most up-to-date developments in artificial intelligence for fault diagnosis; particular emphasis was placed on the use of fuzzy models and fuzzy logic for problem evaluation. The paper introduced the knowledge observer concept described above. Isermann (1997) presented an extremely comprehensive overview of the state of the art in FDI, and Isermann and Ballé (1997) gave a review of recent trends in model-based fault detection and diagnosis. Leonhardt and Ayoubi (1997) summarized various classification and inference methods applicable to automatic fault diagnosis, again focusing on fuzzy logic and neural networks, leading to the consideration of neuro-fuzzy algorithms and also to the development of a new algorithm. Observer-based fault diagnosis of nonlinear systems was addressed by Garcia and Frank (1997), who considered the extension of linear methods to nonlinear systems and also looked at the robustness of these extensions to unknown inputs. The book by Patton, Frank, and Clark (1989) provides a comprehensive survey of a broad range of topics concerned with the fault diagnosis of dynamic systems. In the more specific area of fault diagnosis in rotating machinery, Eshleman (1984) reviewed many aspects of significance, including the vibration analysis of faults in rotors, bearings, seals, dampers, and foundations. Doebling, et al. (1996) presented a broad survey of the literature concerning damage identification and health monitoring of structural and mechanical systems from their vibration properties.

Fault Detection and Isolation (FDI)

The aim of this section is to provide the reader with an understanding of the state of the art in fault diagnosis. Model-based fault detection is, at this time, directly employed in most areas of fault diagnosis. The model-based approach involves the establishment of a suitable process model, either mathematical or signal based, that can estimate and predict process parameters and variables. Isermann (1994) described the main principles involved in model-based procedures and outlined their importance for the realistic modeling of faults. He concluded that more than one method of FDI should be used to best reach an accurate diagnosis. Fault trees and forward and backward chaining are methods of fault diagnosis addressed by Isermann (1993). A comprehensive overview of fault diagnosis methodology is first presented based on process measurements, dynamic models, and parameter estimation to generate fault symptoms. Fault trees and forward and backward chaining then provide the method of fault classification. Fault trees are a heuristic means of decision making, constructed partly from knowledge of physical laws and partly from experience in the field, which may not necessarily be described exactly by these laws. Forward and backward chaining mimics the human process of decision making. All possible outcomes are considered as a first step (forward chaining). The decision-making process is then enhanced by the introduction of additional information, and the most likely outcome due to this extra input is analyzed, which involves the input of yet more data (backward chaining). The procedure continues until either it is concluded or no more likely outcomes are possible.

Natke and Cempel (1991) used the definition that a fault will alter the dynamic behavior of a system to construct a model to detect changes in this dynamic behavior and thus identify faults. Various physical parameters and model sensitivity to fault size are used to detect and locate faults.

Model-based approaches that use statistical analysis and neural networks play an ever-increasing role in the diagnosis of faults in dynamic systems, particularly in those systems where information may be scarce and estimates need to be made. There have been several papers written on this subject. The reader is referred to the surveys detailed in the previous section, where the wide range of topics to which these methods may be applied can be seen. Loukis, Mathioudakis, and Papailiou (1994) showed how, by using statistical pattern recognition, it is possible to optimize the fault detection process. Here, dynamic fault signature measurements are made, from which the statistical properties of the model may be inferred and compared to those made during normal operation (learning phase), allowing decisions to be made by use of a vector system on the strength of the emerging statistical patterns (decision phase). To eliminate the necessity for knowledge of the relationship between estimated system parameters and actual physical parameters, Srinivasan and Batur (1994) proposed a clustering algorithm method to identify faults by first estimating system parameters and then converging them in a fixed-length moving window. A decision-directed clustering algorithm is then included, using mean, covariance, and number of patterns to identify the fault. One main advantage of using this algorithm is that it may be used in an unsupervised learning environment; that is, new faults and operating conditions can be automatically detected by the on-line system. Howell (1994) showed how fault diagnosis could be performed on a nonlinear system that is information poor, such as one with a low frequency of measurement or number of inputs. Two approaches—redistribution (based on the laws of conservation) and minimum number of explanations—are shown to be able to lead to the construction and analysis of candidate sets to provide comprehensive FDI models. An example is given of an information-poor nuclear plant. Wang and McFadden (1996) presented a simulation model for fault detection from bearing vibration using a back-propagation neural network. Impulse defect damage extracted by the high-frequency resonance technique can be readily identified using a two-layer network including one hidden layer, even with a relatively small database.

It can be seen clearly that numerous theoretical models are available for fault diagnosis. However, when these models are tested in practice, they often break down due to their inability to cope with some unexpected or uncertain parameter(s). Improving model robustness with respect to these uncertainties, while maintaining sensitivity, helps to provide the necessary means for inserting FDI models into practical applications. Frank (1994) begins with a survey of the methods of improving robustness in model generation and analysis. The generalized observer scheme (perfect decoupling from modeling errors achieved by increasing the number of inputs), robust parity space check, unknown input observer scheme (state estimation error decoupling), decorrelation filter, and adaptive threshold selection (uncertainties causing residual and decision functions to fluctuate) are all described as means to obtain decoupling from any modeling errors that may occur. Robustness with respect to nonlinear systems is also shown to be attainable. A more general description of robustness and the observer-based FDI approach is given by Patton and Chen (1997), where the passive robust solution (adaptive threshold method) and active robust solutions (uncertainty in residual generation) are considered. Examples are given for the FDI of a jet engine system, a pumping system, and an electric train.

Fault Diagnosis of Rotating Machinery

During the course of the last 50 years, rotating machinery has been studied in more and more detail. A thorough understanding of the principles of rotor dynamics is essential for engineers and scientists involved in the transportation and power-generation industries, as well as many other fields, on which we find ourselves relying to an increasing extent. There have been numerous texts written on the subject of rotor dynamics, notable examples being those by Den Hartog (1934) and Tondl (1965). Den Hartog's book was compiled from material used for a course of lectures as early as 1926, and includes a coverage of basic rotor-dynamic principles, as does Tondl's, with many areas being addressed in much more detail. Several modern dynamics texts also contain sections devoted solely to rotor-dynamic theory, a good example being that of Genta (1993).

Since the analysis and design of rotating machinery is extremely critical in terms of the cost of both production and maintenance, it is not surprising that the fault diagnosis of rotating machinery is a crucial aspect of the subject, receiving ever more attention. As the design of rotating machinery becomes increasingly complex, due to rapid progress being made in technology, so must machinery condition monitoring strategies become more advanced to cope with the physical burdens being placed on the individual components of a machine. Modern condition monitoring techniques encompass many different themes, one of the most important and informative being the vibration analysis of rotating machinery—a topic that has prompted much research to be carried out and a corresponding amount of literature to be produced. Using vibration analysis, the state of a machine can be constantly monitored, and detailed analyses may be made concerning the health of the machine and any faults that may be arising or have already arisen, serious or otherwise. Common rotor-dynamic faults include self-excited vibration due to system instability and, more often, vibration due to some externally applied load, such as cracked or bent shafts and mass unbalance.

Vibration condition monitoring as an aid to fault diagnosis has been examined by Stewart (1976) in much the same way as Smith (1980) and Taylor (1995). Smith covered the general kinds of faults listed above and described qualitatively how they may be recognized from their vibration characteristics, and included effects caused by nonlinearity. Stewart and Taylor also included information on the actual data analysis process—how measured data should be processed in order for a diagnosis to be performed. Downham (1976) gave a broad but comprehensive coverage of recent advances in malfunction diagnosis in rotating machinery by vibration analysis, including a brief history of the main developments in the area over the previous 50 years. Various case studies were outlined, describing such faults as whirl, turbine blade failure, and gear and bearing wear. The general approach to system fault diagnosis was discussed in relation to rotating machinery. Thomas (1984) outlined a typical vibration-monitoring strategy for large (above 500 MW) turbogenerators. He considered the type and frequency of measurements (return to service, on-load, rundown) and the kinds of data analysis methods used in these cases, illustrated by real examples. He also gave a detailed breakdown of the financial benefits to be gained from such a scheme. Göttlich (1988) introduced the idea of the off-line performance map for vibration condition monitoring purposes. The aim of the off-line performance map is to reveal information about the performance efficiency of a rotating machine. A combination of this map and measured vibration data should then enhance the machine's running conditions. The map is constructed relative to data of actual maximum efficiency and design efficiency in nondimensionalized form. By relating performance efficiency data to dynamic and static measurements, it is possible to then control the overall performance level of the machine. Smalley, et al. (1996) presented a method for assessing the severity of vibration (in terms of the probability of damage by analysis of vibration signals) and its related cost using the net present value method. The question of whether to shut down a machine for maintenance was considered, and some guidelines were formulated comparing maintenance and downtime costs against the possible costs that would be incurred by damage.

As condition monitoring systems become increasingly elaborate and complicated, the analysis of data provided by these systems also becomes more in-depth and involved. Hill and Baines (1988) discussed the design of an expert system for the analysis of measured data. An expert system is essentially a computer program able to process input data and then act upon this data to perform an educated diagnosis of a problem. It consists of a knowledge base embodying the knowledge and experience of traditional human experts in the field, such as specialist engineers, machinery operators, and maintenance managers. An inference engine is required to take the knowledge base and operate on the information it receives to perform a diagnosis and arrive at a solution in much the same way as the human expert would. It is concluded that an expert system approach to vibration monitoring may be advantageous, as long as care is taken in the detailed design of the system. If false diagnoses are performed too often, then the cost savings of the expert system may easily be outweighed by the cost of mistakes. By using the Pareto distribution method for machinery diagnostic tests, Cempel (1991) showed that the method he developed for the condition monitoring of tribovibroacoustical processes could be generalized for vibration processes and, therefore, used in vibration condition monitoring. The reliability graph drawn from vibration measurements can be easily transformed to a life curve for a given machine, so that the Pareto distribution lends itself to assessing both the condition of a machine and its residual time to breakdown. Su and Lin (1992) extended a previous vibration model proposed by McFadden and Smith (1984) to describe the bearing vibration caused by a single defect and gave a detailed insight into the analysis of vibration spectra.

Iwatsubo (1976) considered possible errors likely to occur in vibration analysis and how these errors may influence calculations of critical speeds, instability, and unbalance. A statistical approach was used to calculate the mean values and standard deviations of errors, which in turn allows the calculation of statistical values of unbalance response, instability, and critical speeds. The sensitivity of the model with respect to errors in the various model parameters was also determined. It was found that errors in bearing coefficients have a much larger effect on the variance of system instability than do errors in mass and stiffness, which have a predominant effect on the variance of critical speed.

It has already been stated that rotating machinery is becoming increasingly complex: rotors are becoming lighter and faster and tolerances are becoming tighter. With this increase in complexity, it is important to eliminate as many sources of faults as is possible. New techniques are continually being developed to cope with the demand for fault-free machinery. For instance, Halliwell (1996) showed how it is possible to measure torsional vibration—important for the analysis of vibration in gears—with a laser torsional vibrometer, eliminating the need for cumbersome mechanical parts. The laser approach has many significant advantages in practical applications, where vibration measurement has previously caused problems. This is not only because of the practical difficulties of inserting measuring equipment, but because extra mass and stiffness terms are often added to the system by traditional methods.

Sekhar and Prabhu (1995) discussed the effect of coupling misalignment on the vibration of rotating machinery. Shaft misalignment can be a major cause of vibration due to reaction forces generated in the shaft couplings. It is generally accepted that a significant 2× vibration response is a major feature of bearing misalignment. A finite element model of the rotor-coupling-bearing system was developed, and the effect of misalignment was introduced through a coupling coordinate system. The model agrees well with empirical results, where the 1× response is not nearly as significantly affected as the 2× response. By using this model, it is therefore possible to predict the vibration response due to misalignment at the various harmonics—which is valuable in terms of both fault diagnosis and machinery design.

He, Sheng, and Qu (1990) presented a new method for identifying rub failure between the rotating and stationary parts of a machine. The traditional method of spectrum analysis of rubbing is to measure the response spectra. However, a disadvantage of this method is that the rub mechanism may produce noise in certain frequency bands, known as colored noise. The new method proposed, combining principal components and autoregressive spectra (PCAT), was able to identify the chief characteristics of the colored noise. First, an autocorrelation matrix was estimated from the vibration signal, from which the principal components were obtained by use of the orthogonal transformation matrix of eigenvectors. The autoregressive modal coefficients and prediction error are then determined. Computer simulations with noisy data have shown that the PCAT method is an efficient tool for the identification of colored noise in rub failure of large rotating machinery. A thorough analysis of the transient response caused by rotor-stator interaction was given by Ghauri, Fox, and Williams (1996), where the rubbing process was represented by a linear impact model (Coulomb friction). It was shown that, under certain circumstances, sustained rotor-stator contact and reverse whirl could be delayed by asymmetry in the rotor support structure. A low coefficient of friction at the contact interface is also shown to delay reverse whirl. The issue of clearance effects on spiral vibrations due to rubbing was addressed by Childs and Jordan (1997), and it was shown that a clearance at the rub location improves the stability of the system, in particular with regard to unstable spiral vibrations.

Ding and Krodkiewski (1993) showed how static indetermination could be included in the mathematical model for the nonlinear dynamic analysis of a multibearing rotor system. A statically indeterminate system is a system of one or more elements, possessing more supports or constraints than is necessary to maintain equilibrium. A mathematical model is constructed of the rotor-bearing system, which allows for nonlinear dynamic analysis (by introducing nonlinear effects into the bearing terms) and the identification of system configuration parameters. By altering the system configuration, the effect on dynamic performance can be theoretically predicted.

For the diagnosis of anisotropy and asymmetry in rotating machinery, Lee and Joh (1994) developed a method incorporating directional frequency response functions (dFRFs). Anisotropy and asymmetry may cause whirl, fatigue, and instability, as well as influence system characteristics such as unbalance and critical speeds. Complex modal testing was used to estimate the dFRFs. An example was presented showing the proposed method to be very efficient in identifying anisotropy and asymmetry.

Mass Unbalance

Increased running speeds and the requirement for rotating machinery to operate within specified levels of vibration mean that the control of machinery vibration is essential in today's industry. Of the many different causes of rotor vibration, mass unbalance, bowed shafts, and cracked shafts occur most frequently. Procedures for identifying and correcting such faults have developed immensely in recent years, although none as much as those for the identification and correction of mass unbalance. Fault diagnosis methods have tended to move away from relying on the art of human interpretation of changes in parameters, such as noise, vibration, and temperature, and diagnosing faults through experience, to computerized detection and location. With the recent improvements in hardware and software, the diagnosis of faults in rotating machinery has now moved toward on-line computer monitoring and diagnosis, and, in certain cases, automatic correction procedures have been employed. Techniques of balancing rotating machinery to reduce vibration levels have also been the subject of a tremendous amount of research.

Balancing involves placing correction masses onto the rotating shaft so that centrifugal forces due to these masses cancel out those caused by the inherent unbalance mass, thus canceling out vibration. Since, in most cases, it is unlikely that an additional mass can be placed directly in the same plane as the inherent unbalance, special planes, known as balance planes, are often chosen specifically for the purpose of adding balancing weights, especially in larger machines. Balancing is performed on both rigid and flexible rotors, and specific methods have been developed to deal with both cases. Rigid rotors are rotors that exhibit no significant deformation, usually due to a low speed of rotation or a high diameter:length ratio. Conversely, flexible rotors are rotors that undergo substantial deformation while in operation due to their long lengths and high operating speeds. Flexible rotors are often used for the generation of electrical power, an area in which much work has been carried out in the area of balancing. The two main types of balancing are the modal and influence coefficient methods. Modal balancing is the procedure whereby the unbalance forces at each mode considered are canceled out individually. Unbalance planes are chosen and the magnitude of unbalance components is determined depending on the number of modes required by the system. Suitable unbalance masses are then chosen to counteract the forces produced by the unbalance components at each mode. Balancing by influence coefficients involves the selection of correction masses so that vibration is reduced to zero at various, specified shaft locations for various, constant shaft speeds. The actual influence coefficient is the response of the shaft at a given axial point due to a force acting at another (or the same) point along the length of the shaft.

Kirk (1984) illustrated some balancing applications by the use of real practical examples—multistage compressor, cracked shaft, influence of supporting structure—and gave some conclusions and recommendations as a result of this practical experience. Parkinson (1991) gave an extensive coverage of the major aspects of balancing in rotating machinery, and the reader is referred to the references given in his paper for a more detailed history of the development of modern balancing procedures. Parkinson described, in some detail, the process of shaft vibration due to unbalance before reporting on related balancing techniques. Low-speed, rigid balancing was first discussed before a more detailed consideration of the balancing of high-speed flexible shafts was given, including the balancing of a shaft with a bend. International standards were also discussed, as were other more recent developments in the field, such as balancing without the use of trial runs, balancing shafts with a lack of axial symmetry, automatic balancing, and the role of theoretical modeling.

Traditional balancing procedures usually involve the establishment of influence coefficients to determine the balance distribution required to reduce observed vibration. If the number of measured vibrations to be reduced is the same as the number of independent balance combinations, then a set of linear simultaneous equations has to be solved. If, however, this is not the case, then some kind of optimization is required. The difficulty in optimization is in either accurately determining the influence coefficients or in producing a detailed mathematical model of the system, from which the relationship between the measured vibration and the forcing function can be properly estimated. Many methods of balancing use trial weights to determine influence coefficients, whereby weights are placed on the shaft and their effect is observed. In the case of modal balancing, trial weights are categorized in terms of measured or estimated mode shapes. If, on the other hand, a theoretical model is sufficiently accurate, then the unbalance distribution can be determined directly from measured vibration without necessitating the use of trial weights. The principal obstacle in the accurate modeling of a rotordynamic system is in the modeling of the bearings, in particular hydrodynamic journal bearings that are widely used in many applications. Bearing and support characteristics are notoriously difficult to model and often vary with running speed. It is therefore desirable to be able to model a system without knowledge of bearing and support properties and then to identify unbalance using this model in conjunction with measured vibration data.

By using knowledge of the rotor mode shapes, Gnielka (1983) proposed a method to identify the unbalance in an initially bowed flexible rotor without the use of test runs. The only a priori knowledge required is the flexural mode shapes and the generalized masses. The unbalance is identified by measuring the shaft response near a critical speed and solving the differential equation of motion for the bowed shaft using the frequency response function. Knowledge of mode shapes and generalized masses are included, and, using an iterative procedure (because the system is nonlinear), the unbalance is identified from the forcing vector by using the least squares method. An approach for the modal balancing of flexible shafts without the use of trial weights was also suggested by Morton (1985). Morton showed how it was possible to separate shaft and bearing characteristics (which then enabled shaft response functions to be calculated) to identify bearing parameters from a known shaft forcing. This was achieved by describing the bearing behavior in terms of the free modes and the bearing supports using frequency dependent impedances restricted to translational motion; this led to a steady-state equation of motion containing bearing behavior and, more importantly, shaft vibration. Morton's technique for balancing without trial weights can be applied to rotor systems possessing any number or type of bearings.

Lee and Kim (1987) presented a modal balancing method to be performed while the rotor is in operation by use of a balancing head. Research had previously been carried out on this method, but only for low-speed balancing of rigid rotors, not with high-speed modal balancing. A balancing head is a disk or combination of disks mounted on a shaft that carries correction weights. Vibration levels are measured, and the correction weights are moved, during operation, to reduce these levels. The operation of the disk and its control system is an important feature of the method. A manually operated controller controls the unbalance response by continuously monitoring the shaft whirl orbits on an oscilloscope and transmitting signals to the head in terms of both magnitude and direction. Magnitude change is achieved by rotating the two disks in opposite directions; direction change is achieved by rotating the two disks together after the magnitude has been set. The method was shown to be successful by a comprehensive set of experiments in which it was shown that the rotor could be modally balanced very effectively by the head, during operation, at speeds of up to at least 50 Hz. Lee, Joh, and Kim (1990) developed this model and included computer control of the balancing head. Again, experimental results showed that the method worked effectively for the high-speed modal balancing of flexible rotors.

Tan and Wang (1993) developed a theory unifying the modal-balancing and the influence coefficient methods. The unified theory was applied to the low-speed balancing of flexible rotors. It was shown that the modal and influence coefficient balancing techniques could be made equivalent by setting the initial unbalance deflections in both methods to zero. It was then proposed, as long as certain requirements are met, to balance a rotor at low speeds (influence coefficient method) while meeting the requirements of high-speed (modal) balancing. That is, the rotor may be balanced for however many modes are required without the need to run at or above its operating speed, which has obvious advantages for practical applications. It is important to recognize that the proposed method is distinct from low-speed rigid rotor balancing; it is an actual unification of the modal and influence coefficient techniques, applicable to flexible rotors at low speeds. Krodkiewski, Ding, and Zhang (1994) presented a method for identifying both the plane and magnitude of the balance weight required to correct a system whose configuration and unbalance distribution may change while in operation. The method uses a nonlinear mathematical model of the system and the measured vibration response both before and after the change in unbalance occurs. The nonlinear model includes information about the bearings and supports as well as about the shaft itself. The periodic hydrodynamic bearing forces are represented by a Fourier series. The model uses measured data before and after the change in unbalance, and, incorporating an error function, the unbalance plane and magnitude are determined by trial and error until the algebraic equations that represent motion before and after the change in unbalance are solved. A numerical example verifies the identification method, which can be developed for applications such as the loss of a turbine blade or the inclusion of a change in system configuration. Lees and Friswell (1997) presented a method to estimate the unbalance configuration in a flexibly mounted rotor-bearing system from vibration measurements at the bearing pedestals. The only requirement of the method is an accurate model of the rotor. If bearing information is available, then this is incorporated; otherwise, absolute shaft displacement measurements are used, and no knowledge of bearing behavior is necessary.

Bent Shafts

The phenomenon of shaft forcing due to an initial bend has aroused interest over the last 20 years or so, albeit much less than mass unbalance. Bends in shafts may be caused in several ways; for example, due to creep, thermal distortion, or a previous large unbalance force. The forcing caused by a bend is similar, although slightly different, to that caused by conventional mass unbalance. There have been numerous cases in industry where vibration has been assumed to have arisen from mass unbalance and rotors have been balanced using traditional balancing procedures. This has repeatedly left engineers puzzled as to why vibration persists after balancing and why vibration levels may indeed even be worse than before balancing took place. In summary, shaft bow response is a function of shaft speed and causes different amplitude and phase angle relationships than is found with ordinary mass unbalance, which is a function of the square of the speed. It is important to be able to diagnose shaft bow from vibration measurements and thus distinguish between it and mass unbalance.

One of the first extensive investigations into shaft bow was made by Nicholas, Gunter, and Allaire (1976a, 1976b) in a series of two papers. Part 1 ("Unbalance Response," 1976a) discussed the unbalance response of flexible rotors due to shaft bow. Part 2 ("Balancing," 1976b) proposed a balancing theory and gives experimental results for the balancing of a flexible rotor with shaft bow. The theory developed in Part 1 describes the dynamic response of a shaft with both mass unbalance and shaft bow in terms of the response amplitude and phase angle. A broad range of numerical tests were performed to compare responses by looking at the amplitude and phase angle behavior under various conditions: different damping ratios, small and large residual bow, and residual bow equal to unbalance eccentricity. One important difference drawn from these tests is that the change in phase angle from rest, up to the speed where maximum rotor amplitude occurs, is not usually 90° as it is with conventional unbalance. Also, when the bow is 180° out of phase with the unbalance, there always exists a speed at which the rotor amplitude is zero. In Part 2, three different influence coefficient based balancing approaches are considered. Method I calculates the magnitude and direction of the required balance weight by equating, at an arbitrary balance speed, the complex amplitude of motion with the complex unbalance eccentricity, reducing the total shaft deflection to zero at the chosen balance speed. Method II takes into account a large shaft stiffness, compared to mass and damping characteristics of the shaft, in which case the residual bow influence coefficient can be assumed to be unity. The residual bow is subtracted from the rotor amplitude, minimizing elastic deflection. After balancing, the rotor response is reduced to the residual bow at the balance speed, although a small response amplitude remains when operating near a critical speed. If the shaft critical speed is known, then Method III may be applied. This involves calculating the complex balance eccentricity vector in terms of the unbalance and shaft bow vectors by setting the rotor amplitude to zero at the critical and solving. The rotor is then perfectly balanced at the critical without actually having to run the shaft at this speed; the only requirement being that the critical speed is already known. Method III is evidently the optimum balancing method, but if the critical is unknown, then a combination of Methods I and II can also yield effective reductions in response amplitudes.

Salamone and Gunter (1977) investigated the unbalance response of a rotor with a skewed disk in addition to shaft bow. Transfer matrix equations were developed to include these two effects. The skewed disk introduces both angular motion and gyroscopic moments into the system. The responses of rotor systems with combinations of disk skew and shaft bow were simulated, and excellent agreement with previously reported results was shown. At the first critical, it was seen that the response due to the bow was predominant. Both shaft warp and disk skew contributed in equal amounts at the second critical speed, and the skewed disk had the greatest effect at the third critical. It is important to note that at all three critical speeds, the predominant excitation was not that of unbalance. Therefore, a response analysis that considers only unbalance is very unlikely to be accurate if shaft warp and/or disk skew are present. Salamone and Gunter (1980) then included the effect of an overhung rotor on a shaft with disk skew. It was found that, at the first critical speed, a rapid decrease in phase angle at the far bearing was an indication of disk skew. Three stages of balancing were considered in the analysis:

1. A single plane disk correction at the critical speed, calculated from amplitude and phase angle measurements at the far bearing location. Response amplitude was found to be acceptable at the critical speed but soon increased at higher speeds.
2. Far bearing amplitude reduced to zero at a frequency ratio of 3 by determining the influence coefficient for the bearing plane. This method reduces the response at super critical speeds without disturbing the first balancing correction.
3. Near bearing amplitude reduced to zero at a frequency ratio of 3. This unfortunately increases the amplitude at the first critical and at the far bearing. To compensate for this, balance with a couple correction is introduced that comprises two correction weights separated by the thickness of the disk to produce an equivalent balance moment vector.

For small angles of skew, it was shown that these two correction weights are equivalent to a uniform, skewed disk. This balancing correction is successful in reducing response amplitude to zero at the far and near bearings, as well as at the disk, throughout the speed range considered.

Flack, et al. (1982) included the effects of shaft runout, either electrical (material nonuniformity) or mechanical (out-of-roundness or scratches) on bowed shafts with mass unbalance. A compensation effect was introduced to eliminate runout effects, which simply subtracted the runout vector from the total response vector, since runout is constant with speed. Using such an effect to compensate for shaft bow introduces errors, since the bowed shaft response is a function of speed and thus variable. Theoretical and experimental results were presented for various combinations of mass unbalance, bow, and runout. It was shown that the runout did not exhibit the same effects on phase angle change as the bow, described above. For a shaft with bow and runout, using runout compensation, the response was that of a bowed rotor. The experimental results shown compared much better to the theory of a bowed rotor than they did for a rotor with runout.

Parkinson, Darlow, and Smalley (1984) described the differences in whirl experienced by a rotating shaft subject to shaft bow and mass unbalance. By balancing the net whirl (total whirl minus shaft bow), which contains certain components due to mass unbalance and others due to shaft bow, the residual total whirl at running speed is approximately equal to the shaft bow. For practical applications, this is more desirable than simply balancing the total whirl, which leaves a residual bow at noncritical speeds and is balanced only at resonance. The net whirl is also easier to balance, since the same procedure can be used as for mass unbalance. In addition, the response seen after balancing the net whirl shows conventional resonance behavior in terms of amplitude and phase angle. Experimental results were included that confirmed the above findings and showed the balancing of net whirl to be an extremely effective method of balancing a bent shaft.

Meacham, et al. (1988) developed a procedure for the complex modal balancing of shafts with an initial bow similar to those described above. However, although the aim of the balancing procedures was the same, the modal balancing method (instead of influence coefficients) was used to achieve the same end. Method I considers the unbalance and shaft bow acting as an equivalent unbalance modal participation factor (UMPF) at the chosen balance speed. In Method II, the quasi-static rotor response is subtracted from the total response of the system at the constant speed chosen; the magnitude and position of the balance masses can then be determined. Method III uses measured response data from a neighboring speed to reduce vibration amplitude at the critical. A curve-fitting procedure determines the residual bow effect from which the required balance corrections can be found, assuming the modal vectors of the system are constant over the range of measuring and critical speeds. No test runs are required for the proposed method, and an extensive set of numerical results was included, verifying the proposed modal balancing procedure.

Cracked Shafts

Another important fault in rotating machinery is cross-sectional crack propagation in a shaft, generally induced by fatigue. Although the problem has not been addressed in as much detail as mass unbalance, a considerable amount of research has been undertaken in this area. Vibration is caused by the presence of a crack, and it is found, in general, that the vibration amplitude is dependent on the depth and shape of the crack and on the position of the crack in relation to the shaft mode shape. Therefore, it is important to be able to diagnose cracks early on to carry out any necessary maintenance before more damage than is necessary is caused by this excess vibration. The presence of a crack causes asymmetry in the stiffness of the rotor, which is difficult to detect under normal running circumstances using customary monitoring techniques. Another difficulty is encountered when examining a "breathing" crack, which is a crack that opens and closes as the shaft rotates. This means that the shaft stiffness properties actually vary during the course of one revolution as the crack opens and closes. The modeling of crack behavior has been investigated by several authors since the 1970s, and some of the most relevant research is reported below. The reader is referred to a comprehensive literature survey of various crack-modeling techniques and system behavior given by Wauer (1990).

Mayes and Davies (1976) gave an extensive insight into the effects of a transverse crack on a rotor-dynamic system. The crack was theoretically modeled using the virtual work principle, applying the conservation of energy to describe crack behavior. The stress distribution, given by the stress intensity factor, and knowledge of the bending moment around the area of the crack are prerequisites for this approach. It was found that the equations of motion for the system are nonlinear due to the unbalance forces acting on the crack, which cause the crack to periodically open and close. The theory provided showed that the response due to the crack may either be zero (unbalance in phase with crack) or may have two distinct synchronous responses (unbalance 180° out of phase with crack). In addition, it was shown that the response due to the crack is the same as the response due to adding an extra unbalance force of predetermined magnitude and angle to the original unbalance. Experimental results proved that when the phase of the unbalance is between –45° and 135° with respect to the crack, the crack is most significant. Outside these limits, the shaft tends to behave as if it were uncracked. The 2× response is dependent on the unbalance force vector and takes a maximum value when the unbalance and crack are in phase. Gasch (1976) described cracked shaft behavior by simplifying the crack to a model whereby the system stiffness was represented by a time-varying flexibility matrix by transforming rotating coordinates into space-fixed coordinates. The breathing crack was modeled by including additional stiffness terms for the crack for the period when the crack is open and discarding them while the crack is closed. Numerical results exhibited characteristics similar to those of a shaft with asymmetric stiffness properties: a resonance at a frequency ratio of 1:2 and instability just below the first critical speed. In addition, resonances were also seen to occur at frequency ratios of 1:3 and 2:3. It was found that the position of unbalance forces relative to the crack could either open or close the crack.

Nelson and Nataraj (1986) gave a broad survey of previous work on cracked shafts before proposing their own new method in which the shaft is modeled using finite elements. The crack element assumes a breathing crack model, which produces an effective second moment of area around the crack while it is open. A switching function was incorporated to represent the actual opening and closing of the crack, which is periodic and represented by a Fourier series. The function has a value of one for the crack being open and zero for the crack being closed—such a function describes what is known as a switching crack. Only the first few terms of the Fourier series were included in the analysis, and results compared well with previously published analytical and experimental work. One previously unreported phenomenon, however, was that the sign of the minor axis for each harmonic is dependent on shaft speed. This means that as the shaft speed varies, the harmonics experience reversals in the direction of whirl. The same kind of beam element was also used by Schmied and Krämer (1984), although in this case only a simple switching function was used, and similar findings were reported. Bachschmid, Diana, and Pizzigoni (1984) obtained similar results, albeit with nonlinearity taken into account in the time dependent stiffness matrix representing the crack.

Jun, et al. (1992) presented a crack model developed from traditional fracture mechanics theory. Cross-coupled stiffness terms, as well as the direct stiffness terms used exclusively by the aforementioned authors, were incorporated into the analysis to represent the behavior of the crack when only partially open. Using only direct stiffnesses, the crack may only be fully open or closed. In reality, however, there must exist a period of time when the crack is only partially open. The cross-coupled method therefore more adequately represents a breathing crack than the on-off switching model. It was found that a crack could best be identified at 2× rotational speed. This is because the response due to unbalance at this speed is much less significant than at 1×, leaving the response due to the crack much more identifiable. Gasch (1993) introduced a perturbation method into his analysis, allowing cross-coupling stiffness and dynamic response terms to be excluded under stipulated weight dominance circumstances. A very thorough analysis provided suggestions for the detection of cracks, such as the long-term monitoring of mean additional static deflection and trend analysis over long periods. It was reported that a crack may be recognized by analyzing the corresponding vibration response signal. The 1×, 2×, and 3× response amplitudes will all increase in direct proportion to crack size.

Huang, Huang, and Shieh (1993) used local flexibility coefficients and the principle of energy changes to examine the behavior of the shaft over one revolution using a time-varying stiffness approach to represent the breathing crack. The matrix defining the crack is determined by measuring the crack-released energy for various crack orientations in the shaft. Again, it was found that the response was proportional to crack depth. Subcritical responses were also reported. Seibold and Weinert (1996) used banks of Kalman filters to detect the position and depth of a crack whereby each filter represents a different damage scenario. The crack can then be diagnosed by assessing the probabilities of each of these different scenarios. Meng and Hahn (1997) investigated the diagnosis of cracks in the subcritical and supercritical speed ranges using cross-coupling stiffness terms for the purpose of on-line monitoring schemes. Through a series of numerical tests, it was found that the model used agreed with predictions of the above authors and that a crack caused backward whirl of the rotor, the amplitude of which is proportional to crack depth.

Conclusion

This paper has provided a broad review of the state of the art in fault diagnosis techniques, with particular regard to rotating machinery. It has been seen that new model-based fault diagnosis systems are being developed rapidly in order to meet the demand for increasingly intelligent condition monitoring systems for the maintenance of modern industrial processes. It is anticipated that future developments in fault diagnosis systems will be heavily concerned with the improved design of expert systems and neural networks for the continuous monitoring of machinery. On-line condition monitoring strategies will become increasingly commonplace in a greater range of systems as computer hardware and software become more widely available, more robust, and less expensive. Operating tolerances will become tighter, and the cost of shutting down a machine for maintenance will tend to be assessed against the cost of failure and lost production.

Condition monitoring techniques for the diagnosis of faults in rotating machinery need to be improved in order to be able to identify, as quickly as possible, the many different kinds of faults that may occur in a rotor-dynamic system. Vibration response measurements yield a great deal of information concerning any faults within a rotating machine. The identification of common mass unbalance by vibration analysis is very well developed and can be performed in many ways. However, the identification of faults such as bowed or cracked shafts, rubbing, and bearing misalignment remains relatively basic in comparison. These areas still require much research to be carried out for comprehensive fault diagnosis schemes to be devised, which can automatically detect any faults that may arise in a system and give information on the best correction procedure to be used.

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