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MACHINERY DIAGNOSTICS
VIA MECHANICAL VIBRATION ANALYSIS
USING SPECTRAL ANALYSIS TECHNIQUES

by

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Mechanical vibration analysis affords a reliable means to selectively identify specific machinery faults. As such, it plays a key role in diagnostic work on individual units and in progressive maintenance monitoring programs where substantial diagnostic and prognostic capabilities are considered essential. A physical machinery diagnostics model was developed that was designed to incorporate some of the more common machinery faults found in rotating machinery relating to shaft, bearing, gear, and alignment defects. The results of spectral analysis techniques used to detect these simulated faults are displayed and discussed with special emphasis on gear train diagnostics. Also included are a description of one of the current U.S. Navy machinery vibration monitoring programs, and an initial study regarding a proposed technique for providing a graphic display of gear faults.

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Machinery Diagnostics
Via Mechanical Vibration Analysis
Using Spectral Analysis Techniques

by

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Submitted in partial fulfillment of the
requirements for the degree of

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ABSTRACT

Mechanical vibration analysis affords a reliable means to selectively identify specific machinery faults. As such, it plays a key role in diagnostic work on individual units and in progressive maintenance monitoring programs where substantial diagnostic and prognostic capabilities are considered essential. A physical machinery diagnostics model was developed that was designed to incorporate some of the more common machinery faults found in rotating machinery relating to shaft, bearing, gear, and alignment defects. The results of spectral analysis techniques used to detect these simulated faults are displayed and discussed, with special emphasis on gear train diagnostics. Also included are a description of one of the current U.S. Navy machinery vibration monitoring programs, and an initial study regarding a proposed technique for providing a graphic display of gear faults.

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I. INTRODUCTION

A. GENERAL

Vibratory motion is a phenomenon inherent to all types of machinery regardless of their material condition or state of performance and is typically measured in terms of either the physical response of the machine itself or the sound produced by the vibratory motion that is induced. Thus, the vibrations in general may be separated, respectively, into two categories; mechanical vibrations and acoustic vibrations (termed machinery noise[1] by many authors). Machinery diagnostics involves the measurement and analysis of various phenomena associated with machinery operation and is specifically aimed at the detection and identification of machinery faults. Vibrational behavior is a prime indicator of machinery condition and so plays a key role in machinery diagnostics and health monitoring. Mechanical vibration measurements are favored as the measures of merit in the evaluation of machinery condition and performance, whereas acoustic vibration measurements, although applicable to machinery condition assessment, have greater importance and more widespread use in work regarding noise control and noise reduction analyses. In this regard, Lyon [Ref. 1] states

> The greatest difference between diagnostics and noise reduction lies in their respective goals. A machine operating properly and without faults can still be very noisy, and a machine that has developed a major fault may operate quietly.

Consequently, acoustic vibrations are addressed in part, but the focus of this paper remains on the measurement and analysis of mechanical vibration signals with special emphasis on dynamic signal analysis techniques as they apply to machinery diagnostics and machinery maintenance programs.

Initially established as one of the primary goals of this thesis work was the development of a machinery diagnostics model to satisfy two specific objectives. First, to provide a working model which could be used to simulate some of the more common machinery faults and fault detection techniques that were to be discussed; and second,

1 Throughout this paper, the term noise is specifically reserved for referring to those portions of a vibration signal which come from signal contamination by non-machine related sources and, hence, undesirable and of no interest except for the degree to which its presense may affect the detectability of the signals of interest.

to create the means by which a new analysis method and display technique regarding gear defects could be experimentally tested and evaluated.

This paper summarizes and discusses some of the measurements and current techniques employed in mechanical vibration signature analysis as applied to detailed diagnostics conducted on individual machines and to machinery maintenance programs in general. The remainder of this introductory section provides background material regarding vibration analysis and its applications to machinery diagnostics. The later sections discuss, in order, one of the current U.S. Navy surface fleet maintenance monitoring programs, basic signature analysis measurements and methods, the diagnostics model that was developed, a special study on gear train diagnostics, and lastly, a summary of conclusions and recommendations for continued study in this field.

B. BACKGROUND AND APPLICATIONS

1. Why Vibration Analysis

Both consciously and subconsciously, operating engineers routinely use at least four of the five human senses to varying degrees to assess the condition of the machinery under their care. Sight, hearing, touch, and (although to a lesser degree) smell are useful in monitoring overall plant status, but sight and smell become virtually useless in evaluating the condition of an individual machine until long after an abnormality, or an abnormal trend, has become quite obvious. Normally they are limited to detecting the existence of a problem which has advanced to a point where some form of corrective action is called for without delay, whether it be as minor as a simple adjustment that may be made with the unit running, or as major as the immediate shutdown of a piece of equipment. On the other hand, hearing and touch are more sensitive to small variations in operating conditions and, with respect to an individual machine, the onset of specific problems such as pump cavititation, bearing defects, drive belt defects, mechanical looseness, and the like, may be detected. In general , though, what is being felt or heard is the vibration, or the sound produced by the vibration, which results from some specific change which has occurred in either the operating parameters or the material condition of the machine components or their alignment. Unfortunately, even with the significant dynamic range and filtering capabilities of the human ear which enable the selective identification of small signals (sounds) in the presense of large random signals (background noise), the changes so detected may be due to specific maladies or component defects which are well on their way toward necessitating an unscheduled shutdown for repairs. Consequently, vibration analysis is a logical choice of a field to explore and in which to develop measurement and analysis techniques which can serve as natural extensions of those basic human senses which are inherently more responsive to, hence more informative about, machinery condition.

2. The Basic Concept

Vibration analysis is based upon the concept that once a machine is placed into service and a baseline vibration signature[2] is obtained, any subsequent change in its operating parameters or material condition will be reflected by a change in its vibration signature. More importantly, the converse is assumed to be true, that if there is no change in its signature, then there has been no change in the operating parameters or material condition of the machine or any of its components. This basic concept is well-established both in theory and in practice. In accordance with its strict definition, the use of the term signature implies a uniqueness, and rightly so. Much of the analysis work done relies in part upon pattern recognition in order to categorize the type(s) of machinery fault(s) present, and this is commonly the first step taken in any analysis procedure.

Under controlled laboratory conditions or simulations using artificially produced signals, the duplication of results for identical conditions is not difficult to obtain. This is evidenced by several of the figures which appear later in which a signal generator was used to provide the input signals; each figure developed this way is referenced as such. In actual practice in the field, point-for-point reproduction of signatures for repeated conditions are not truly expected nor obtained; however, the same conditions will yield signatures which are unmistakably "the same". Indeed, it is precisely this high degree of repeatability that promotes high confidence and popularity in the use of vibration analysis for machinery diagnostics and for short and long term maintenance monitoring needs. Mechanical vibration analysis covers a wide assortment of instrumentation, methods, and techniques used to collect and analyze vibration data either in its raw form or after it has been processed to display it in alternate forms which facilitate the detection of specific machinery faults. Where vibration monitoring is implemented as part of an overall machinery maintenance program, the more useful, hence more common, forms of display are in the frequency domain. The equipment required varies from simple hand-held meters, to sophisticated multi-channel solid-state signal analyzers which offer a multitude of data processing functions and display

2 The term signature refers to any graphical display of signal information regarding a vibration parameter; velocity amplitude vs. time, acceleration amplitude vs. frequency, etc.

formats, to systems where permanently installed sensors feed into a computer equipped with diagnostic software.

3. Application to Machinery Maintenance Programs

Machinery maintenance is well-recognized as accounting for a substantial portion of overall plant operating cost. Small advances in maintenance programs reap large benefits, not only in cost savings, but also in improved safety, availability, efficiency, etc. Success stories are abundant in the literature on this subject, and payback periods for the initial investments are typically reported in terms of months. Consequently, there are many people in various engineering disciplines actively pursuing improvements to existing methods and technologies of machine condition and performance analysis, efforts which are extensive and well justified.

Maintenance programs may be viewed as belonging to one of several hierarchal levels based on whether or not a structured maintenance plan is in effect and on the degree to which a plan incorporates the aforementioned methods and technologies that are available which relates directly to the intended goals of the plan.

a. Crisis Maintenance

The lowest level program is popularly referred to as a crisis maintenance program. This is one where no specific monitoring, performance evaluation, condition evaluation, or maintenance routine exists. Such a program relies strictly upon observations which may be made by the operators for gaining any forewarning of trouble, with the result that machines and/or their components all too often degrade to a state of being unfit for service before any maintenance or repair efforts are undertaken. Note that this is not a criticism of the performance of operating engineers, but rather a statement on the general inadequacy of the naked human senses to detect machinery degradation at a sufficiently early stage to prevent serious problems, as well as a statement on how rapidly some machinery faults may develop and grow to unacceptable limits.

b. Preventive Maintenance

The next program level is that of the preventive maintenance program in which maintenance is carried out on a regular schedule which is based upon a specified time interval, a specified number of operating hours, or some other measure of machinery operating life. An example of this would be automotive maintenance schedules which typically specify a limitting number of months or miles driven between maintenance work. This is certainly an improvement on the previous level, but the quality of this type of program relies upon the ability to accurately determine the optimal interval(s), and it presumes that all machines of the same class (thus assigned the same intervals) will degrade identically in fashion, rate, and amount. As accurate as they may be, the assigned intervals, at best, can reflect only statistically averaged measures of what have proven to be acceptable intervals. In order to be even minimally conservative in avoiding serious outages, these intervals result in wasted time, money, and effort in servicing those units which are performing above the average. In short, the intervals will only be optimal for those units which degrade exactly as does the average unit of the class. Those which perform below average may fail before the maintenance is done as well as cause premature development of faults in related components; those which perform above average receive unnecessary maintenance and also invites the added risk that their condition may actually be worsened by the maintenance, especially if it involves opening of the unit and/or routine replacement of components. There is also the consideration that, to be cost effective, such a program must be limitted to addressing a finite number of the higher probability faults which has the effect that the program may essentially be blind to many problems that can lead to chronic trouble and repetition of work that only treats the symptoms.

c. Predictive Maintenance

The final level is that of the predictive maintenance program in which, as the name implies, machinery faults are detected at the early stages of their development so that maintenance needs are able to be predicted, with the result that maintenance is performed only when it is needed and only on those components which need it. Figure 1 shows what is commonly referred to as the "bathtub curve" which displays how wear rate varies with time for most machinery components. Specifically, it shows the run-in period characterized by a decreasing wear rate as the initial manufacturing and assembly imperfections become smoothed out, followed by a period of normal operating wear at a constant rate, followed by a period of increasing wear rate due to creation and growth of defects until final failure occurs. Time-based preventive maintenance programs may interrupt the service life of a component which may still be in its normal wear rate period, whereas predictive maintenance programs will only remove a unit for maintenance when it is in its final stage of serviceable life, i.e., somewhere in the increasing wear rate portion of the curve. Close monitoring and experience can provide very good estimates of projected time to failure which allows the programs to realize optimal maintenance intervals on a machine-specific basis.

Although predictive programs cannot assure that unexpected failures will not occur since some faults may still develop and proceed to failure too rapidly, they can

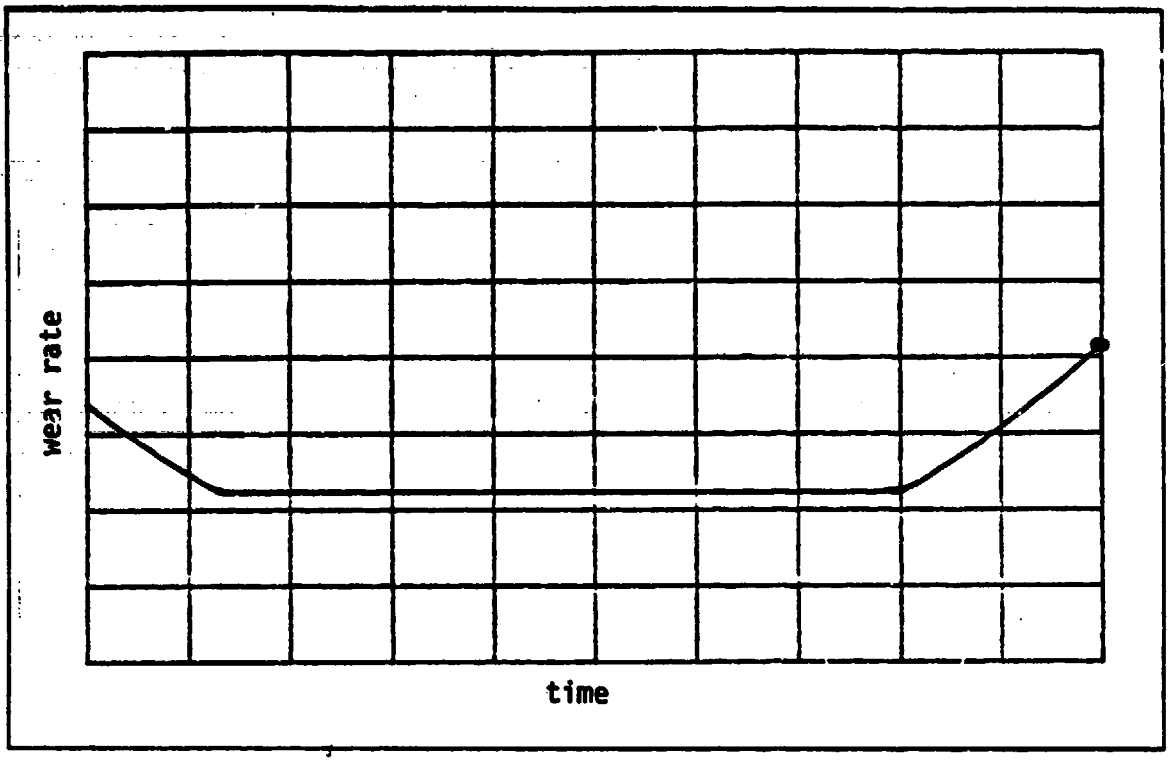

Figure 1. Component wear rate versus time.

and do successfully avoid unnecessary maintenance work and expense. Additional benefits of these programs include increased productivity due to longer operating time between repairs, reduced spare part inventories and reduced repair times since the exact components in need of repair or replacement are identified long before the work is commenced, and the ability for advanced planning and scheduling of service interruptions rather than their unannounced arrival which is an attendant problem of the other program types.

The early detection of faults which enable this type of program to be truly predictive is made possible by a variety of sophisticated technologies which most of the current literature treats under the general headings of condition monitoring or health monitoring. Predictive programs vary in the degree to which they employ these technologies. The simpler programs generally incorporate periodic measurements of vibration data and performance data, along with basic analyses of system fluids such as lubricating oils, coolants, or working fluid media. The more advanced programs use continuous monitoring schemes employing permanently installed sensors which give continual on-line measurement of condition and performance parameters and feed these directly to central computers and automatic data logging devices. The repeatability and accuracy with which mechanical vibration analysis techniques can identify specific machinery faults has also led to the demand for their use in the development of expert systems with artificial intelligence where the signals are automatically analyzed and the fault condition is automatically outputted to the operator. For simple systems, this replacement of the vibration analyst with a computer that can decode the signal may not be too difficult. Unfortunately, most signals are too complex and require the analyst to employ methods which vary from case to case. Even identifying the same type of fault may require different techniques that change from one machine to the next. In short, the procedures that are followed by the analyst are not structured to the point where there are universally aplicable to all diagnostic work. This is primarily what retards progress in the full development of artificial intelligence applications for these expert systems.

Most of the literature on the subject tends to reserve the term *condition monitoring* for use in specifically addressing predictive maintenance methods, although Ilvonen [Ref. 2] applies it to both preventive and predictive programs. In the mid 1970's, the conventional preventive maintenance programs were beginning to give way to predictive maintenance theories and practices. As originally established, these new programs were essentially two-headed. The overall program was termed "On-Condition Maintenance" which meant that units of equipment would only receive maintenance service as needed based upon their condition. The term "Condition Monitoring" was applied to one facet of the program which monitored the condition of units of secondary importance, but allowed them to remain in service until they failed. This was done in order to generate a data bank of machine vibrational data which could help establish what vibration levels were to be considered normal and abnormal. This was necessary at the time because of the limited amount of statistical data available regarding machinery vibrations. [Ref. 3]

As the term is currently used, it involves the acquisition of information which is used for evaluating the condition of machinery. The wide variety of instrumentation and analysis techniques employed were classified by Mathew [Ref. 4] into six main categories: aural, visual, operating variables, temperature, debris monitoring, and vibration monitoring. The first four generally include conventional methods which have long been in use except perhaps for some of the newer technologies such as boroscopic exams, thermography, and acoustic monitoring. Debris monitoring includes both oil and gas path monitoring. In wear debris analysis, oil samples from sumps and/or samples from magnetic plugs strategically placed in the lubrication system are analyzed using ferrography and spectrometry in order to determine the concentration and type of debris found. This information can then be used to determine the rate of wear of specific components. The gas path debris analysis similarly attempts to isolate the source and nature of the debris, specifically, in order to differentiate between fuel contaminants or combustion products and actual material debris such as from eroded or broken blading. Also included is a collection of process variables data such as flow rates, pressures, temperatures, etc. which are conventionally obtained and recorded, often with the use of an automatic data logging device. No one information group is able to stand alone and provide sufficient data to effectively run a maintenance program. In the more advanced programs, all of the aforementioned work in concert to provide operators and management personnel with status and long run trend information which is used to establish maintenance schedules. Each serves to supply system information which the other cannot.

4. Other Applications

Vibration analysis has been applied in all facets of engineering including design, manufacturing, operations and control, maintenance, and in surveys, inspections and tests. In design, much of the work is related to mathematical modelling and model testing; actual field work in vibrations have their impact on design by providing feedback on the service performance which, at times, may call for redesign to adequately correct a certain problem. Also, it provides more accurate estimates of such statistical data as mean time between failures, maintenance downtime, and other associated time, material, labor and cost figures used by the design engineers to do systems analysis and life cycle cost studies. In manufacturing, vibrations of certain components such as lathe spindles, cutting tools, etc. are monitored to control tolerances of manufacture and to indicate to the operators when the components should be replaced or renewed in order to maintain the required tolerances, and vibration readings are used in quality control applications of the manufacturing process as well. Vibration signals have been incorporated into control system designs to provide feedback on system performance or response to operational changes. And vibration readings are also being used more and more as part of machinery service and installation acceptance criteria.

II. U.S. NAVY MACHINERY VIBRATION MONITORING PROGRAM

A. PROGRAM OVERVIEW

The U.S. Navy conducts several different vibration monitoring programs for the machinery onboard its surface fleet. A total of approximately 32,000 machinery units are involved, with about 12,000 of these being monitored under the Systems and Equipment Maintenance Monitoring for Surface Ships (SEMMSS) program; the exact number of units monitored on any given ship depends upon the vessel class. Of the various programs, the SEMMSS program is of particular interest because it is an example of an increasingly common event occurring in many industries; namely, the replacement of a preventive maintenance program with a predictive maintenance program in which mechanical vibration monitoring was relied upon heavily to make the transition. The SEMMSS program is administrated by the Naval Ship System Engineering Station (NAVSSES) located in Philadelphia, Pennsylvania. Initial studies were commenced to access the program's effectiveness; in particular, the effect it has had on overall maintenance cost and ship availability figures. Although final reports are not yet available since the study is still underway, discussions with program administrators indicate that its impact should prove to be extremely positive. NAVSSES functions to provide technical guidance, including the training of the personnel who perform the task of data acquisition in the field. It also sees to the selection, procurement, disposition, and maintenance of the field monitoring instrumentation; and it is responsible for overseeing data processing procedures and the development and distribution of documentation and reports of survey results. Some of these tasks are performed by a private engineering firm specifically contracted to do this work.

The program currently calls for vibration readings of each unit to be recorded and analyzed every three months. With the number of units involved and their geographic distribution, the attendant logistics problem and the need for minimal interruption of vessel operations were dealt with by the establishment of Performance Monitoring Teams (PMTs). The teams are based in the vessels' home ports, equipped with portable data acquisition kits, and are responsible for field testing, data collection, and transmission of the data to the analysis center. The field offices have dynamic signal analyzers for local analysis needs and for conversion of the analog data to digital data which can then be transmitted via modems over commercial telephone lines. The analysis center handles the data processing and analysis, the permanent data storage, and the report preparation phases, then furnishes NAVSSES with the final written reports of the survey results. The vessels each receive a copy of the report on their machinery which includes a summary of which units were tested, which ones were not tested and why[3] , a prioritized listing of machinery in need of attention along with specific recommendations on what work should be expected, and an updated copy of the vibration severity history of those units which were tested.[Ref. 5]

B. DATA ACQUISITION

Each machinery unit is fitted with two or three small transducer mounting discs made of 416 stainless steel, secured to the casings in way of the bearings using a high-strength epoxy compound, each covered with a removeable protective cap. The choice of disc material provides pieces which are resistant to the adverse environmental conditions and allow for magnetic attachment of the transducers. Accelerometers fitted with magnetic bases are used exclusively with a silicon-based grease applied as a lubricant/couplant which protects the machined surfaces of the disc and magnetic base while improving the transmissibility characteristics of the arrangement. The suitability and performance of this method of attachment was verified by a special study which compared transmissibilities for various arrangements to the quality obtained with stud-mounted assemblies [Ref. 6]. The number and location of monitoring points are unit class specific; i.e., all identical machinery units throughout the program have identical transducer mount locations. Most units have one radial fitting at each load bearing (in the same plane) and one axial fitting at one of the bearings, normally the one nearer the coupling. Exceptions to this are centrifuges and purifiers which only have radial fittings. The prescribed location of all discs is cataloged in a written program guide which allows the PMT technician to accurately locate any mounts which need to be replaced or reaffixed. This, along with the fact that they are "permanently" affixed, supports strong confidence in the validity of measurement comparisons made between monitoring periods as well as between units of the same class. Of course, the impedance of each mount location will be different from one machine to the next, but at least the number of variables is somewhat reduced by ensuring that those readings which are to be compared to one another are at least taken at precisely the same location on each unit every time.

[3] Typical reasons for not testing include no water available, boilers secured, maintenance/repairs in progress, etc.

Low impedance coaxial cable is used to transmit the signals from the accelerometer to and through a broadband velocity meter and on to a specially designed FM tape recorder where the analog data is stored on standard, high quality cassette tape. The velocity meter displays the overall broadband vibration velocity amplitude on a decibel scale referenced to 10^{-6} cm/sec rms. This reading is manually entered on a written record maintained by the PMT technician. The duration of each reading is "timed" by referring to the counter on the tape recorder. The recorders are fitted with multiple heads for recording the data plus a separate voice track on which the technician may record comments or other information regarding the measurement or test conditions. After all readings for a ship have been taken, the tapes are brought back to the local PMT office where a signal analyzer is used to convert the analog data to digital data which is then temporarily stored locally in a microprocessor. Using a modem and commercial telephone lines, the data are then transmitted to the analysis center. Other equipment used by the technicians includes a portable tachometer for measuring machine rpm at time of testing, a transducer calibration kit for periodic verification of transducer performance. and a dual trace oscilloscope which is used primarily to verify signal quality and tape recorder performance before each reading is taken.

C. DATA PROCESSING AND ANALYSIS

At the analysis center, signal analyzers and microprocessors are used for processing, display, analysis, and storage of the data. The analysis is performed with the data displayed in terms of velocity decibels versus orders of revolutions, and all final written reports are similarly prepared. The ability to display the data in orders is accomplished by referencing the spectral display to the rpm reported by the technician. Use of an orders representation of the spectra is advantageous for monitoring machines with variable operating speeds since it will automatically maintain the same spectral registry of those events that track directly to shaft speed, and this includes most of the common machinery faults. For example, imbalance creates a spectral line at 1 x rpm regardless of what that rpm may be. A machine operating at 1200 rpm will show a spectral line at 20 hertz, at 1800 rpm it will show at 30 hertz; but in either case it will show at one order in an orders representation. Exceptions to this are events which occurr at the same frequencies regardless of shaft speed. Events such as electrical signals from motors or resonant excitations of natural frequencies are common examples of this. When a component is excited into resonance, a spectral line will appear at the component's natural frequency and will not move as the machine speed changes. It may, and normally does, change in amplitude with changes in shaft speed, but its spectral location will remain fixed with respect to frequency. The assured consistancy of spectral location for most major events when displayed in terms of orders facilitates trending studies by eliminating one of the variables (operating rpm) which normally would be an extra factor to consider when comparing past and present data.

Reliability of trending information is very good since all acquired data is permanently stored. As the program continues, the data bank generated provides an increasingly valuable resource for statistical studies aimed at establishing and updating suitable criteria for alarm levels and condition evaluation of individual classes of machinery. At present, the criteria for alert are:

- a rise of over 6 dB in any amplitude as compared to its previous level,
- a rise above a level established statistically as two standard deviations above the mean value for that class machine, and
- a rise above the absolute levels set by military specifications.

A violation of any one of the above is considered an alarm or warning condition. An important distinction between the selection of these criteria and the criteria used by similar programs is the tracking of individual events (spectral lines). In some programs, the criteria are based only on overall broadband amplitude limits; this leaves the program "blind" to any smaller but more rapidly developing faults that may exist. For example, a 95 dB reading at 100 hertz and a 75 dB reading at 250 hertz would give an overall amplitude reading of 95 dB, the higher of the two. Assuming an alert level set at 100 dB, should the 100 hertz reading remain steady while the 250 hertz reading increases by 10 dB every three months, it would not register any warning until nine months later when its overall level would then be 105.

D. DOCUMENTATION AND REPORTS

The final written reports consist of tabular and graphical data. One graph shows trend history in the form of overall broadband readings plotted against time for the current and the several prior visits. A series of spectral graphs are made, one for each mounting location on each unit, and many of these are also shown on two separate order ranges. Typically, a high speed device may have one series of plots covering from zero to 100 orders, and a second series covering from zero to 10 orders in order to provide better resolution of lower order events. This is important because it is often problems in the lower frequency ranges that occur first and cause problems which appear later on in the upper frequency ranges.

The turnaround time from data collection to receipt of results and recommendations can be very fast, within one working day. This is a benefit because it allows the PMT technician to conduct followup testing on any unit which may warrant closer scrutiny based upon initial readings that were taken, and the ability to do so without need to re-visit the vessel at some future date which may be a less convenient time with regard to interruption of vessel operations.

Vibration data alone is not all that is reviewed before recommendations are formulated; it is augmented by information on the performance parameters of the units. In this way, the recommendations reflect an accounting for other variables which may have created either normal or abnormal changes in the signatures. The long term storage of the data permits the creation of waterfall plots used to simplify trending studies, and (ideally) improves the quality of information on machinery history statistics and the suitability of alert level values and evaluation criteria.

E. FUTURE DEVELOPMENTS

Future developments of the program are to include the use of digital data acquisition devices in place of the analog units now being used. This would allow direct transmission of the data when collected and cut out one major data processing procedure in the system. This would also allow for immediate channelling of data into local microprocessors onboard for on-site analysis which is another feature under consideration. Having these capabilities onboard ship will also permit the use of vibration readings as a machinery installation acceptance criteria. There is also a planned change over to electronic transmission of both the data and the reports. And finally, there is consideration being given to extending the monitoring interval from three to six months.

III. MECHANICAL VIBRATION SIGNATURE ANALYSIS

A. VIBRATION FUNDAMENTALS

A concept that is basic to the understanding of vibrations is one that Lyon [Ref. 1] emphasizes throughout his book, the concept of source-path-receiver. The oscillatory motion of a machine represents its response to exciting forces whose sources may be internal or external to the machinery unit. Examples of internal sources are rotor imbalance, coupling or bearing misalignment, or worn or damaged mating components such as gears; all representative of material or alignment defects which may be corrected to reduce the amplitude of the forces produced by them. Examples of external sources are load variations, flow conditions, or vibration of adjacent structures or equipment; all representative of problems which cannot be attributed to any physical defect of the machinery unit itself. The transmission of the forces from their sources follows one of two paths of prime interest to the vibration analyst, one in the form of the mechanical transmission of forces from one component to the next, the other in the form of the acoustic emissions emanating from each component which is set into vibratory motion. This describes the two basic types of signals that are measured and analyzed, mechanical and acoustic, which coincides with the two general fields that have developed. The instrumentation used is different for each of these two signal types, but the essentials of the analyses of the signals follow similar paths.

1. Where To Start

For design and analysis work, a real system is normally first identified using many terms such as linear or nonlinear, discrete or continuous, etc. which collectively serve to fully define the system so that an appropriate mathematical model may be developed for analytical study of the system and its response to various excitations. The result of a typical analysis is the description of system motion in terms of its displacement[4] expressed as a function of time. From this point, time derivatives of the displacement function may be taken to obtain expressions for velocity and acceleration and the motion may then be discussed in terms of any of these parameters.

Similarly, in the practice of measuring and analyzing the vibrations of real systems, the starting point is a time history of the motion received as either the

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[4] The term displacement is used in its most general sense; it may include translational and/or rotational components.

displacement, velocity, or acceleration amplitude versus time. A single time history record is referred to as a *sample*, and a collection of samples is referred to as an *ensemble*. Equation (1) represents the total, but finite, time history of a given process as being a set (ensemble) of N individual time history records (samples). Figure 2 displays the concept graphically.

$$x(t) = \{x_1(t), x_2(t), \dots x_N(t)\} \tag{1}$$

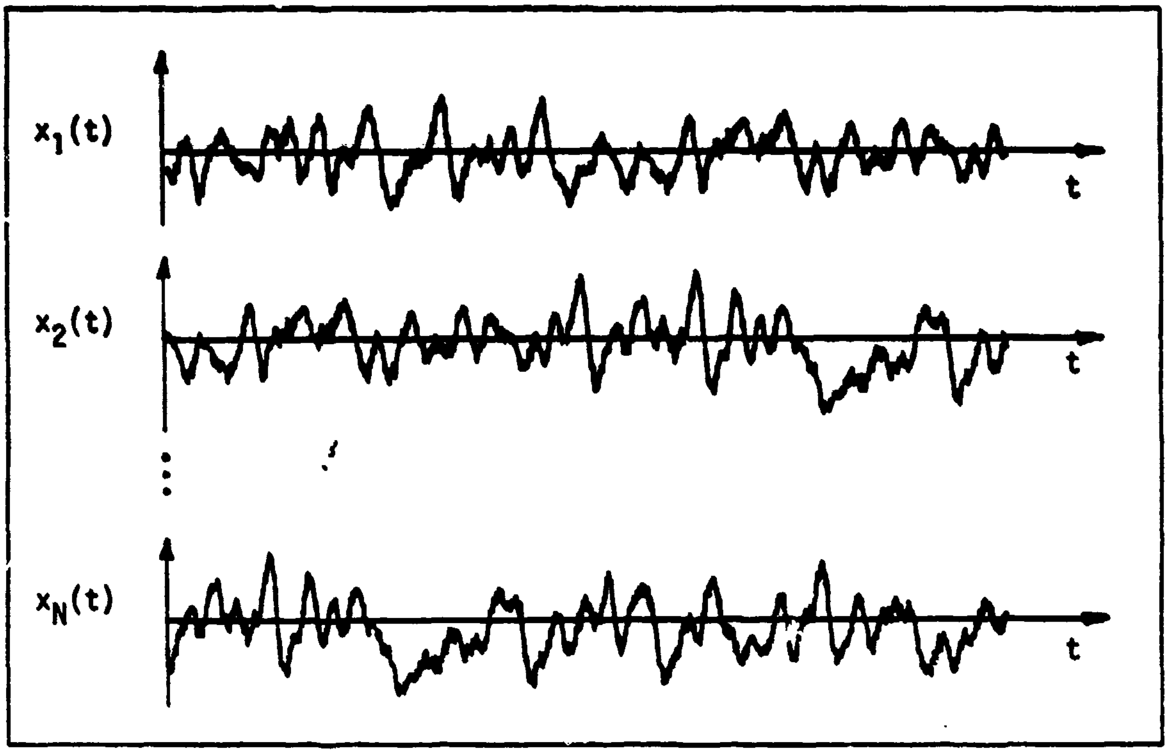


Figure 2. Time history of x(t) represented as an ensemble of N samples.

2. Random Vibrations

Processes in general can initially be classified as either deterministic or non-deterministic. Deterministic processes are those which have events that repeat themselves exactly and at fixed intervals. For deterministic processes, only one sample is required in order to predict (determine) future events. Processes which cannot be classified as deterministic are called non-deterministic or random or stochastic, and statistics and probability theory are relied upon in order to study and describe the processes and the systems which they represent. Although many processes may sometimes be adequately modelled as being deterministic, all real processes are technically classified as random processes.

Using statistical averages of their data, random processes may be further classified as being either stationary or non-stationary. A stationary process is one where each of all possible statistical properties as averaged over the ensemble converges to to a finite value as the number of samples in the ensemble increases. A non-stationary process is one where any one statistical property fails to converge. A weakly stationary process is one where not all statistical properties, but at least certain essential ones, have convergent limits. The basic criteria normally applied to establish weak stationarity are that the mean value, equation (2), the mean square value, equation (3), and the autocorrelation, equation (4), have convergent limits.

$$\mu_x(t_1) = \lim_{N \to \infty} \frac{1}{N} \sum_{i=1}^{N} x_i(t_1) \tag{2}$$

$$\psi_x^2(t_1) = \lim_{N \to \infty} \frac{1}{N} \sum_{i=1}^{N} x_i^2(t_1) \tag{3}$$

$$R_{xx}(t_1, \tau) = \lim_{N \to \infty} \frac{1}{N} \sum_{i=1}^{N} x_i(t_1)\, x_i(t_1 + \tau) \tag{4}$$

Authors differ somewhat with regard to which statistical values should specifically be included in the criteria used to establish stationarity. Bendat and Piersol [Ref. 7] perhaps stated it most appropriately in saying

> For the special case where all average values of interest remain constant with changes in the time t_1, the data are said to be stationary.

This would make it a case-specific determination. In practice, proving a process to be weakly stationary is usually sufficient to allow the process to be treated as if it were fully stationary. Bendat and Piersol [Ref. 7] expand on this where they state

> In most laboratory experiments, one can usually force the results of the experiment to be stationary by simply maintaining constant experimental conditions. For example, if one is interested in the surface pressures inside a pipe due to high velocity air flow, stationary data will be generated if the air flow velocity, density,

and temperature are held constant during each experiment....In many field experiments as well, there is no difficulty in performing the experiments under constant conditions to obtain stationary data.

The classification of a process as either stationary or non-stationary gives an idea of the "degree" of randomness of the process and serves as an indication of how relevant and meaningful statistical values may be in describing the process.

A further classification of stationary processes is in identifying them as either ergodic or non-ergodic. A stationary process is ergodic if the values of statistical properties as averaged over time for a single sample (equations (5), (6), and (7)) yield the same values as obtained to establish stationarity (equations (2), (3), and (4)).

$$\mu_x = \lim_{T \to \infty} \frac{1}{T} \int_0^T x(t)dt \tag{5}$$

$$\psi_x^2 = \lim_{T \to \infty} \frac{1}{T} \int_0^T x^2(t)dt \tag{6}$$

$$R_{xx}(\tau) = \lim_{T \to \infty} \frac{1}{T} \int_0^T x(t)\, x(t+\tau)dt \tag{7}$$

This implies that, similar to the case of the deterministic process, only a single sample is required in order to obtain information about the entire process history. Another time domain estimate[5] of importance is the cross correlation, equation (8).

$$R_{xy}(\tau) = \lim_{T \to \infty} \frac{1}{T} \int_0^T x(t)\, y(t+\tau)dt \tag{8}$$

By taking the Fourier Transform of the time history records, the data are transformed into the frequency domain. In this domain there are several spectral estimates of importance; namely, the linear spectrum (equation (9)), the power spectrum

[5] The term estimate is used because, although values are theoretically defined by integrals taken over all time, actual values are computed for finite record lengths, thus they can only be estimates of the true values.

(equation (10)), the cross spectrum (equation (11)), the frequency response function (equation (12)), and the coherence (equation (13)).

$$F_x(f) = \mathrm{FT}\,[x(t)] \tag{9}$$

$$G_{xx}(f) = F_x\, F_x^* \tag{10}$$

$$G_{yx}(f) = F_y\, F_x^* \tag{11}$$

$$H(f) = \frac{G_{xy}}{G_{xx}} \tag{12}$$

$$\gamma_{xy}^2(f) = \frac{G_{xy}\, G_{xy}^*}{G_{xx}\, G_{yy}} \tag{13}$$

where:

FT = Fourier Transform

F_x^* denotes the complex conjugate of F_x

G_{xy}^* denotes the complex conjugate of G_{xy}

3. Stochastic Modelling and the Fast Fourier Transform

By using trandsucers to convert physical motions into electrical signals which are then displayed on a CRT, a model of the actual process is being generated; in the case of machinery vibrations which are random, it is a stochastic model.

The analysis of time signals and their spectral representations is not new. They have long been studied by many, especially by electrical and electronics personnel, and activity increased tremendously after the coupling of the Fast Fourier Transform with solid-state and computer technology developments. As a result, many of the vibration analysis techniques used today regarding time and frequency domain displays of the same data rely on a few basic relationships which are well established and tested. Therefore, much of the vibration analyst's work is left to relating the signal information from the stochastic model to what is being shown about the real system it represents. The clear and precise spectral display of periodic signals is what gives spectral analysis its strength as a machinery diagnostics tool, especially when one recognizes that when machinery is operating, the forces and motions produced are very highly cyclic, especially in the case of rotating machinery. Just as any function may be represented as a Fourier series sum of sine and cosine functions of various discrete frequencies and amplitudes, a machine's overall vibration signal may be viewed similarly. When the signal is decomposed, the discrete frequencies and amplitudes obtained can be directly related to specific components and/or conditions occurring inside the unit.

B. MEASUREMENT AND PROCESSING FUNDAMENTALS

In the acquisition, processing, and display of vibration data, there are certain terms and aspects which should be understood in order to effectively measure and analyze the signals; and if designing a system, some are crucial to understand so that the appropriate hardware/software items are included in the design to minimize errors. Solid-state analyzers often have all required components already incorporated in their design. Even so, knowledge of these aspects will assist greatly in a user's choice of signal display format and its interpretation. Each of the following subheadings briefly introduces some of the more important terms and aspects.

1. Aliasing

Aliasing is a problem which occurs when there is no prefiltering of the raw data. For a given frequency span selected for analysis, the absense of any prefilters will result in frequency components above the upper limit of the selected span to be reflected or "folded back" into the span being analyzed. The reason for this is related to the rate at which the data is sampled. A common example of this is the apparent reversed rotation of a spinning object such as a car tire as it turns at different speeds. Blackburn [Ref. 8] uses the wagon wheel example, but one perhaps more common to engineers might be the apparent rotation reversals of a spinning shaft as viewed under a strobe light while the shaft speed varies. Elimination of this effect is attained by ensuring that the data sampling frequency is at least twice the maximum frequency desired to be analyzed. It is accomplished in practice by prefiltering the data with a low-pass filter (called an anti-aliasing filter) set at the maximum frequency to be analyzed. Off-the-shelf analyzers have an anti-aliasing filter incorporated in them; often they are reset automatically to the maximum frequency that the user selects for a particular measurement run.

2. Leakage

Leakage is a phenomenon occurring when unweighted non-periodic samples are processed. If a perfectly periodic signal is sampled such that the beginning and end of each sample coincides with the period of the signal and at its zero crossing, then the unweighted FFT of that sample will be accurate. However, a non-periodic signal will have an FFT display where the amplitudes are distortedly spread out across the domain of the display. To counteract this effect, the data is multiplied by a weighting function, commonly called a windowing function or simply a window, which has the effect of shaping the data in the sample so as to force the amplitudes at the beginning and end of the record toward zero. In this way, the data is forced into a periodic form and "accurately" transformed by the FFT into the frequency domain. The word accurately is used in quotes because this windowing process distorts the data from its true nature which means that all the displayed frequencies and amplitudes will not be exact, but these inaccuracies are overcome by reanalyzing portions of the data over smaller frequency spans to improve the resolution. There are several basic types of windows, each for a specific application, and there are many variations on several of these types. Off-the-shelf analyzers generally offer selection from a listing of several windows and some provide a user-defined window option so that the user may enter his/her own weighting function. Computer-based systems normally include these as menu-selectable items which come with the software.

It is often this problem of leakage which demands that several measurement runs be made covering various frequency spans in order to complete a diagnostic study. Use of smaller spans will improve both the amplitude and the frequency resolution. The selection of how small to go and when to stop is up to the analyst and usually based upon what is being searched for and how accurate the results need to be. In diagnostic work, the refinement may be quite detailed; in machinery monitoring applications it is less important what the degree of refinement and more important that the measurement parameters be consistent from one set of readings to the next.

To show the effects of leakage, a measurement was taken using a uniform window. The measurement display is shown in Figure 3. The wide spread of the peak at 30 Hz shows how the phenomenon of leakage presents itself. Ideally this peak should appear as a straight vertical line at 30 Hz indicating a vibration which is physically related to some machinery event occurring at 30 Hz (1800 rpm). In this case it was shaft imbalance where the shaft speed was 1800 rpm.

3. Windowing

Windowing is the application of weighting functions to the raw data samples. Of the many types of windows, four of the more commonly encountered ones are mentioned here; uniform, exponential, Hanning, and flat top.

The uniform window, also called the rectangular window, is merely a unit step function. Its affect on the sample is to weigh all components equally. It is mainly used for transients, impulses, or any other self-windowing signals where the amplitudes naturally attenuate to zero by the end of the time record.

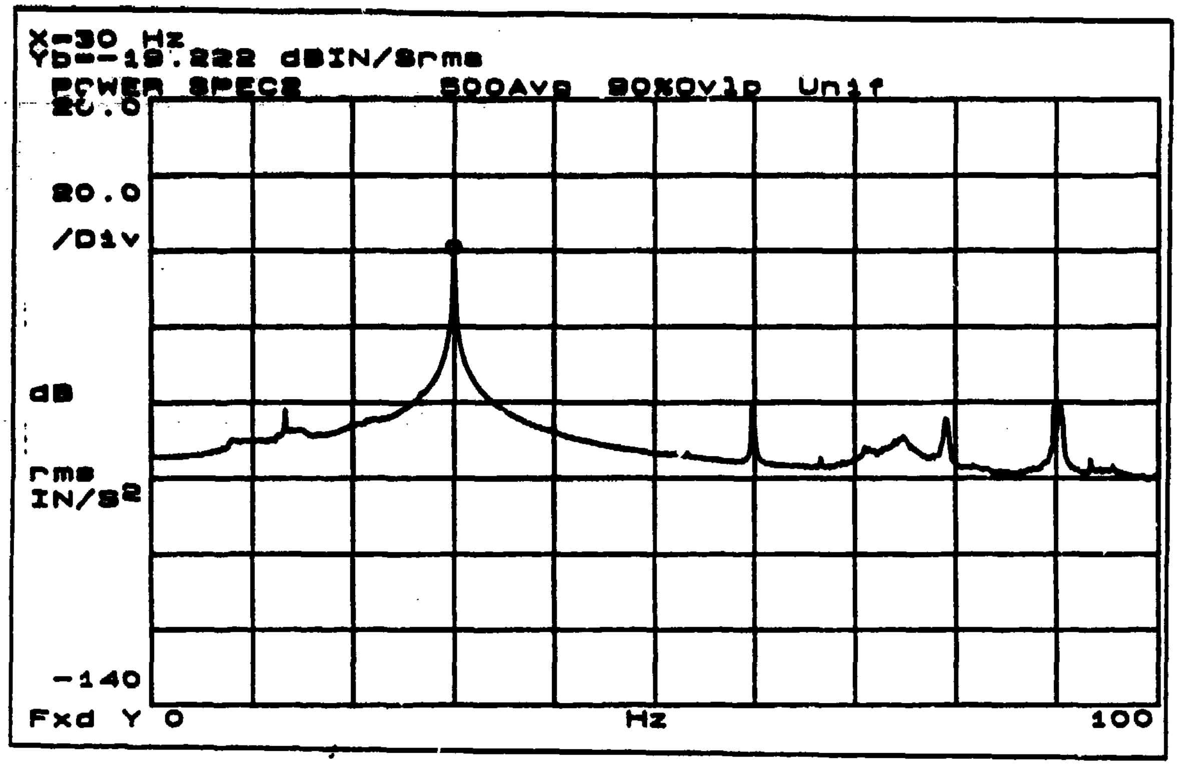


Figure 3. Measurement taken using the uniform window.

The exponential window may be used for helping force transients to zero by the end of the time record, if necessary, or sometimes merely to help reduce the effects of noise. For example, an eight millisecond time sample may contain a transient which dies out in three milliseconds. What is left is five milliseconds of nothing except whatever noise might be present. An exponential window may be applied which attenuates to zero at around three milliseconds to eliminate the five milliseconds of noise from the signal.[Ref. 9]

The Hanning window is perhaps the most commonly used for nonperiodic signals. The Hanning function is given by

$$u_h(t) = \frac{1}{2}(1 - \cos\frac{2\pi t}{T}) \quad 0 \leq t \leq T \tag{14}$$

This window heavily shapes the beginning and end of the sample to zero and its representation in the frequency domain appears as a very tall and narrow central lobe located at mid-span in the frequency domain display, with a series of successively smaller side lobes on either side. This shape results in rapid amplitude attenuation either side

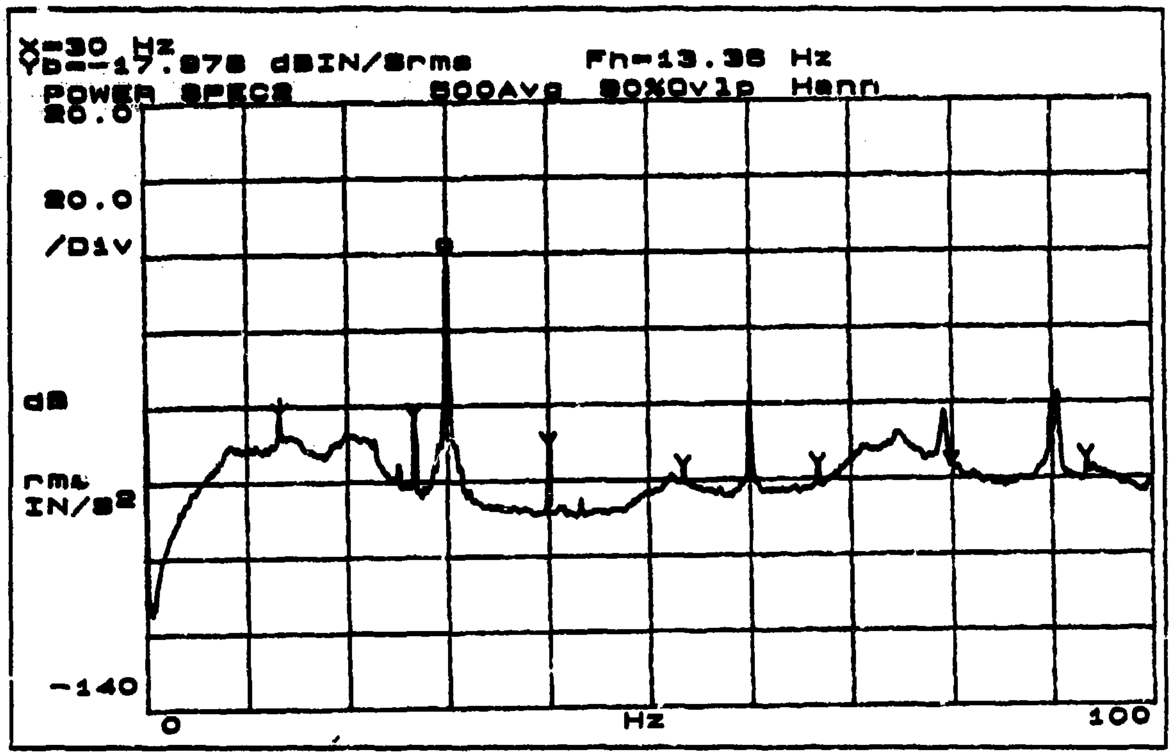


Figure 4. Measurement taken using the Hanning window.

of center span but gives very good frequency resolution, thus it is a common choice when accuracy of frequency readings is important. Of course, the overall problem of leakage still exists; therefore, the absolute resolution of spectral line location will also still depend on the span of the baseband.

Figure 4 shows the results of the same measurement that was made to obtain Figure 3, but this time using the Hanning window. The spectral peak at 30 Hz is seen to be much better resolved. It can also be seen that the distortion caused by using the uniform window resulted in almost total masking of the spectral lines at 26.72 and 40.08 Hz. In Figure 4, the special markers function was used to show the peak at 13.36 Hz and all of its harmonics which appear in the spectrum that indicate the presense of a significant event. In this case it was a drive belt defect created by wrapping a piece of tape around the glued butt joint of the drive belt which gave a once per revolution vibration pulse. The use of the Hanning window clearly shows each discrete harmonic peak.

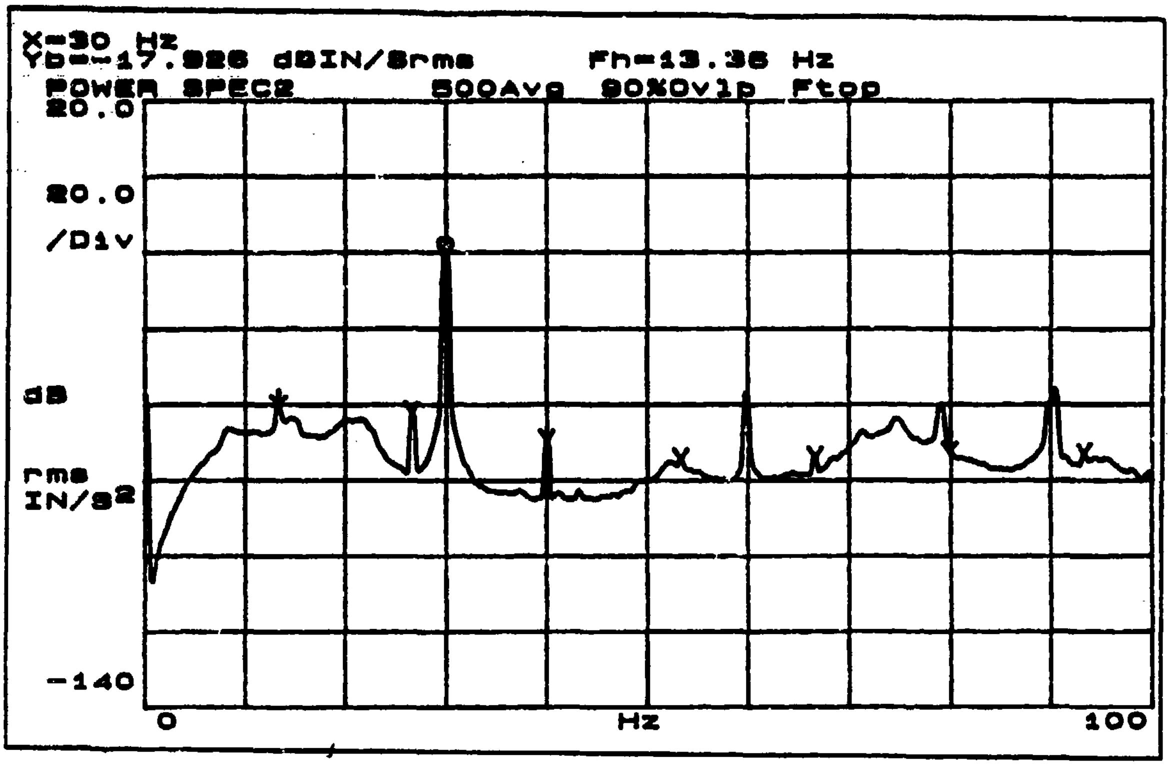


Figure 5. Measurement taken using the flat top window.

The flat top window is similar to the Hanning window, but has broader lobes and relatively flatter curvature at the top. This provides greater accuracy in amplitude measurements, but at the sacrifice of frequency resolution, and again the overall resolution depends heavily on the span of the baseband. Where accuracy of both frequency and amplitude are desired, the basic diagnostic approach would be to start with broad baseband measurements with the Hanning window to locate frequency ranges of interest, followed by measurements using smaller spans centered around particular frequencies of interest, and a final measurement made with the flattop window on a very narrow span. In practice, for general maintenance monitoring work, only one window type and one or two baseband spans would normally be used; otherwise, the analysis would be endless and the number of interpretations and comparisons required in doing trend studies would be enormous. The detailed analysis routine would only be done on a case by case basis where initial testing of a particular machine indicated that it should be diagnosed more thoroughly. The measurement made using the Hanning window (Figure 4) was repeated using the flat top window and is shown in Figure 5. It can be seen how the Hanning window results in thinner, sharper spectral peaks which gives good definition to the spectral locations (frequencies) of the periodic events occurring in the process being measured. In contrast, the flat top window gives better amplitude accuracy but, as seen in the widened and rounded-off appearance of the spectral peaks, the frequency resolution is poorer. Similar to the Hanning window, the flat top window was also very good at clearly showing the drive belt defect and its harmonics.

4. Averaging

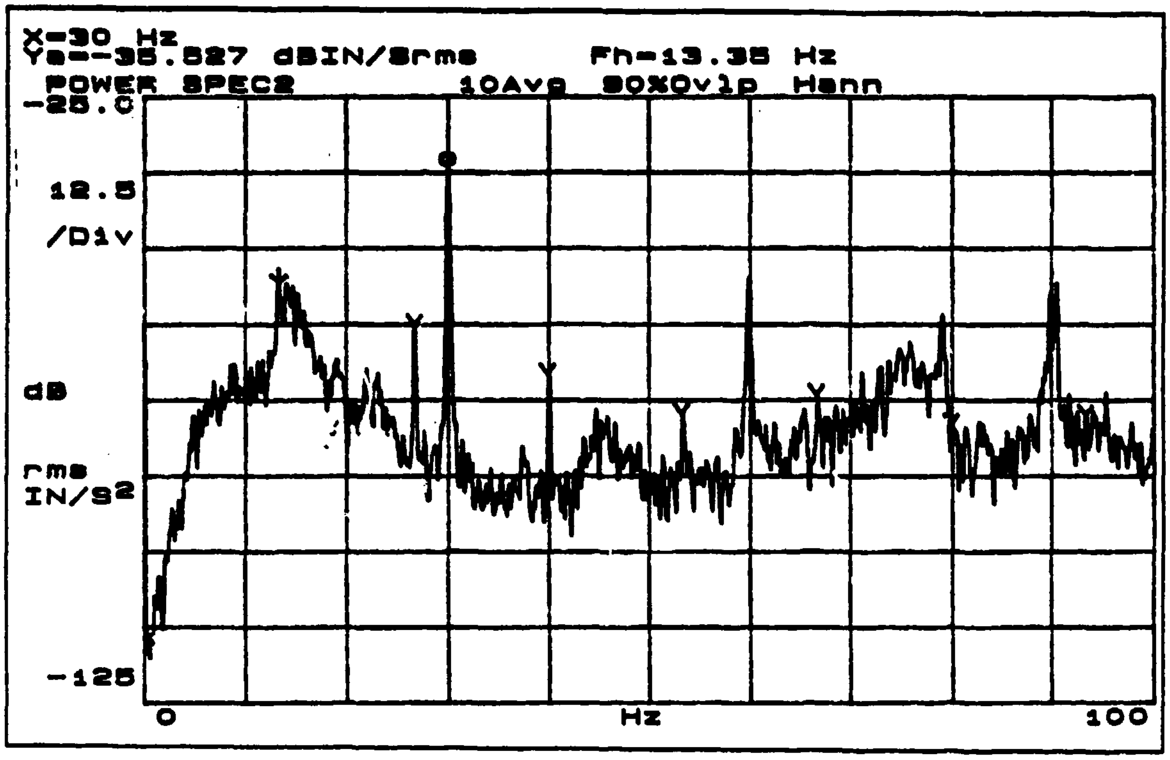


Figure 6. Average after 10 samples.

As the name implies, averaging means acquiring the averaged result of N samples; however, there are several methods of averaging that are used. RMS (root-mean-square) averaging is accomplished by averaging the transformed data samples. That is, the time records are transformed via FFT and then averaged. RMS averaging tends to smooth out the display of information by causing the amplitudes to tend toward their mean values. It does not actually improve SNR (signal-to-noise ratio), it merely causes both the signal and noise to average out to their respective mean values. The number of samples to be included in the average is selected by the user and is

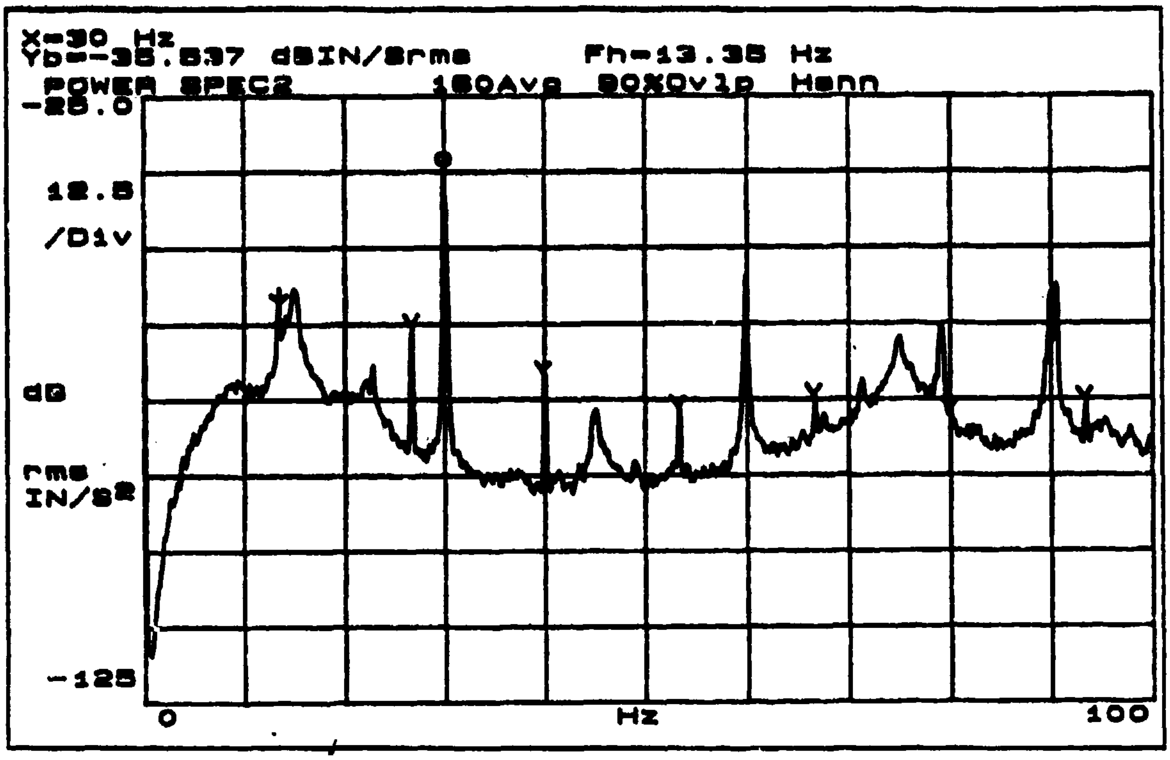
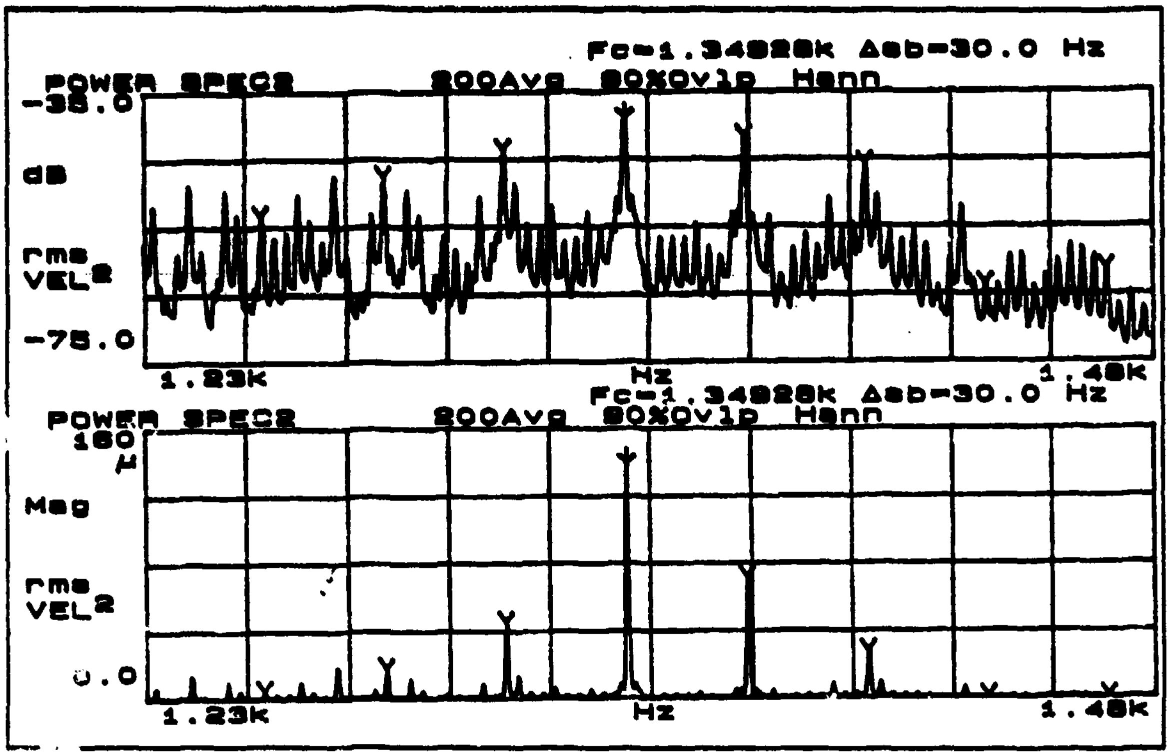


Figure 7. Average after 160 samples.

normally based upon two criteria: the degree of noise contamination of the signal, and the type and accuracy of information that the analyst wishes to obtain.

To show the affects of averaging, measurements of the same process were taken, each with a different number of samples included in the average; the results are shown in Figure 6 and Figure 7. Comparison of these figures shows how the signals of interest are made clearer as both the signals and the noise in the data approach their mean values. Although there is rather small change in the amplitudes except for several discrete frequencies near 10 Hz, this is not always the case. The use of the relatively small baseband span of only 100 Hz will, by itself, tend to give fairly good resolution within only a very few averages. What does become clearly resolved are the discrete frequency components of the data which initially are obscured by the noise. In addition to the improvement of the display through the use of averaging, the resolution was further enhanced in this case by the use of the Hanning window in taking these measurements. The slight width to some of the peaks may appear to be due to leakage. Their widths are more directly attributable to the fact that these are power spectrum measurements and due to there having been a slight bit of wander in rpm during these particular test runs.

5. Amplitude Scales



ıre 8. Displays using linear amplitude scale (upper) and decibel scale (lower).

Amplitudes may be displayed in terms of linear, log, or decibel scales. Decibel scales a by far the popular choice, mainly because of the ability to display both large and small amplitudes at the same time and with equal resolution.[6] The decibel is defined as

$$dB = 20 \log \left[\frac{\textit{measured value}}{\textit{reference value}} \right] \tag{15}$$

Time domain data are normally expressed in raw form, i.e., using a linear amplitude scale, whereas frequency domain data are most often expressed using the decibel scale. A comparison between linear amplitude and decibel scales applied to the same measured

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6 All decibel scales for the figures in this paper are referenced to unit value of the dimension of the scale.

data is shown in Figure 8. The figure shows a classic gear-related phenomenon called sidebanding. The main central peak at approximately 1350 Hz is associated with the frequency of tooth mesh, and the sidebanding frequency of 30 Hz is associated with the shaft speed of one of the gears. The usefulness of the decibel scale is seen by noting that there is also a sidebanding frequency of three Hz present, a fact that is barely visible in the linear amplitude scale display.

6. Miscellaneous Definitions

There are certain other terms which one should at least be familiar with in order to easily follow discussions of vibration analysis methods. For brevity, these are summarized here in definition format. Most of these were taken directly from the glossary of terms contained in a technical publication which discussed dynamic signal analyzers and their application in vibration work [Ref. 10].

Aliasing — A phenomenon which can occur whenever a signal is not sampled at greater than twice the maximum frequency component. Causes high frequency signals to appear at low frequencies. Aliasing is avoided by filtering out signals greater than 1/2 the sample rate.

Anti-aliasing filter — A low-pass filter designed to filter out frequencies higher than 1/2 the sample rate in order to prevent aliasing.

Averaging — In a DSA (Dynamic Signal Analyzer), digitally averaging several measurements to improve accuracy or to reduce the level of asynchronous components.

Band-pass fliter — A filter with a single transmission band extending from lower to upper cutoff frequencies. The width of the band is determined by the separation of frequencies at which amplitude is attenuated by 3 dB (0.707).

Bandwidth — The spacing between frequencies at which a band-pass filter attenuates the signal by 3 dB. In a DSA, measurement bandwidth is equal to [(frequency span)/(number of filters)] x (window factor). Window factors are: 1 for uniform, 1.5 for Hanning, and 3.63 for flattop.

Block size — The number of samples used in a DSA to compute the Fast Fourier Transform. Also the number of samples in a DSA time display. Most DSAs use a block size of 1024. Smaller block size reduces resolution.

High-pass filter — A filter with a transmission band starting at a lower cutoff frequency and extending to (theoretically) infinite frequency.

Keyphasor — A signal used in rotating machinery measurements, generated by a transducer observing a once-per-revolution event. The keyphasor is used in phase measurements for analysis and balancing. (Keyphasor is a Bentley-Nevada trade name.)

Leakage	In DSAs, a result of finite time record length that results in smearing of frequency components. Its effects are greatly reduced by the use of weighted window functions such as flattop and Hanning.
Low-pass filter	A filter whose transmission band extends from dc to an upper cutoff frequency.
Octave	The interval between two frequencies with a ratio of 2 to 1.
Spectral map	A three dimensional plot of the vibration amplitude spectrum versus another variable, usually time or rpm. Also known as a cascade plot or waterfall plot.
Tracking filter	A low-pass or band-pass filter which automatically tracks the input signal. A tracking filter is usually required for aliasing protection when data sampling is controlled externally.

C. SIGNAL MEASUREMENT AND PROCESSING EQUIPMENT

1. Transducers

The proper selection of transducers is one of the more important choices to make in the initial design of any vibration measurement system. A myriad of types and specific models suited for special applications are available from which to choose.

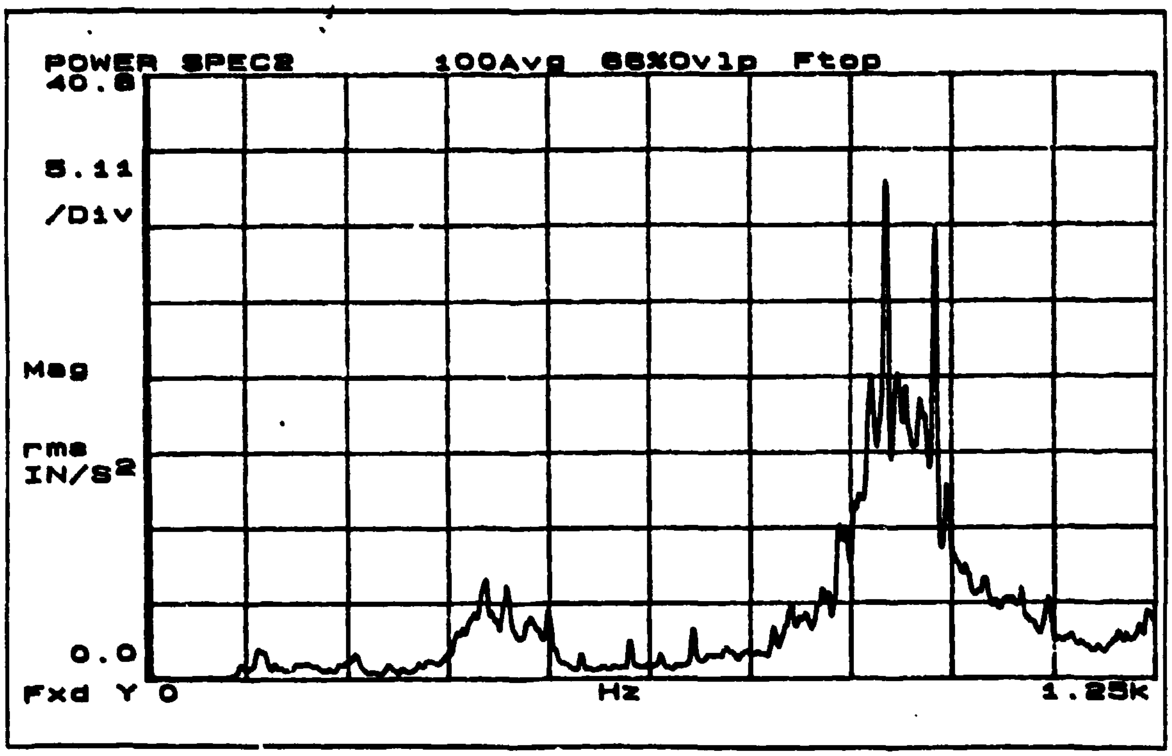

Figure 9. Acceleration measurement.

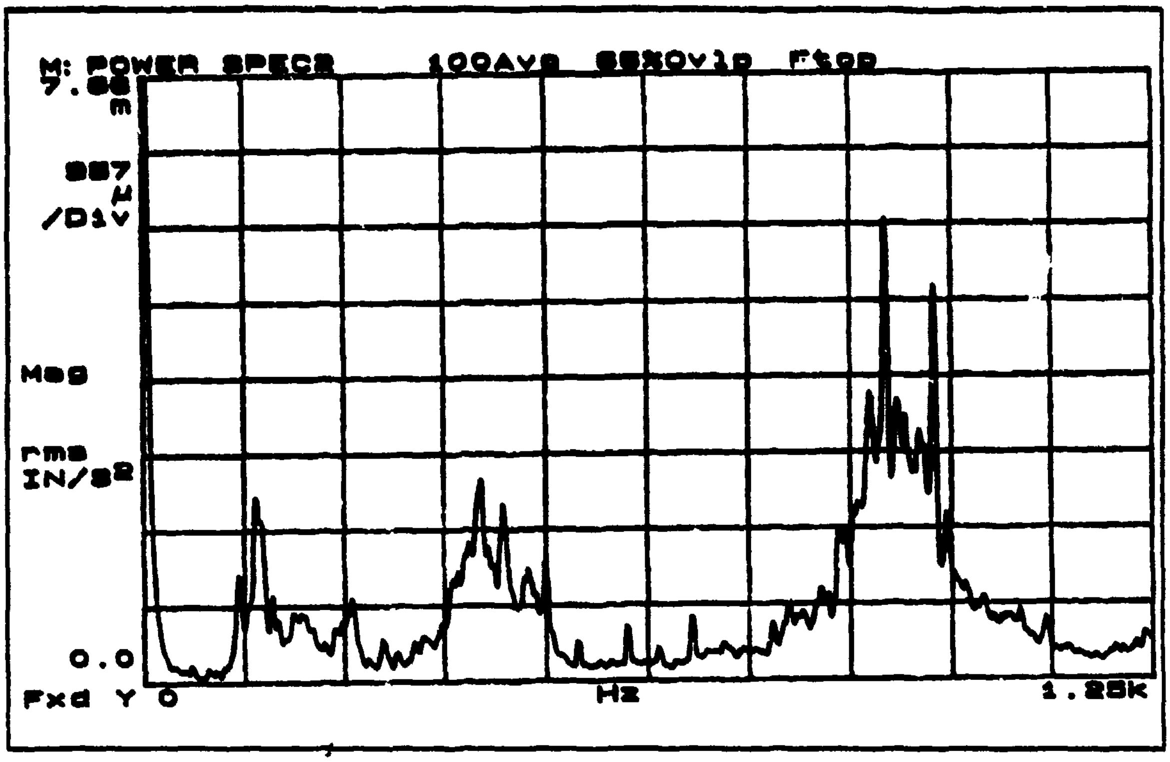

Figure 10. **Velocity measurement.**

The first consideration is to select between a displacement, velocity, or acceleration transducer as a basic type. This decision is largely based upon the type of equipment to be analyzed, the frequency range to be analyzed, and the specific purpose of the analysis. Figure 9 through Figure 11show three different displays of the same measured data. The first one shows a vibration acceleration measurement that was taken which was then integrated twice to yield displays of vibration velocity (Figure 10) and vibration displacement (Figure 11). The distortion present in the velocity and displacement displays near zero Hz is due to the integration process done by the signal analyzer; actual measurements taken directly with velocity and displacement transducers would not exhibit this distortion. This shows how the choice of units will vary the appearance greatly. These differences can be used to advantage in the following way; to enhance low frequency events (such as imbalance and misalignment), use displacement measurements; to enhance high frequency events (such as the onset of rolling element bearing defects), use acceleration measurements.

For general overall severity of vibration, velocity is normally chosen since it allows for comparisons that are essentially frequency-independent and, therefore, also

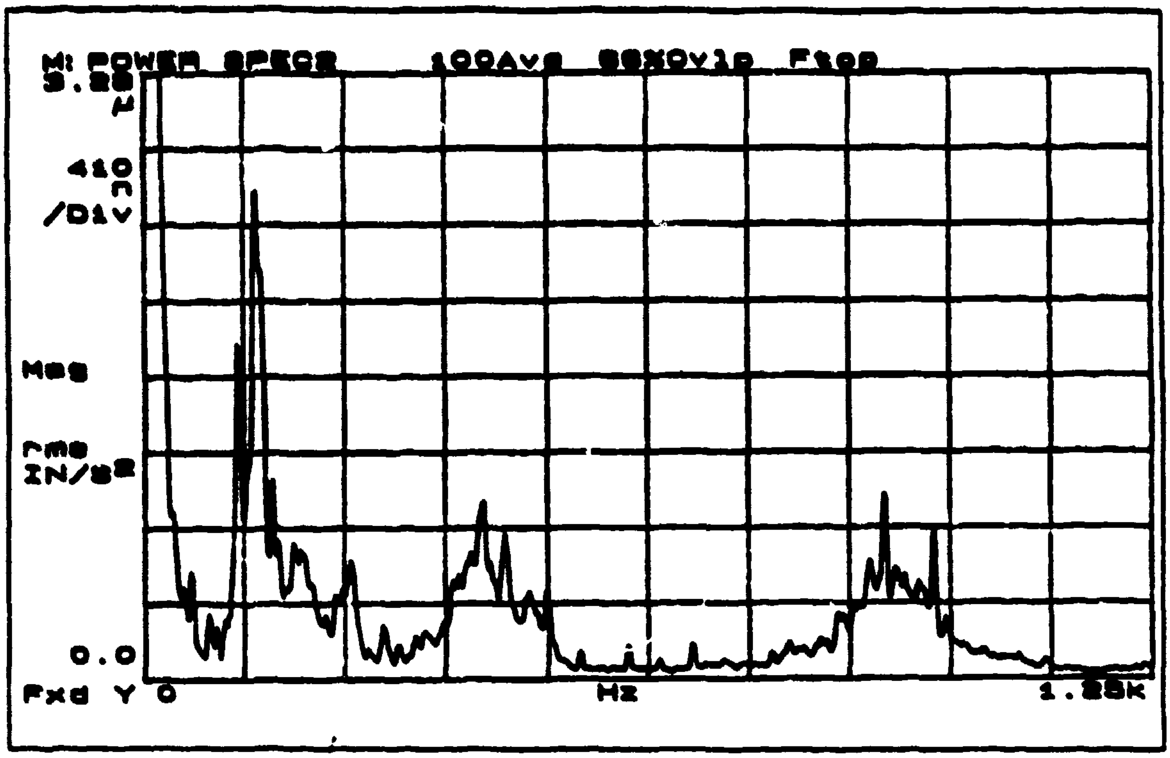

Figure 11. Displacement measurement.

requires the least dynamic range of the three. In general, displacement types are recommended for frequencies up to approximately 600 Hz, velocity types for the range from approximately 10 to 1000 Hz, and acceleration types for all higher frequencies. The limits are mainly imposed by the natural frequencies and the inertial properties of the transducers. Another consideration is the mechanical impedance of the measurement point(s). If vibrations are transmitted poorly to the casing, as is the case where hydrodynamic bearings are involved, then direct measurement of the shaft motion with a displacement transducer (a.k.a. proximity probe) is needed. For general measurements via hand held probes using portable meters, velocity transducers are the more suitable choice. By far, however, accelerometers are used in almost all other situations. Current technology provides accelerometers which are suited to a wide range of applications. An added advantage is that, with simple integrating circuitry, the acceleration signal may be converted to velocity or displacement with little measurable introduction of noise. However, if one of the other types is used and differentiation is done to convert the signal to another form, the signal conditioning process involved generally adds significant noise to the signal.

2. Signal Conditioners

Velocity transducers require no signal conditioning since the principle of operation (spring-loaded magnet moving through a housing-mounted coil which induces an electro-motive force in the windings of the coil) generates its own output voltage signal. But displacement and many accelerometers require signal conditioning equipment. Accelerometers are available which contain integrated circuitry which allows them to be directly connected to an analyzer provided that the analyzer is designed to accomodate this. In some cases, signals may require amplification and this normally is attained by line power amplifiers in place of the line power units. Also, all signals will be required to pass through anti-aliasing filters which, technically, should be included as a type of signal conditioner, however, many analyzers will usually have these designed into their hardware which makes them go unnoticed and often forgotten.

3. Portable Equipment

Portable meters of more common use are displacement meters, velocity meters, and sound level meters which measure wideband overall levels of vibration or sound. In general they are used to satisfy simple periodic monitoring needs regarding the equipment of secondary importance in predictive maintenance programs, and are used as the primary detection and measurement devices in lower level monitoring programs. Again, many varieties abound, some have selectable units of their displays to allow either displacement or velocity measurements to be made with the same meter, others have different filtering capabilities such that either overall unfiltered or filtered wideband readings may be taken, and some include switchable settings to allow display of either peak or rms values. These meters are of general use to plant operators who wish to take overall periodic readings on specific units and they can be used quite effectively for maintenance monitoring applications, but only on a small scale, and they are not adequate for detailed diagnostic work.

Aside from the meters, there are many compact portable solid-state analyzers with FFT capabilities, screen display of spectra, and printed output of tables or graphs. Computer technology has made these units powerful and affordable, and they may suffice for maintenance monitoring needs, but they are not fully adequate for comprehensive diagnostics, especially when compared to the larger, more sophisticated dynamic signal analyzers.

4. Non-portable Equipment

Semiportable equipment includes devices which are essentially larger, more fully functioned vibration meters which have adjustable or tunable filters plus other miscellaneous options. Several known designs operate by manually tuning the narrow-bandpass filter up through a frequency range of interest and observing deflections on the frequency and amplitude meters that are mounted on the device. Many of these can be connected directly to X-Y plotters to generate spectral displays and some are fitted with strobe light connections which enable a strobe light to be triggered by the incoming vibration signal. This can be a big time saver in many instances by helping to isolate which component in a group is the one that contains the faulty condition, especially when the component is out in the open such as when the defects are associated with the drive belts of an assembly. As various attributes and special functions are added, these units evolve into more highly sophisticated units known as real time analyzers or dynamic signal analyzers. These have digital filters, analog-to-digital converters, built-in microprocessors with preprogrammed FFT and many other special functions which are usually menu or soft key selectable from the main control panel. Input signals from transducers are received as analog signals which get prefiltered, digitized, then processed, and many also have conventional input/output ports for access to and from digital sources and storage media. These are extremely well equipped to handle most diagnostic needs. The next step up from the dynamic signal analyzers is a totally computer-based system. Various companies carry lines of hardware and software which provide a user with all that is required to establish a complete diagnostic system. Computer-based systems are typically only used in very large plants where, mainly based on the economics of the situation, it is more advantageous to opt for a continuous monitoring system and/or there are a very large number of monitoring points to be covered.

D. TIME DOMAIN

As previously mentioned, there are occasions when an analysis may be done directly in the time domain. At least in some cases the time waveform can give immediate indications which, even if not useful to directly identify a specific fault, might be useful in classifying an event which helps determine appropriate subsequent techniques, and often it can be used to support observations made in other domains. The impulsive events associated with bearings and gears, the appearance of modulations which are common in faulty bearings and gears, and the truncation of amplitudes occurring when mechanical looseness is present are all instances when the time domain waveform shape or pattern may help lend strong direction to continued analysis efforts. Transient events are also very clearly displayed, making the time domain a natural one to use for their study. Several specific time domain measurements and techniques are of particular importance; a brief discussion of each follows.

1. Analysis Measurements

a. Autocorrelation

Recognizing that the prefix "auto" means "self", the term autocorrelation becomes self-explanatory; it is essentially a measure of a signal's correlation with itself. Referring back to equation (6), it is not a single value, but rather a distributed function of the independent time variable τ, which is why it is more properly referred to by its full name, the autocorrelation function. The function is obtained by multiplying the signal by an increasingly time shifted version of itself. In doing so, any periodic content in the signal will be reinforced and any nonperiodic content will eventually die out.

The autocorrelation function serves as an indicator of the periodic content of a signal. As such it may be useful in detecting the presense of periodic events which are hidden in signal noise. An example of this is shown in Figure 12 where the autocorrelation and the filtered linear spectrum for a particular event are shown. For this measurement, a 15-tooth gear and a 50-tooth gear were running in mesh, lightly loaded, and the 50-tooth gear had one tooth intentionally removed. The shaft speed for the defective gear was 300 rpm. The autocorrelation measurement shown has a basic periodic wave whose period is 3.125 milliseconds as measured from peak to peak using the cursor controls. This corresponds to a frequency of 320 Hz, just a little higher than the meshing frequency which is discretely shown in the accompanying linear spectrum. The difference is due to the digitization of the data and the resolution attainable for the frequency span that was used for the measurement. The distinctive periodic spikes in the autocorrelation arise due to the missing tooth which creates a periodic impulsive strike. When a synchronized trigger signal is available, time averaging is the better technique to use to separate the signal from the noise.

b. Cross Correlation

The cross correlation is a two channel measurement that shows the degree of time domain similarity between the two signals. In the autocorrelation measurement, signal components which were similar (periodic) were reinforced when the signal was compared to itself. Lik-wise, the cross correlation is obtained by multiplying the signal on one channel by an increasingly time shifted version of the signal on the other channel. Both measurements enhance those components which are similar, but only the autocorrelation measurement may declare them to be periodic because the similarities are in the same signal.

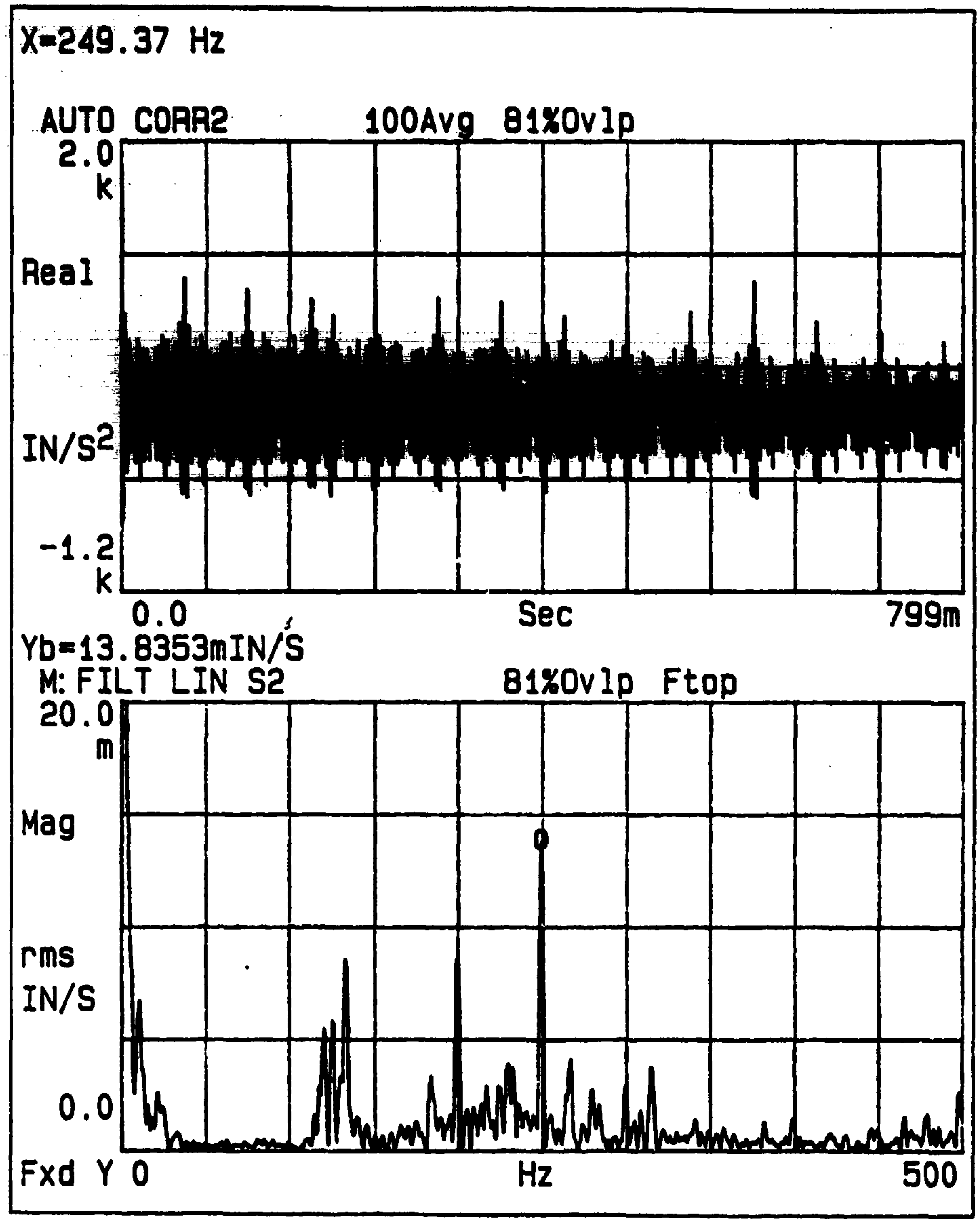

Figure 12. Autocorrelation and filtered linear spectrum measurements of gear train events.

c. Crest Factor

The crest factor is determined by ratioing the averaged peak amplitude to the averaged RMS amplitude. This factor provides a quantitative measure of a signal pattern. In gear analysis, it gives indications of certain specific conditions. For example, a broken tooth will raise the peak level but have minor influence on the RMS level; conversely, heavy overall wear will reduce RMS values but show little change in peak values. [Ref. 11]

d. Kurtosis

Kurtosis is defined as the fourth central moment of the data normalized by the square of the variance. It functions similarly to the crest factor in that it is a numerical value which reflects changes in peak levels or the number of peak impulses occurring. Kurtosis measurements have been successfully applied to both gear and bearing analyses. Normal values are taken to be around 3.0; as defects begin to appear, the kurtosis value increases. For rolling element bearings, the values will grow to around 10, then as the defects spread and overall degradation occurs, the kurtosis value begins to fall again back toward the initial level. This is noted in a Bruel & Kjaer publication [Ref. 12] and supported by research done by Reif and Lai [Ref. 13], Swansson and Favaloro [Ref. 14], Stronach, Cudworth, and Johnston [Ref. 15], and many others. As explained by Braun [Ref. 9], crest factor and kurtosis measurements are able to be meaningful measures of peaks in the signal because of their relation to the probability density function of the data. The probability density function will be affected at its extremeties by peaks occurring in the process, and moments of the data will reflect changes in these portions of the probability density function curve.

2. Analysis Techniques

a. Time Domain Averaging

Time domain averaging (a.k.a. time averaging, synchronous averaging, or linear averaging) is a special technique used to greatly improve the SNR and to focus measurement on one particular component or event occurring in a machine. In this technique, a keyphasor or other timing signal is used to trigger the start of each sample record data capture. In this way, any events that are synchronous with the trigger will appear at the same offset in each sample record; all non-synchronous events will occur at random times throughout the samples. As the number of records sampled increases, there is greater reinforcement of synchronous signal amplitudes and greater attenuation or cancellation of asynchronous signal amplitudes. Since any noise in the data will be totally random, it too will eventually average out to zero, leaving a very clean time trace of only the synchronous events. This technique is of particular use in analyzing gearboxes. Even when intending to analyze a gear or shaft event on a gearbox component which is "buried" (i.e., its shaft does not penetrate the casing), the technique may still be used by triggering off either the input or output shaft and incorporating a multipler in the signal line which can multiply the signal pulses by the appropriate factor to time the signal to the component of interest; the correct factor is obtained from a knowledge of the gearbox component geometry. As tested in gear analyses by Favaloro [Ref. 11] and McFadden [Ref. 16], it gives results which provide more than mere detection; the data can be displayed in a form which gives a direct graphic display of the geometry of the event. A difficulty to be overcome when using the technique is the accurate timing and tracking of the event as machine speed may vary slightly. Smith [Ref. 17] states that an inaccuracy in speed tracking of only one-tenth of one per cent will give severely degraded results if more than eight averages are taken; often several hundred may be required. Favaloro [Ref. 11] presents the mathematic formulae for obtaining the signal average, equation (15), and the correlation coefficient, equation (16), and states that the signal average is assumed to be stable when the correlation coefficient is greater than 0.99.

$$\bar{x}(t) = \frac{1}{N} \sum_{n=0}^{N-1} x(t + nT) \tag{16}$$

$$r = \frac{\sum_{l=1}^{n} (\bar{x}_{l,N})(\bar{x}_{l,N/2})}{\left(\sum_{l=1}^{n} (\bar{x}_{l,N})^2 \sum_{l=1}^{n} (\bar{x}_{l,N/2})^2\right)^{0.5}} \tag{17}$$

where:

n = total number of data points in a record

N = number of records averaged

T = coherence time, or time for one gear revolution

E. FREQUENCY DOMAIN

Much of the work done in mechanical vibration analysis is done in the frequency domain. Especially with regard to rotating machinery which can be viewed as a collection of periodic events occurring simultaneously, the FFT of the time signals provide very discrete information about each specific event involved. The time domain methods mentioned earlier are mostly geared toward investigations of specific events that are known to be present or specifically searched for. The methodologies in the frequency domain tend to be more investigative in nature in that, more often than not, they are used to see what events are occurring at unacceptable levels as opposed to selectively searching for fine details about events that are known to exist. So for routine diagnostic work the analysis generally begins more globally to scan for trouble areas shown in the spectral displays, then follow-up scans at higher resolutions (smaller frequency spans) are conducted to isolate and better define specific faults. The measures obtained are quantitative, but mostly on a relative scale rather than on an absolute scale. There are certain absolute standards in existence, e.g. ISO 2372 published by the International Standards Organization, and many manufacturers publish their own recommendations and guidelines for acceptable levels of vibration that their products should show in service. Most of the literature shows, though, that the end users of vibration analysis equipment generally have their own in-house standards and limits which often are far more stringent than those mandated or recommended by outside sources. Many of the limits used in maintenance monitoring programs are based on relative changes in amplitudes detected between two consecutive readings, and it may often occur that this type of criteria is violated before any fixed absolute limits are approached. Some of the more commonly used measures and techniques done in the frequency domain are briefly discussed below under separate headings.

1. Analysis Measurements

a. Line Spectrum

A line spectrum is simply the Fourier Transform of the time signal of the motion being measured. This is by far the most commonly used measure in spectral analysis as applied to machinery diagnostics. It incorporates the three basic pieces of information normally looked for: which frequencies are showing high amplitudes, what are the amplitudes, and how do they compare to each other and to previous records. The vibrations are highly direction oriented and this is of great importance in conluding what faults exist. Therefore, it is common to take readings in three coordinate directions; two radial readings mutually perpendicular, and one axial reading. With this information at hand, the analyst is well-equipped to commence comparisons of the data to identify the sources of trouble. Line spectrum measurements are obtainable through the use of a single channel, that is, a single transducer signal fed into a signal analyzer or FFT processing device, and this is what is normally done for routine monitoring work. In the case of detailed investigative diagnostics carried out on a particular machine to locate one or more problems which could not be isolated by the information provided by monitoring data alone, there would normally be a second transducer or at least a key phasor used in order to provide phase information. This is the main drawback to all forms of single channel measurements which do not use a keyphasor, they can provide no phase information.

b. Autospectrum

The autospectrum (also known as the power spectrum) gives a measure of the power contained in the signal at various frequencies. Although not normally used in machinery diagnostics to any great length, it can provide a means to quickly see where the problem areas are and where the analysis should focus its attention. As such, it can find usefulness in diagnostic work, but is not useful in maintenance monitoring activities. Readings of the power levels in the signals are of greater interest and use to acoustic engineers working on noise control methods and analyses. It also can, and usually is, obtained as a single channel measurement and, as with the line spectrum, will provide no phase information when so obtained.

c. Cross Spectrum

The cross spectrum is a two channel measurement which gives a mixture of information regarding the power spectrum of each of the two signals. Referring back to equation (11), it represents the amplitude product of the two spectra and the phase difference between them. It is mainly used in the computation of the transfer function which is obtained by dividing the cross spectrum by the power spectrum of the input signal. Beyond this, the literature gives very little mention of this measurement; it appears to be of little use in diagnostic work to date.

d. Frequency Response Function

The frequency response function is a two channel measurement which gives the ratio of a system's output to its input. It contains both magnitude (gain) and phase information displayed as functions of frequency. The measurement is normally used in modal analysis work for determining natural frequencies, damping factors, and other structural response characteristics.

e. *Coherence*

Coherence is a two channel measurement which indicates the amount of power in the output power spectrum that is related to the power in the input power spectrum. It is a function with amplitude ranging from zero to one and plotted as a function of frequency. It provides an index of the quality of the transfer function measurement and, consequently, should always be reviewed before any gain or phase information from the transfer function is accepted as valid. Coherence values less than unity are normally due to poor resolution (frequency analysis span too large), nonlinear system behavior, uncorrelated noise, or uncorrelated input signals [Ref. 7].

IV. MACHINERY DIAGNOSTICS MODEL

A. MODEL DESIGN AND DEVELOPMENT

Since the most useful aspect of the model was to be its ability to show machinery faults and their characteristics, no specific detailed design was required; however, the selection of components and their fabrication and assembly were carried out with care and precision to ensure that the model could also give vibration patterns representative of machinery that was well-balanced, well-aligned, and in otherwise normal, defect-free condition. In this way, meaningful comparisons between normal and abnormal conditions could be made. The faults intended to be included were damaged anti-friction bearings, damaged gears, rotating imbalance, mechanical looseness, defective drive belts, and misalignment. This assortment was felt to provide a reasonable number of common faults which could be incorporated into a single assembly whose size, weight, portability, and power and load requirements were within practical limits with respect to its intended immediate and future uses.

1. General Arrangement

Figure 13 shows a plan view of the model. It consists of four parallel shafts which may be interchangeably driven by either gears or flexible belts. The first three shafts as viewed from right to left were used in conducting misalignment, gear defect, and bearing defect experiments; the fourth shaft was used in conducting imbalance, drive belt defect, and mechanical looseness experiments. The specific construction of the balance discs on the fourth shaft also allow them to be used to demonstrate a typical rotor balancing procedure which may be of use in future vibration laboratory exercises.

The drive selected for the assembly was a one fifteenth horsepower, variable speed, permanent magnet dc motor. The motor controller provided constant speed control and had two separate speed setting knobs, one graduated from zero to 100 per cent (100 per cent gave a rated speed of 2650 rpm), and one for extended speed range, graduated from 100 to 200 per cent. It was also equipped with a three-amp circuit breaker and a switch to change motor rotational direction. A dc ammeter was installed to measure motor current draw which was used as an indicator of relative load. The actual load was not a required measurement for the experiments that were conducted, but it was necessary that many of the runs be done at the same load for sake of making valid comparisons of other parameters, and the meter served this purpose.

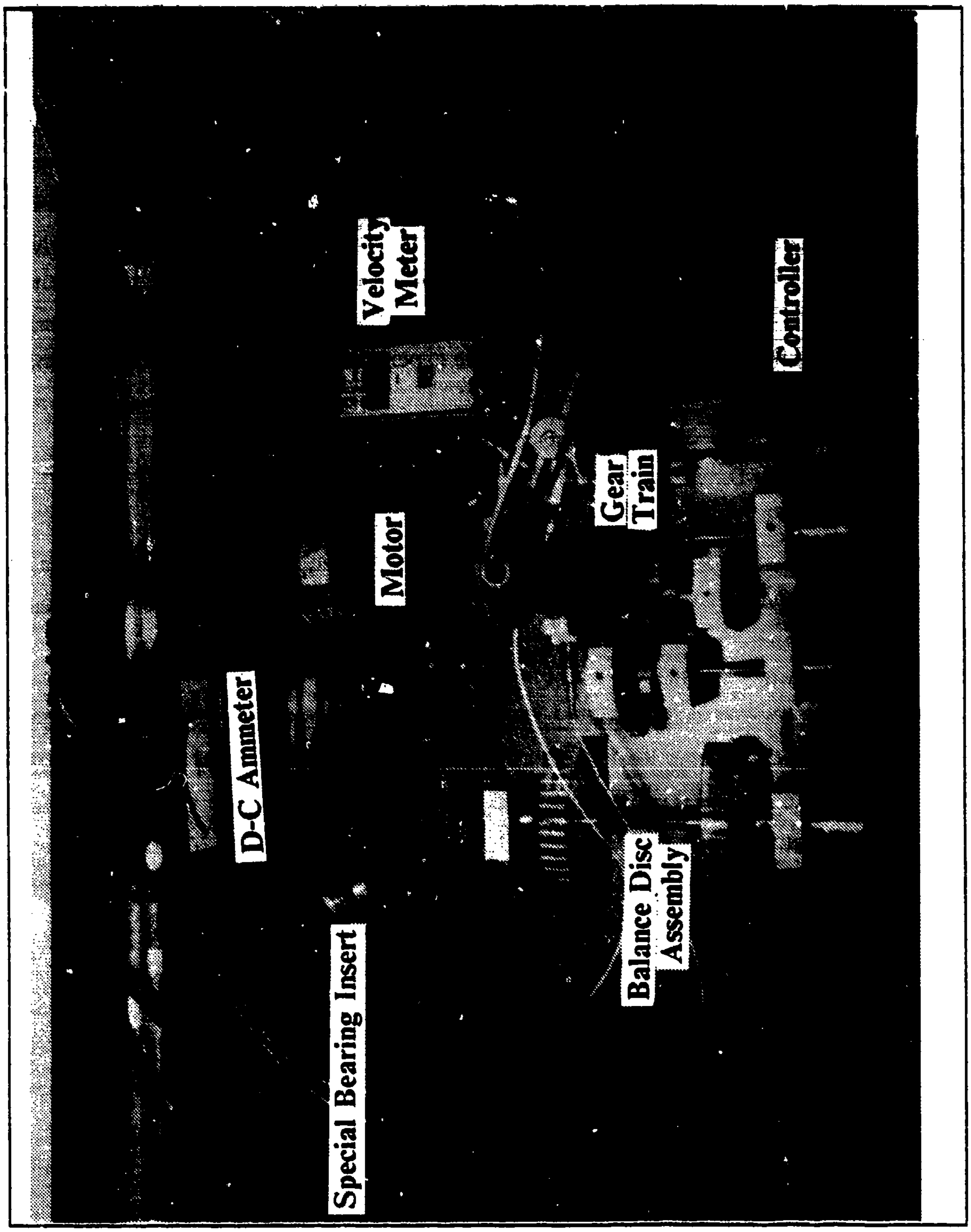

Figure 13. Plan view of the model.

Initially, an arrangement using a small 60:1 speed reducer run in reverse was used as a load to be driven by the third shaft of the gear train. Its availability and lack of need for ancillary equipment were prime reasons for its selection. This unit was sufficient to provide a countertorque which required approximately half the motor rated current and was a very steady load. Unfortunately, the arrangement suffered several material failures and associated complications in its connection to the gear shaft and it was eventually abandoned in favor of a direct friction drag on a disc mounted on the forward end of the third shaft of the train (the disc is not shown in the figure; a small single groove pulley can be seen in the place where the disc was mounted). Although the friction drag was acceptable for most of the experiments that were conducted, a load source with greater stability would be required to avoid any speed changes due to load variations if more detailed experiments are planned for this model.

2. Component Details

The shafts were cut from a single six foot length of 0.375 inch (+0.0000 to -0.0002) C-60 case hardened and ground steel rod. The selection of material and tolerance was intended to guarantee minimal deflection under load and a secure fit with mating components.

The anti-friction bearings used were Fafnir model AS3K radial ball bearings with a static load rating of 312 pounds and a dynamic load rating of 830 pounds. The inside and outside diameters were listed as 0.3750 inch (+0.0000 to -0.0003) and 0.8750 inch (+0.0000 to -0.0004), respectively. Ball diameter was measured to be 0.155 inch; and pitch diameter was calculated from the above dimensions as 0.675 inch. It was intended to obtain the lightest duty bearings as possible in order to keep required loads to a minimum; however, bearing sizes smaller than this would have greatly limited the size and selection of available gears to fit the same shaft. Obvious alternatives were to use a stepped shaft or to use as small a shaft as desired and install bushings in the gears to make up the difference in diameters. This led to two concerns; one was that bearings any smaller would be extremely difficult to disassemble, damage, and reassemble as was contemplated for the bearing study; the other was that adding more interfaces and material types (i.e., the bushings) would be adding more places where signal attenuation might occur due to transmissibility losses. For a detailed study on either gears or bearings alone, the choices would be simple, but the combined effort made compromise a necessity. No thrust bearings were used anywhere in the design since the only axial loads present would be due to angular misalignment of the spur gears, and the small amount of axial load able to be absorbed by the radial bearings was assumed to be sufficient to handle any loads so generated.

All the gears selected were common steel stock spur gears with three eighths inch face width, 14.5 degree pressure angle, and had no specified hardness or surface finish attributes. For detailed experiments of gears, these attributes would be of importance, but at the time of selection of these components no such studies were contemplated. Two 50 tooth, two 70 tooth, and four 15 tooth gears were obtained. The selection was guided by what was readily available in the 0.375 inch bore size and was otherwise based on what the meshing frequencies would be for various gear arrangements. This latter point was important in order to avoid overlapping of their events in spectral displays when the full train of gears was actively in mesh.

The pulleys were fabricated from two-inch thick aluminum plate machined to provide radiused grooves to accommodate flexible belts of circular cross-section. Envisioned for light loads and primarily to be used for step up or step down of shaft speeds when the gears were not to be in mesh, the round belts were considered to be a good choice because they were simple, effective, and inexpensive. The belts were made from straight lengths of three-sixteenths inch Buna-N type o-ring material which were cut to length and adhesively bonded. Several belt sizes were made, all similarly constructed.

Two balancing discs were fabricated from one inch thick plexiglass in the shape of a gear blank. This shape was chosen in order to keep the weight concentrated near its periphery, and a series of holes spaced at intervals of ten degrees were drilled and tapped in the outer rim to accommodate cap screws that were used to create and alter the state and degree of imbalance. The specific use and arrangement of the holes was designed to allow the same assembly to be used for demonstrating a rotor balancing procedure, if desired, in future vibration laboratory sessions. The choice of plexiglass was made to ensure that the amount of imbalance required in order to have a noticeable affect would be minimal.

The bearing blocks were fabricated from three-quarter inch aluminum plate and measured four inches high by two inches wide. These dimensions are common to all blocks except for the two ball bearing blocks on the third shaft which required slight milling of one side of each block to provide extra clearance from adjacent gears. In each of five of the blocks there is a 0.875 inch straight bore located 2.75 inches from the bottom to accommodate the ball bearings. One of the blocks (the first one down from the top of the figure on the last shaft of the gear train) has a 1.25 inch bore to accommodate one of two special inserts that were made which were fitted with the undamaged and the intentionally damaged ball bearings, one each per insert. The use of the inserts enabled a quick means to change the bearings with minimal effort and movement of other components. The three remaining bearing blocks shown in the figure were each fitted with a nominal 0.375 inch bronze bushing; two of these blocks were the main bearings for the balance disc shaft, and the third block served as a steady bearing for the extended portion of the third shaft of the gear train. The bearing blocks could have been directly drilled and tapped to allow for stud mounting of the transducers, but it was desired to also test out the use of magnetic mounts which is the method practiced by the Navy in their machinery maintenance monitoring program. To satisfy this desire, small steel pads measuring one inch by three quarters of an inch by one quarter inch thick were fabricated and were drilled and tapped to accommodate the transducer mounting studs. These pads were finish ground and then affixed to the tops and sides of the bearing blocks (some side mounts were omitted due to clearance restrictions). In accordance with a special study regarding the attachment of transducer mounts, these pads were affixed using a cyanoacrylate ester compound (super glue) which reportedly gave the most favorable overall performance in the transmissibility studies that were conducted.

A three quarter inch plexiglass plate measuring 20 inches by 24 inches was used as the base for the assembly. Plexiglass was chosen because it was available on-hand in stock, was very easy to work with, and (for the thickness used) it was considered sufficiently rigid to serve as a suitable foundation and yet help minimize the overall weight of the assembly. A thin hard rubber mat was laid beneath the base to secure it from sliding and to prevent any vibrations which might be caused by the base resting on the hardwood countertop. Foam padding was initially used for this purpose, but it was found that this allowed excessive motion of the entire assembly which caused resonant frequencies of the model as a whole to appear in the spectral displays, especially during the imbalance studies. The use of the hard rubber mat avoided this.

3. Component Assembly

All ball bearings in the assembly were pressed into their housings (bearing blocks or sleeve inserts) and onto the shafts. All balance discs, gears, and pulleys were secured in place by means of set screws which were threaded through tapped holes in the hubs of the components to bear against the shafts. Each of the bearing blocks was secured by two 10-24 steel capscrews which were passed up through counterbored holes in the base. For the blocks of the gear train, these holes were made slightly oversized to allow for small alignment adjustments in assembling the train.

The input shaft was coupled to the three-eighths inch motor output shaft using a short section of one-half inch reinforced rubber hose that was force fit over the ends of the shafts. This was found to be sufficiently tight to transmit the torque of the motor without slippage and yet allow for small alignment inaccuracies as well as dampen out any motor related vibrations form the assembly. Two of the 15 tooth gears are carried on this shaft between two ball bearing supports; the near end of the shaft extends through the forward bearing and is fitted with a double groove pulley.

The intermediate shaft of the gear train carries two of the 50 tooth gears and one of the 15 tooth gears between ball bearing supports. Neither end of this shaft was extended to allow for individual drive via flexible belt since the size of the gearing made for very small clearances between bearing blocks and moving parts at this point in the assembly.

The final shaft of the train carries one of the 70 tooth gears at midspan between two ball bearing supports and extends forward through a steady bearing. The forward end of this shaft is fitted with either a small single groove pulley (as pictured in the plan view of the model) or the friction disc that was used to provide system load. Alternately, any combination of pulleys may be interchanged among the various shafts to suit the needs of the user, with the exception that there is insufficient clearance to mount the two large pulleys on the first and third shafts of the train simultaneously.

The fourth shaft carries the two balance discs between plain bearing supports and two aluminum collars just outboard of the discs which served to limit axial travel of the shaft. In use, the light weight of the assembly coupled with the freedom for axial travel allowed by the use of the sleeve bearings would tend to make the shaft assume an axial position as determined by the pulley and belt alignments. The collars were merely provided in anticipation of the possiblity of belt failure or disengagement in operation and were secured to the shaft with sufficient clearance from the bearing blocks to preclude any rubbing contact which could create additional events to appear in spectral measurements. The forward end of the shaft extends through the forward bearing and is fitted with another double groove pulley which has two different diameters to allow some versatility in the speed step up/step down arrangement that may be desired. Upon completion of the assembly, the discs were turned so that their locating set screws were symmetrically offset, then one was secured in place via its set screws. The other disc was then rotated just enough to bring its holes into index with those of the first disc and its set screws were then tightened. Finally, the pulley and the two collars were similarly adjusted to have a symmetric distribution of their set screws before they were tightened. This procedure placed the final assembly in a state of near-minimal imbalance which was considered sufficiently accurate to use as a baseline condition for future imbalance studies.

The mounting of redundant gears on the first and second shafts of the train allowed for one set to be retained in good condition while the other could have defects intentionally seeded in, and changeover from one set to the other could be accomplished with as little variation in the set up and alignment as possible, thereby improving the validity of comparisons to be made.

A close-up view of the gear assembly is shown in Figure 14 where the full train is seen in mesh, and with the defective gear on the intermediate shaft pictured in mesh with the site of its missing tooth appearing at the top of the gear.

B. TEST EQUIPMENT

An overall view of the model and measurement equipment layout is shown in Figure 15. Referring to the figure, the main components of the test equipment included a signal analyzer, a signal power amplifier rack, two accelerometers, a disc drive, an optical tachometer, a plotter, and a dc ammeter. A description of each piece of equipment follows.

1. Measurement Devices

For measuring shaft speeds, small pieces of reflective tape were attached to each shaft (or to components on the shaft) and an optical tachometer, AMETEK model number 1723, with an operating range of 100 rpm to 9999 rpm displayed the speed in terms of rpm on a LED readout. The dark coaxial cable leading from the back of the tachometer display and connected to the front of the analyzer was a later modification made to the unit which tapped into the once-per-revolution pulse signal it received from the optical probe and fed this as a trigger signal to the analyzer. This trigger was required in order to use the time domain averaging technique which the analyzer supports.

A dc ammeter (seen best in Figure 13), Weston model number 901, was used to read motor current on a scale of zero to one amperes. The motor rated current is 0.75 amperes; most experiments were run in the range of 0.25 to 0.50 amperes. This current reading was merely used as a relative load indicator. For the experiments conducted, the absolute value of the load was not required to be measured; it was only necessary that,

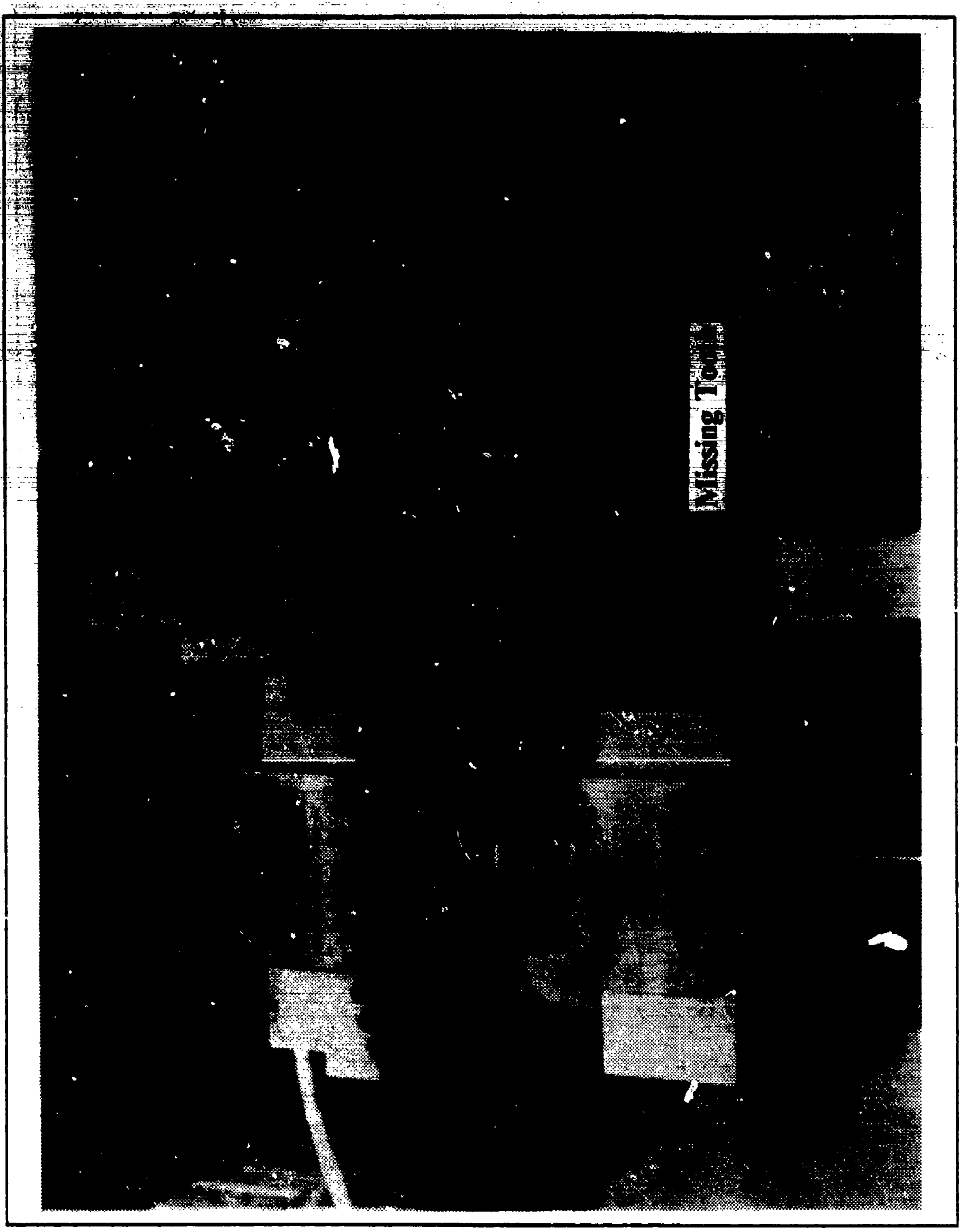

Figure 14. Close up view of gear train; defective gear in mesh.

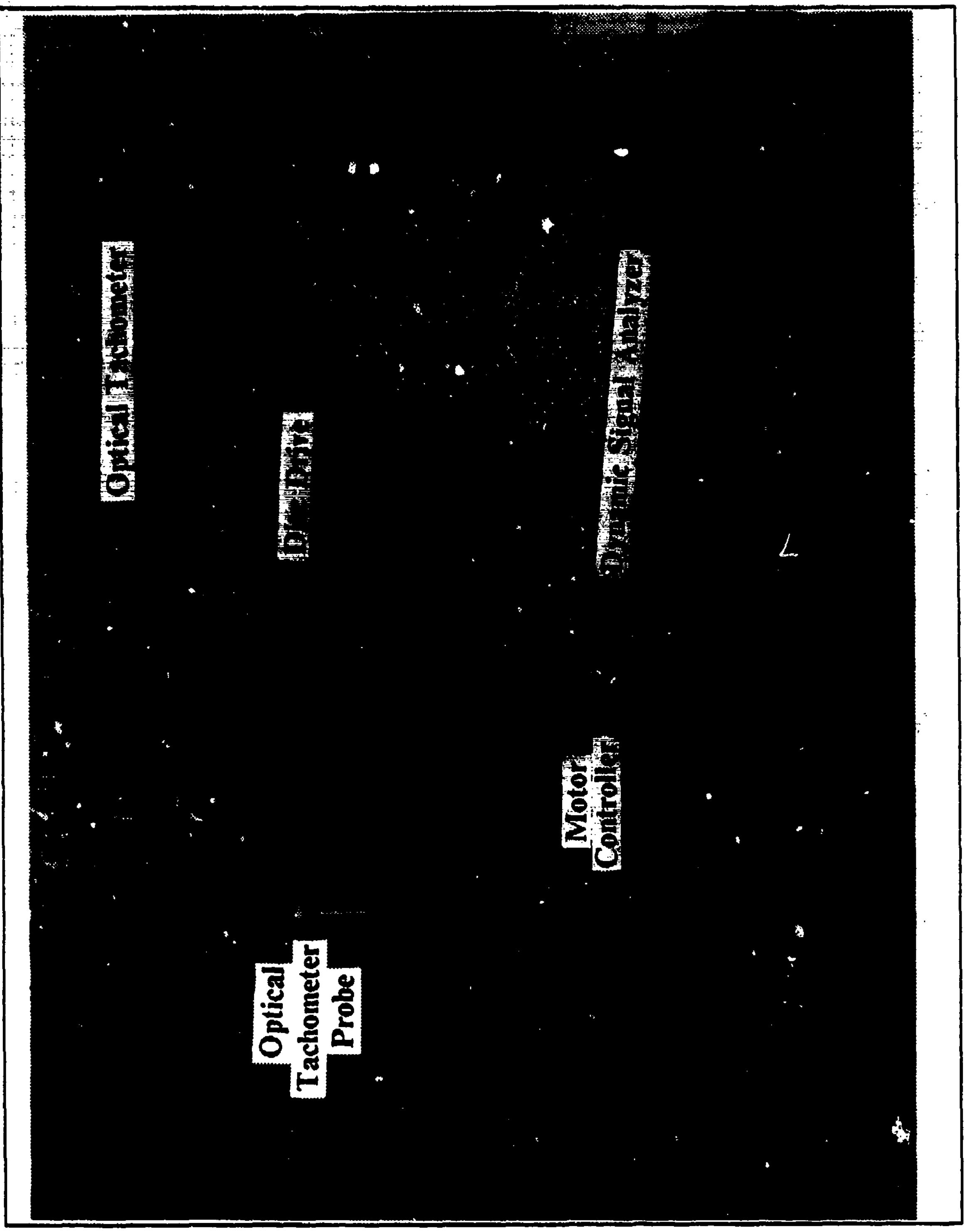

Figure 15. Front view of model and measurement equipment.

where repeated conditions were sought, there was a means to ensure that the loads were repeated as well.

The transducers used were PCB Piezotronics, Inc. model number 302A06 quartz accelerometers, ground isolated with built-in amplifiers, rated for a frequency range of 0.7 to 10,000 Hz, and with a voltage sensitivity of 10.01 mV/g. These were either stud mounted or, in some cases, magnetically mounted using the magnetic attraction bases (PCB model 080A27) provided with the accelerometers. In both cases, a thin coat of silicon grease was applied as a couplant/lubricant at all interfaces between the transducers and the mounts. The transducer cables used were PCB model 002C standard coaxial cables fitted with 10-32 micro and BNC plug connectors.

2. Data Processing, Display, and Storage

The transducer cables were connected to a PCB model 483A08 six channel power supply/amplifier fitted with individually adjustable gain settings from zero to 100. Gain settings of 10 on each channel were used throughout the experiments. The outputs from the amplifier were connected via standard coaxial cables to the input channels of a Hewlett Packard model 3562A two-channel dynamic signal analyzer (DSA). All subsequent operations of filtering, processing, display, storage, and output were controlled via the functions incorporated in the DSA. Data to be permanently stored was output to a Hewlett Packard model 9122 dual flexible disc drive, and plots were created by outputting to a Hewlett Packard ColorPro model 7440A plotter.

For several initial measurements, a PCB model 302B08 accelerometer, having the same specifications as the others, was used in conjunction with a battery powered PCB model 396B velocity vibration meter which measured overall vibration velocity in the frequency range from 10 to 10,000 Hz and displayed it on a digital LED readout in terms of decibels referenced to 10^{-6} cm/s. Two 10-32 micro fittings on the meter served as throughputs that could be directly connected to the analyzer channels. One provided an acceleration signal with voltage sensitivity of 10 mV/g, the other provided a velocity signal with sensitivity of 1000 mV/g. These were used on many of the initial single point measurements to directly provide an acceleration measurement on channel 1 and a velocity measurement on channel 2. For later work with two point measurements, the previous setup (accelerometers and amplifier) was used exclusively and any velocity measurements desired were obtained by integration and the other math functions available in the DSA.

C. MODEL TESTING

1. Initial Tests

In preparation for model testing, several specific initial tests were conducted to verify the performance of the measurement equipment and its arrangement. An additional test was also found necessary soon after the first model test had begun.

a. Transducer Calibration

To ensure that the newly acquired accelerometers had not been damaged in transit, a simplified calibration test was performed in order to compare their measured sensitivities to those reported in the literature which accompanied them.

A solid, cylindrical steel weight with a small through-drilled hole at the top was suspended by a short length of string that was connected to the midspan of four rubber bands which had been secured end-to-end. The ends of the outermost rubber bands were connected to solid supports which were spaced apart to provide tension in the bands. Each accelerometer was in turn connected to the bottom of the weight using the magnetic attraction base that was provided and connected to the analyzer via the power amplifier. The analyzer was then set up for a time domain measurement with a specified trigger level and delay. The point of attachment of the string to the rubber bands was then given a quick downward strike. This caused a 1g change in acceleration to be experienced by the weight and accelerometer. The maximum amplitude observed on the time domain display of this measurement indicated the amount of voltage generated for an acceleration of 1g, hence gave a direct measure of the transducer voltage sensitivity.

b. Transducer Mounting

Although there were studies conducted regarding the suitability of magnetic mounting of the transducers, a comparison test between stud and magnetic mounting performance was made to ensure that similar results would prevail in such use on this particular model, and also to verify that the steel mounting plates on the bearing blocks had been adequately prepared and affixed in place. A total of four measurements were made, 500 averages taken in each case. First, two stud mounted transducer measurements were taken, one using a Hanning window and one using a flat top window; then two magnetically mounted transducer measurements were taken, again one using each of the window types. The Hanning windowed measurements were compared for the accuracy and repeatability of the frequency resolution of the spectra, and the flat top windowed measurements were compared for the accuracy and repeatability of the amplitudes of the spectra.

c. *Tachometer Test*

A performance check on the optical tachometer was conducted in accordance with the manufacturer's instructions. This involved aiming the probe at a fluorescent light and observing a certain value on the digital display. The operating guide emphasized the importance of sufficient size and proper placement of the reflective tape used on the device being measured to ensure proper readings would be obtained. Consequently, to verify that the amount and application of reflective tape were correct, the rpm display of the tachometer was compared against shaft frequency readings obtained using the signal analyzer. An intentional imbalance was imposed on one of the shafts and a high resolution measurement was made using a Hanning window for frequency accuracy. The test was repeated several times at various speeds between 200 and 2000 rpm.

d. *Model Resonance Test*

Initially the model was supported on foam padding to isolate it from the hardwood countertop and to secure it from sliding. Shortly after commencing the imbalance tests it was noted that there were several frequencies that could not be accounted for. Several simplified resonance tests confirmed that resonant motion of the model as a whole was the problem. The tests were conducted by taking a single channel power spectrum reading and exciting the base of the model with a hardwood stick. Used as a crude but effective measure of the frequency response of the unit, it provided information regarding the frequencies at which the unit naturally would tend to vibrate as indicated by frequency bands in the spectrum where significant power levels developed when the unit was impulsively excited.

2. Machinery Fault Simulation Procedures

a. *Imbalance*

The balance disc assembly of the model was specifically designed for displaying imbalance and, therefore, was used for this simulation. With the shaft assembly having been pre-adjusted to obtain a state of minimal imbalance, the vibration signature of this state was obtained to serve as a reference, then intentional imbalances were created, measured, and the signatures compared to the original baseline signature.

The balance disc shaft was fitted with the two balance discs, two aluminum collars which served as physical stops to limit shaft axial travel, and a four-inch double groove pulley. When originally assembled, this shaft and its components had been adjusted to obtain minimal residual imbalance by the procedure described earlier under Component Assembly and had not been altered since that time. All the cap screws in the rims of the balance discs were removed and the optical tachometer probe was supported in the clamp stand and trained on the reflective tape on the balance disc shaft to provide the most accurate speed measurement. With the two 15 tooth gears on the input shaft locked in position out of mesh, a flexible belt was installed around the input shaft and balance shaft pulleys which gave a one-to-one drive ratio. The balance disc shaft was brought to, and held at, a speed of 899 rpm. Power spectrum and filtered line spectrum measurements were then taken covering a baseband span from zero to 100 Hz and using 500 averages, 90 percent overlap processing, and a Hanning window. The signal was measured and received as vibration acceleration, and the filtered linear spectrum display was then integrated to convert it to vibration velocity.

One cap screw was then inserted at the same location in each disc and the measurement procedure repeated. All other conditions were exactly duplicated except that the shaft speed fluctuated slightly between 899 and 900 rpm, with the latter being more prevalent. This procedure was repeated several more times until, finally, all the original cap screws were reinstalled, six per disc, evenly spaced. The shaft speed for the final run was steady at a value of 898 rpm.

b. Misalignment

Following the imbalance tests, the balance shaft was retained in its final state, i.e., with six cap screws symmetrically spaced in each disc. The far bearing block (away from the pulley) was then loosened, skewed horizontally relative to the shaft, and re-secured. This was done to intentionally create a misaligned bearing condition. With the signal analyzer in the same setup state (500 averages, 90 per cent overlap processing, and a Hanning window), the shaft was brought up to a speed of 1200 rpm. Two accelerometers were stud mounted to the bearing block; one in the axial direction and one in the vertical direction. Filtered linear spectra measurements were then taken and converted to velocity readings via integration as before.

c. Belt Defect

The misaligned bearing block was re-squared to the shaft, re-secured, and all screws were removed from the balance discs. To simulate a belt defect, a piece of electrical tape was wrapped around the glued butt joint of one of the drive belts which was then installed around the four-inch pulleys mounted on the input and balance shafts. An accelerometer was located in the horizontal direction on the bearing block adjacent to the balance shaft pulley. A filtered linear spectrum measurement was taken on a baseband of zero to 100 Hz using 500 averages, 90 percent overlap processing, and a

Hanning window. The shaft speed was maintained at 899 rpm and, again, the measurement was integrated to yield vibration velocity.

d. Mechanical Looseness

To simulate mechanical looseness the mounting screws for the far bearing block (away from the pulley) of the balance disc shaft were backed off slightly to permit motion of the bearing block relative to the base. The cap screws of the balance discs were left symmetrically distributed on their discs and the shaft brought up to a speed of 800 rpm. A filtered linear spectrum measurement was made with the two accelerometers in place as they had been for the misalignment test, and the analyzer setup state the same as well. Several variations on the arrangement were attempted to attain a distinctive signature. The mounting screws were tightened and re-loosened to various degrees, and the cap screws in the discs were repositioned closer together and in greater number to help excite motion of the block. Typical indications sought in reviewing the measured data were a large number of harmonics without significant peaks in the axial direction, along with truncations of the time domain waveform.

e. Bearing Defects

The model was fitted with six radial ball bearings, one pair for each shaft of the double reduction gear train. Two additional bearings of identical make and size were acquired, each to have an intentional defect imposed; an inner race defect in one, and a ball defect in the other. The bearings used were Fafnir model AS3K with seven balls per bearing and the following dimensions.

- inside diameter (bore)........................0.375 inch
- outside diameter................................0.875 inch
- pitch diameter...................................0.675 inch
- ball diameter.....................................0.155 inch

From equations (18) through (21), the bearing characteristic frequencies were calculated as follows, where f_s is the shaft frequency.[7]

- ball passing frequency, inner (f_i)......4.368 f_s Hz
- ball passing frequency, outer (f_o).....2.632 f_s Hz
- ball spin frequency (f_b)....................3.784 f_s Hz
- cage frequency (f_c)............................0.376 f_s Hz

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7 Note that if the frequency axis of a spectral display is converted into orders of revolution, then the coefficients of f_s become the characteristic orders.

To create the defects, the bearings were disassembled and a single defect made in each using a fine tooth file to generate a flat spot. The author's previous experience in evaluation of bearing condition was limitted to visual inspection and feeling for roughness in the bearing's motion. With no prior experience as to the size of defect required to create a significant spectral line, the defects were made just large enough that they would be noticeable upon visual examination.

Reassembly of the bearings was very difficult due to their small size and the lack of any tools on hand that may be specifically designed for this procedure. As a result, only one of the bearings was subsequently able to be tested. The bearing with the ball defect suffered from extreme binding which is presumed to have been caused by slight deformation of its outer race as it was being held in place to install and secure its other components. Although unfortunate, it was seen as a minor loss to the study since only about 10 percent of all ball bearing failures are attributed to faults in the balls or cages. A repeated attempt to obtain a ball defect sample was not felt justified due to the difficulties in assembly/reassembly of such small components, and the relative impact its exclusion might have on the thesis work as a whole.

The experiments involved acquiring vibration signatures for identical conditions with a different bearing installed at the monitored location for each run. The far bearing block of the third shaft of the gear train was the one specially fitted with the replaceable inserts which housed the bearings to be tested. With the good 50 tooth gear on the intermediate shaft in mesh, and the optical tachometer trained on the input shaft, the input shaft was brought up to a speed of 1800 rpm. Multiplying this by the reduction ratio of the gear set, a value of 115.7 rpm was calculated as the speed of the third shaft; therefore, the frequencies of interest were expected to be found at approximately 10 Hz and below. With a baseband span of zero to 50 Hz, 90 percent overlap processing, a Hanning window, and the number of averages set at 50, a power spectrum measurement was taken of the good bearing with the accelerometer stud mounted in the vertical direction.

f. Gear Defects

The examination of gear defects primarily concentrated on two specific areas; a missing tooth, and the spread of tooth wear. One of the 50 tooth gears on the intermediate shaft had one of its teeth removed by filing, and one of the 70 tooth gears had a successive number of tooth profiles slightly filed to simulate the spread of localized wear. The predominant phenomenon to look for in each case was sidebanding of the meshing frequency by the shaft frequency. As described by Smith [Ref. 17], the relative amplitudes of the meshing frequency and its sidebands gives an indication of the extent of modulation or damage. For both studies the same procedure was followed, only the frequencies of interest differed. For measurements of the first reduction gearmesh events, a transducer was mounted vertically on the bearing block of the intermediate shaft nearer to the defective gear. For measurements of the second reduction gearmesh events, a transducer was mounted vertically on the far bearing of the third shaft of the train. The signal analyzer was set for a small frequency span centered on the meshing frequency of the gear to be studied. Since it was decided ahead of time that the input shaft speed would remain at 1800 rpm, the meshing frequencies for the first and second reductions were calculated to be 450 Hz and 135 Hz, respectively. The flat top window was used in some cases and the Hanning in others, the choice depending upon the number of spectral lines appearing in the region of interest and their separation. A high number of averages were taken and the overlap processing and fast averaging capabilities of the DSA were used to advantage. It was expected that in each case the meshing frequency would be quite prominent; and that in the case of the missing tooth the impact event would cause high sidebands, whereas the gear showing wear would present smaller sidebands at first which grew with increased damage.

D. SPECIAL GEAR STUDY

1. Gear Fault Display Using DSA Math Functions

A new technique for displaying gear defects was explored. The technique, if successful, would provide a display of a selected gear in a gear train in a manner which would duplicate the gear profile on the screen of the DSA and, theoretically, would show sufficient details of profile irregularities to enable it to be utilized in identifying specific locations of gear wear and/or damage [Ref. 18]. The study was conducted in two parts. First, the concept was simulated using artificially produced (pure) signals obtained through use of the DSA internal signal generator, then an actual measurement was made on the model in an attempt to prove the practical application of the technique.

The technique involved obtaining time domain displays of the signals from two transducers mounted in mutually orthogonal directions in the radial plane of a gear train bearing and utilizing the math functions of the DSA to convert these into a single trace which contained both real and imaginary components. Assuming the gear mesh and shaft rotation events could be satisfactorily extracted and were contained in the measured signals, a Nyquist coordinate representation of the final combined signal would provide a detailed profile of the gear.

To isolate the gear of interest, the demodulation feature of the DSA was initially thought to be of use in that it deals specifically with the carrier wave/sideband wave phenomenon which characterizes the event and might, therefore, provide the required gear mesh plus shaft rotation temporal waveform. It was hoped that it might prove to be a single measurement which could provide a time domain waveform that was created strictly from the carrier and sideband waves, with minimal noise or signal content from other machinery events. Further review of the demodulation process and the results it provides showed that it would not be useful for this purpose. In place of this, it was attempted to obtain only the gearmesh event time domain signal from the model and combine this with a synthesized signal representing the shaft rotation event.

The first task was to verify the mechanics of the procedure by using simulated signals to see how an ideal result would appear. The simulation selected was to show a 50 tooth gear which was rotating at a shaft speed of 600 rpm. This would then involve a 10 Hz signal (the shaft frequency) and a 500 Hz signal (the gearmesh frequency). The first step was to obtain two time domain traces of one of the two signals and store them in local memory. For the simulation conducted, the 10 Hz signal was selected. With both channels of the DSA active, the source output from the DSA internal signal generator was connected to the channel 1 and channel 2 inputs. The frequency range selected for the measurement was 2 kHz. This selection was arbitrary; the only requirement was that the time domain of the measurement display be large enough to accomodate at least one full period of the final time traces to be displayed in the Nyquist coordinate system; less than a full period would give an incomplete plot. A Hanning window was used and, since these would be pure signals, only one average was needed to be obtained. The source signal was set as a 10 Hz sine wave with a source level (amplitude) of 1.0 volt. The measurement was then set to trigger on the channel 1 signal reception at a trigger level of 1.0 volt. To simulate the fact that the real measuremant made on the model would be done with two transducers which would be orthogonally mounted, a time delay equivalent to one fourth of a period was introduced on channel 2; for the 10 Hz signal this was calculated to be 25 msec. With all measurement parameters having been established, trace A was set to display the time record of channel 1 and trace B was set to display the time record of channel 2. The measurement was then commenced and the acquired traces were then stored in local memory, trace A stored in "saved data 1" and trace B stored in "saved data 2", as they are annotated on the DSA local memory softkey menu.

The next step was to obtain a similar measurement for the 500 Hz signal. The only changes in the procedure were to select a 500 Hz sine wave as the signal to be generated, to reduce the source level (signal amplitude) to a level which would be representative of the degree of modulation to be expected in the real system, to match the trigger level to this new source level, and to adjust the trigger delay to correspond to one fourth of the period of a 500 Hz wave. Accordingly, the source level was set at 25 mV (an assumed amount of modulation effect), the trigger level was set to 25mV, and the channel 2 delay was set at 0.5 msec. The measurement was then started and the time records obtained. The remainder of the procedure strictly involved the use of the math functions of the DSA in manipulating these waveforms.

The 500 Hz waveform on trace A was added to "saved data 1" (trace A of the 10 Hz signal) and then multiplied by the complex number (1,0) to make this signal a complex waveform with no imaginary part. Then the 500 Hz waveform on trace B was added to "saved data 2" (trace B of the 10 Hz signal) and then multiplied by the complex number (0,1) to make this signal a complex waveform with no real part. The two traces were then added together to create a single trace which now contained both real and imaginary parts which were non-zero. This trace was then displayed in the Nyquist coordinate system.

The next part of the study involved following this same procedure with a real signal obtained from the model relating solely to the gearmesh (carrier wave event) combined with an artificially produced signal which would represent shaft rotation (sideband wave event). Two transducers were mounted on the bearing block next to the undamaged 50 tooth gear; one vertically and the other horizontally. The frequency span was set to 10 Hz and was centered on 500 Hz. A Hanning window was used and time domain averaging was selected with the number of averages set at 5. With the intermediate shaft of the gear train assembly at a speed of 600 rpm, the measurement was commenced. When completed, the waveform math procedures detailed above were followed exactly as described; the 10 Hz waveforms were still in local memory from the previous simulation run.

V. RESULTS AND DISCUSSION

As a general comment on the figures which accompany the discussions that follow, and contrary to common practice in diagnostic work, line spectrum displays were not used exclusively throughout. Although machinery diagnostics measurements are predominantly in terms of line spectra, in many cases the power spectra of the measurements were used. This was done mainly for the sake of allowing displays and discussions of signature comparisons to be clearly viewed and followed by the reader, but also in part because of the low signal levels often measured. Many signatures obtained from the model were very low in amplitude, some so close to the noise floor of the measurement that line spectrum displays became too confusing and cluttered with the peaks created by the noise. Wherever signals were sufficiently elevated and distinct, or where actual amplitude values were of interest, filtered line spectra displays were used. Where it was only intended to show general features of the signatures, power spectrum displays were used.

For most measurements, an overlap processing value of 90 percent was used. This was done to allow the measurement process to run as quickly as possible in order to overcome the difficulties of maintaining constant apparatus rpm over long periods of time and thus avoid the attendant distortion and errors that are associated with speed drift. The windows used for the measurements were either Hanning or flat top, the choice made was based on the relative importance of obtaining accurate frequency resolution, the amount of speed drift that was experienced, and the clarity with which important features to be discussed would be seen in the final display that was to be achieved.

A. INITIAL TESTS

1. Transducer Calibration

The time domain measurements for the simplified transducer calibration tests are shown in Figure 16. The notations "Ov 1" appearing at the upper right hand corner of each trace indicates that an overload occurred on the channel during the measurement. This does not invalidate the measurements since, as can be seen in the figure, the signal clipping which occurred and caused the overload condition only affected that portion of the time signal beyond the point of interest. The clipping is shown by the flat portion of the trace beginning at approximately 100 msec. The

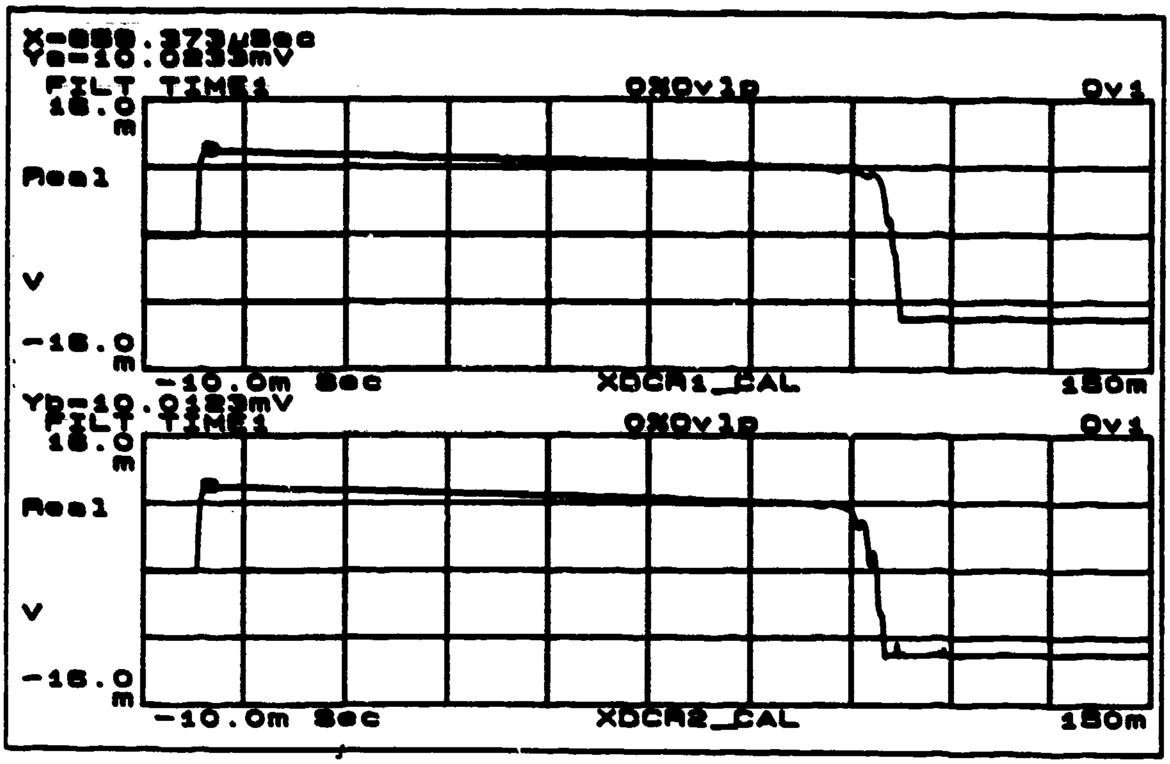

Figure 16. Transducer calibration results.

portion of the signal that was vital to the measurement, i.e., up to the first maximum amplitude and the subsequent roll off of this amplitude, was unaffected by the clipping and overload. The introduction of a negative time delay allowed the trace to be displayed away from the ordinate axis so that a clear and distinct view of the event was attained. The notations at the upper left hand corner of each trace indicate the "Y" values (amplitudes) associated with the cursor locations shown on the traces. The sensitivity as listed in the documentation which accompanied these transducers was 10.01 mV/g for each. The measured sensitivities of 10.0233 mV/g and 10.0123 mV/g compared well enough to the documented values to presume that the transducer calibration values had been unaffected by any adverse shipment or handling conditions which they may have experienced.

2. Transducer Mounting

The Hanning windowed measurements for the stud mounted and magnetically mounted transducer tests are shown together in Figure 17. Direct comparison of the two traces shows that there is no loss of frequency information in using the magnetic mount. The flat top windowed measurements for each attachment method are shown

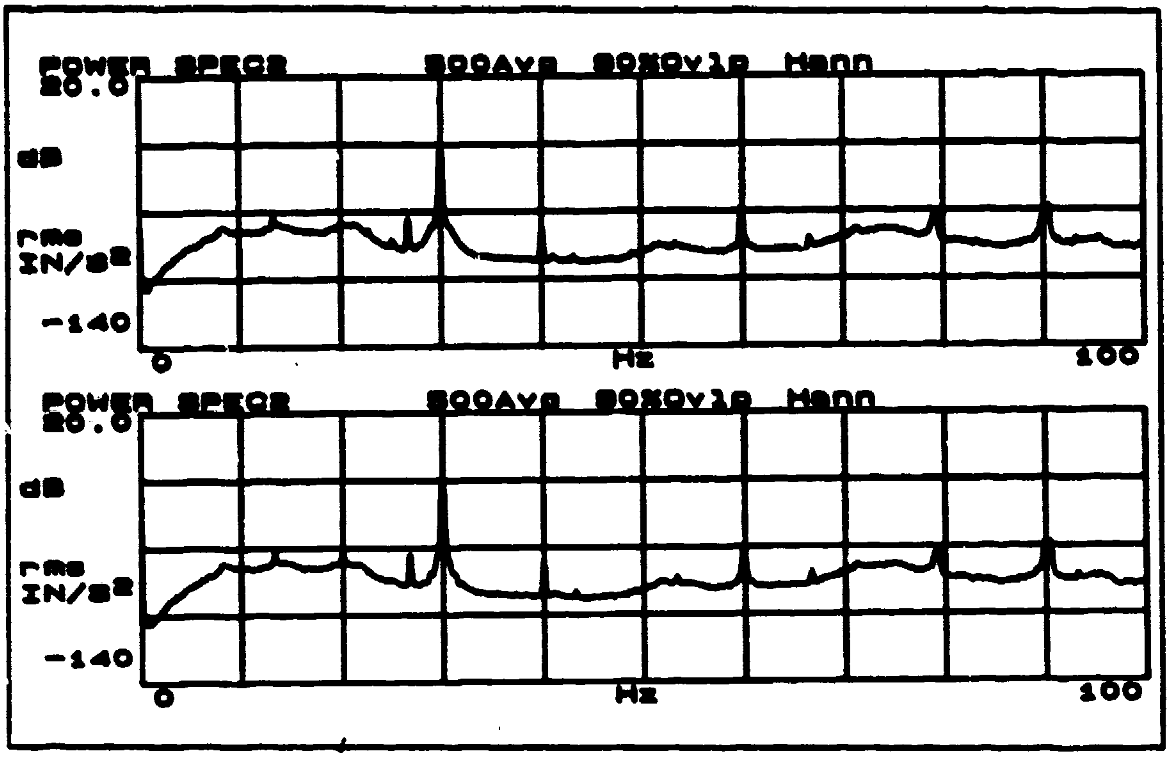

Figure 17. Transducer mount comparisons: stud mounted (upper) and magnetically mounted (lower) using the Hanning window.

together in Figure 18. The cursor controls were used to mark and measure the various amplitudes and found no appreciable differences between any associated peaks. The cursor marker was left on the pair of peaks which had the greatest disparity; the amplitude values are shown at the upper left hand corner of each trace. In both figures the upper trace is the stud mounted measurement and the lower trace is the magnetically mounted measurement.

3. Model Resonance Test

The result of the resonance test ("bump" test) of the model is shown in Figure 19. The marker in the figure is shown placed on the peak which was first noticed as an extraneous, unexplained frequency component appearing in the first imbalance tests. Replacement of the foam padding with a hard rubber mat eliminated the appearance of this component from future tests.

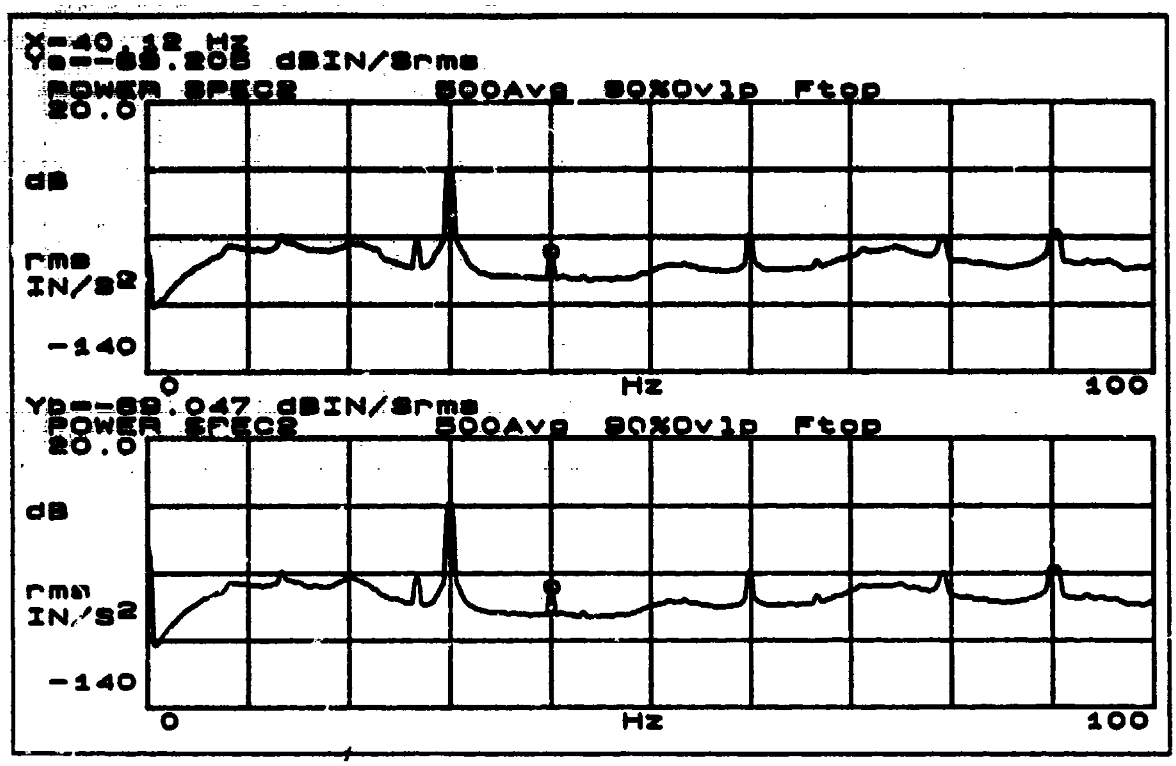

Figure 18. **Transducer mount comparisons: stud mounted (upper) and magnetically mounted (lower) using the flat top window.**

B. MACHINERY FAULT SIMULATIONS

In maintenance monitoring applications, the diagnostic procedures are usually limited to comparisons of spectral information obtained through use of a single transducer; namely, frequencies and relative amplitudes of signatures taken in an axial and two orthogonal radial directions. In work on an individual machine or component which requires special diagnostic attention, additional information is normally obtained by using a keyphasor or, more commonly, a second transducer which allows phase information to be included in the analysis. A technique common to both applications is a thorough review of the geometry of the machinery being analyzed before the analysis starts; often this allows the analyst to begin his/her task with a known list of frequency events to expect, or at least where in the spectrum to look for these events.

1. Imbalance

Rotating imbalance exists whenever the center (or centerline) of rotation does not coincide with the center (or centerline) of mass. All rotating equipment will have some degree of imbalance; well-balanced units will merely have very small amounts.

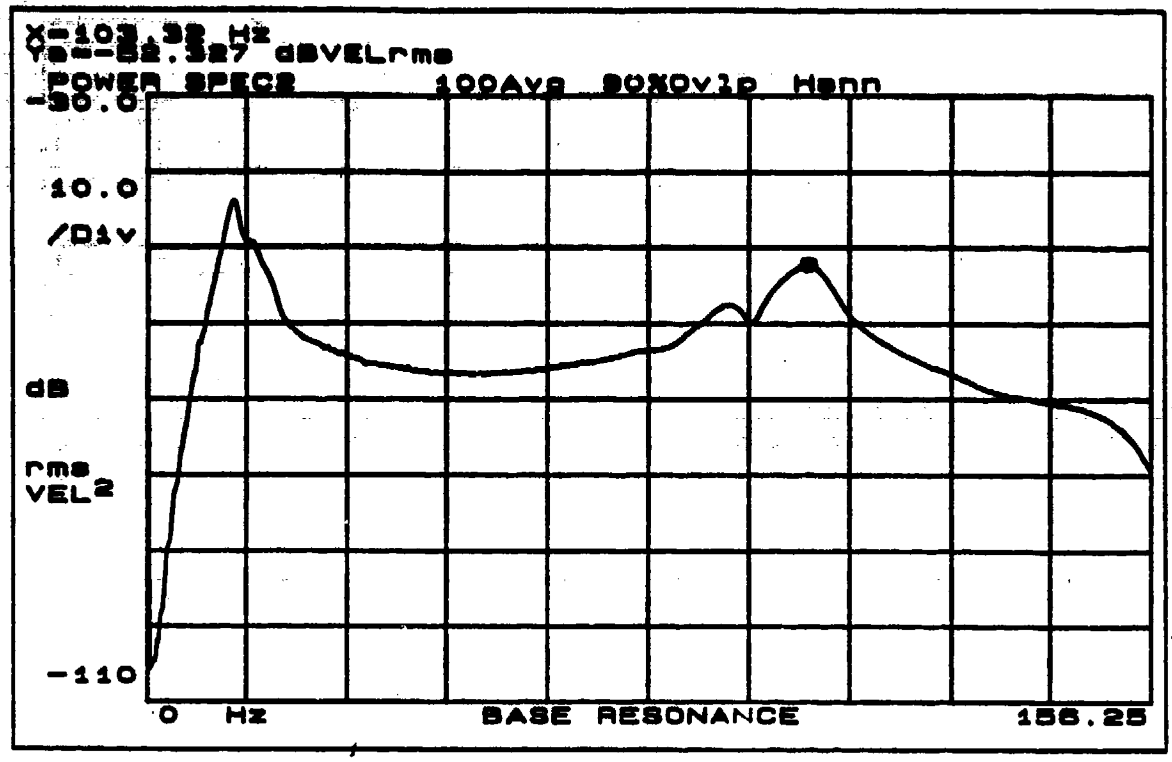

Figure 19. Model resonance (bump test) results.

Pure imbalance generates a spectral line appearing at the frequency of the shaft rotation and is ideally of the same amplitude in all radial directions and has no amplitude appearing in the axial direction. If phase readings are taken, the phase should track to the transducer location; i.e., there will be no fixed absolute phase angle for the event with respect to a fixed point on the machine. In practice, where measurements are taken on bearing casings, deviations from the ideal amplitude relationships mentioned are primarily due to assymmetric transmissibility characteristics of the casings (assymmetric geometry). Mathematically, the amplitude of a rotating imbalance is equal to $me\omega^2$ [Ref. 19]. This shows that it will vary with changes in mass (as when erosion or material deposit occurs), eccentricity (as when rotor sag is present), and angular frequency (as when the shaft speed changes). It is usually the change in amplitude with change in shaft speed that allows it to be distinguished from other faults which have similar signatures.

For measurements taken on the model, the baseline signature obtained for the balance disc assembly which shows the amount of residual imbalance, i.e., how much imbalance exists with all cap screws removed from the discs, is shown in Figure 20.

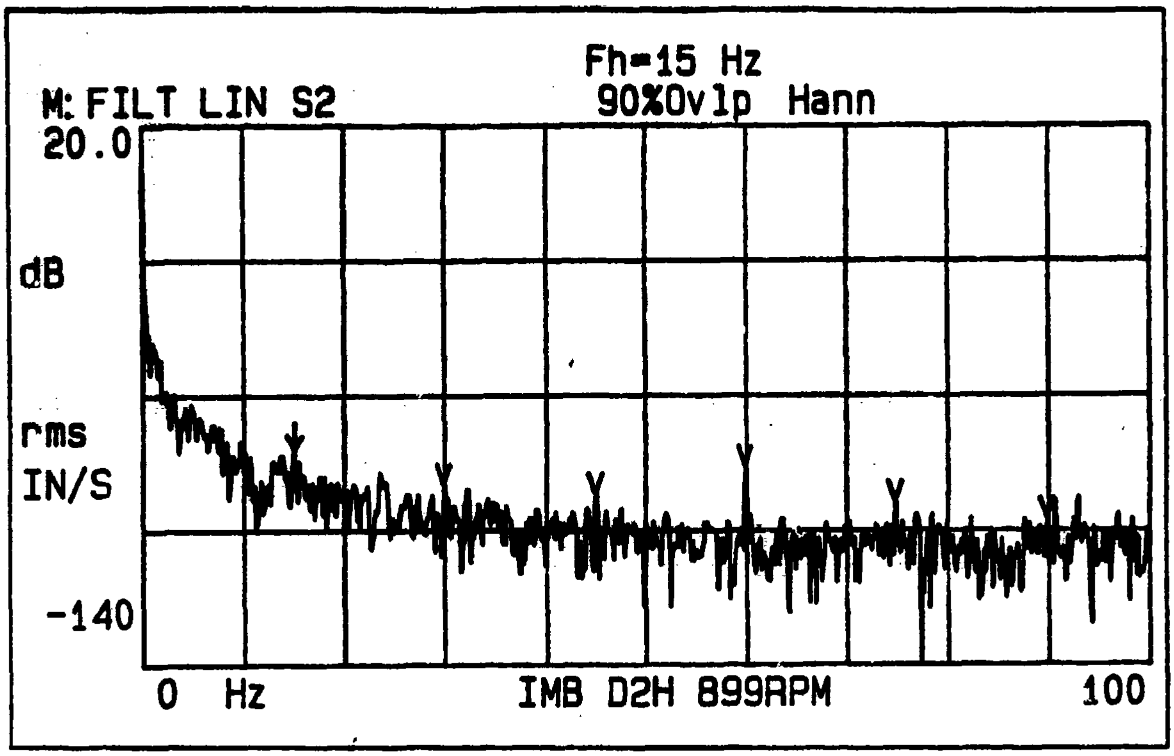

Figure 20. Baseline imbalance signature.

Figure 21 shows the effects due to the addition of one cap screw to each disc, and Figure 22 shows the effects when six evenly spaced cap screws are in each disc. The markers appearing in the traces were made with the harmonic special markers function of the DSA. The fundamental harmonic is marked with an arrow at a frequency of 15 Hz (as indicated by the notation above the trace) and all higher harmonics are marked with chevrons. Using the cursor controls, the 15 Hz peak in the baseline signature had a measured amplitude of -75.477 in/sec dBrms. With the dB scale referenced to 1.0 in/sec rms, this is equivalent to a linear amplitude of 0.0001683 in/sec rms, a value indicative of an extremely smoothe running device. Even without this measurement, the extreme low level of this amplitude is readily apparent by the fact that it is almost completely buried in the noise floor of the display.

Comparisons of the figures showed that the higher harmonic peaks were virtually unaffected by the added mass, but the fundamental frequency increased significantly with only one capscrew added, and reduced almost completely to its original amplitude when the six evenly spaced screws were inserted which served to reestablish the original symmetry and dynamic balance conditions. The amplitudes of the

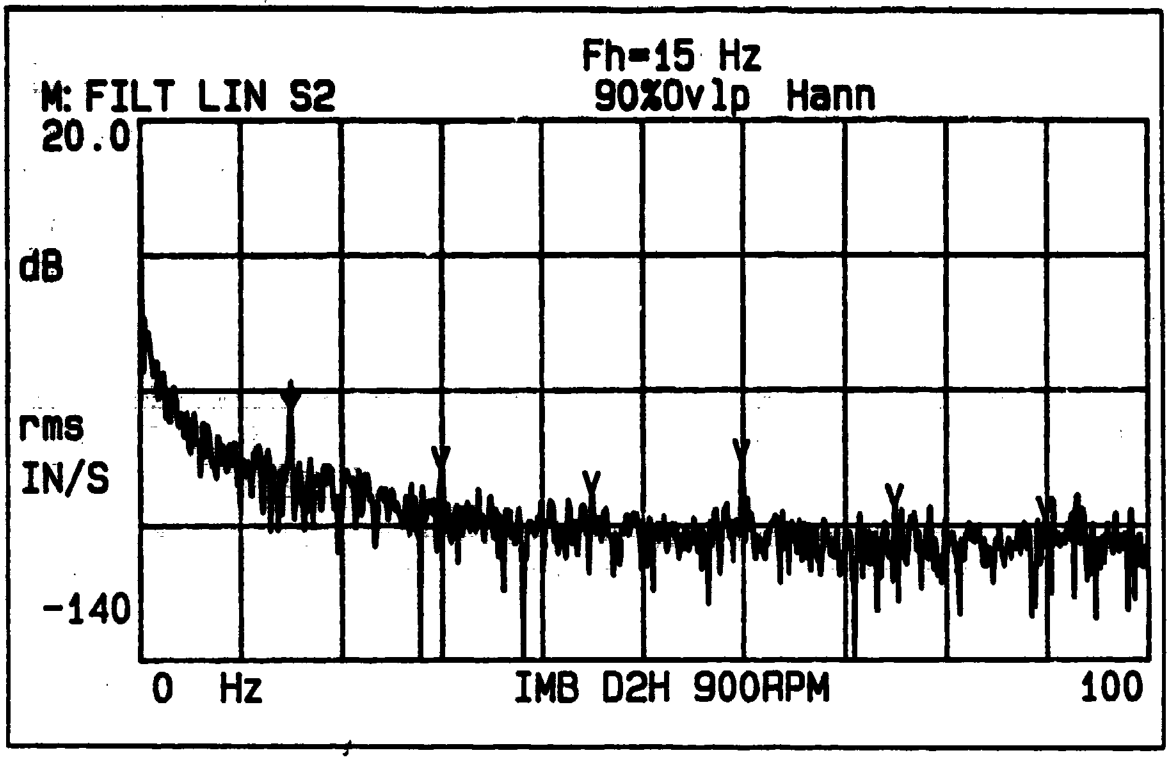

Figure 21. Imbalance signature with one imbalance mass per disc.

fundamental frequency peaks of the second and third figures were measured to be -64.558 in/s dBrms (0.0005917 in/sec rms) for the second and -74.802 in/s dBrms (0.0001819 in/sec rms) for the third. Although these are very small values on an absolute scale, the increase due to the addition of only two screws was by a factor of over 3.5 which is a respectable relative change in level.

The presense of the higher harmonics indicates that looseness and/or misalignment is also present, and it is impossible to tell from this signature alone which one it may be; an additional reading in the axial direction or a reading which could provide phase information is required in order to distinguish which condition exists. Another possibility in this case is excessive bearing clearance which can also give harmonics of shaft frequency [Ref. 10].

2. Misalignment

Misalignment includes bearing, shaft, or coupling misalignment, or a similar condition such as a bent shaft, and it is most often found as a consequence of improper assemly or installation. Misalignment is characterized by a large second harmonic of the shaft frequency in the radial and axial directions, axial phase readings across the ends

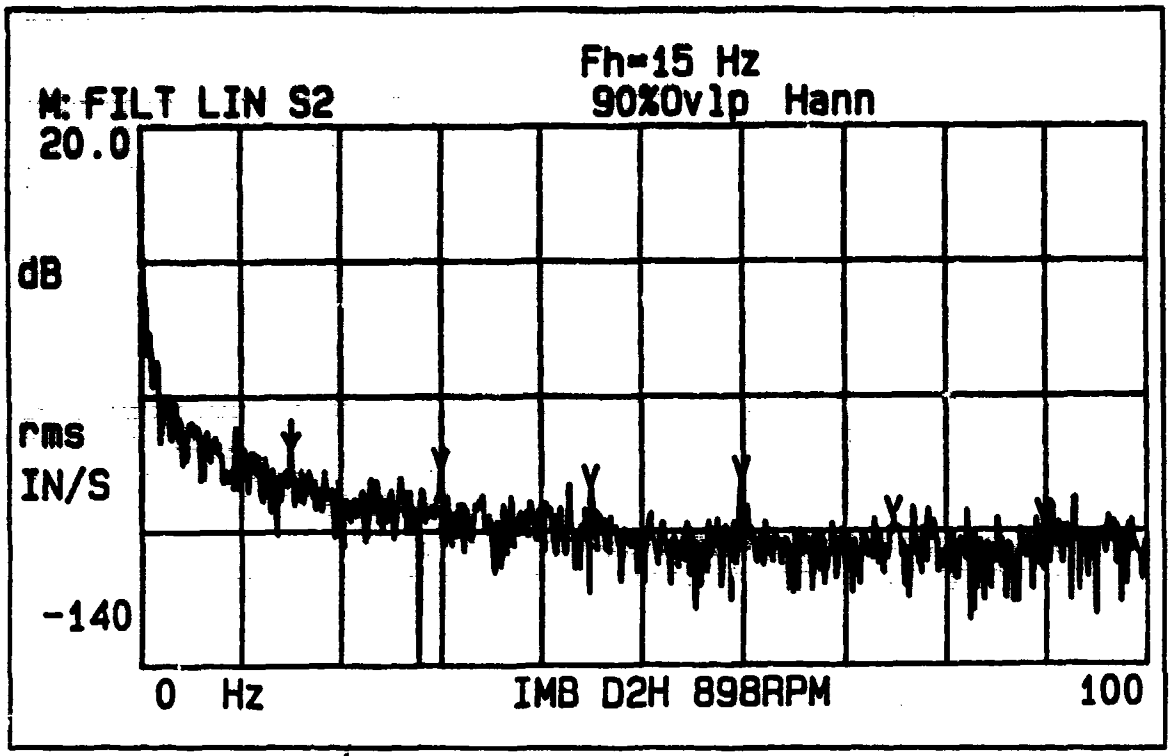

Figure 22. Imbalance signature with six imbalance masses per disc.

of a shaft or across a coupling that are 180 degrees out of phase, and may often present a large number of shaft harmonics.

The signatures shown in Figure 23 show the distinctive features of this machinery fault. The vertical measurement clearly shows an imbalance at 20 Hz, a significant peak at the second harmonic, nothing at the third harmonic, and a minor peak at the fourth harmonic. The high second harmonic peak is a rather solid indicator of misalignment by itself, especially in the absense of a third harmonic which, if present, would have made looseness a possibility to consider. In the lower graph, the high level of axial components is a decisive factor in diagnosing this as misalignment. The relative amplitudes between the first and second harmonics may be used as a rough measure of the severity of the condition. The use of axial phase readings taken at the ends of a shaft or across a bearing where a shaft bend has occurred can identify these faults, but this is only applicable to rigid rotors, i.e., rotors which operate at speeds well below their critical speeds.

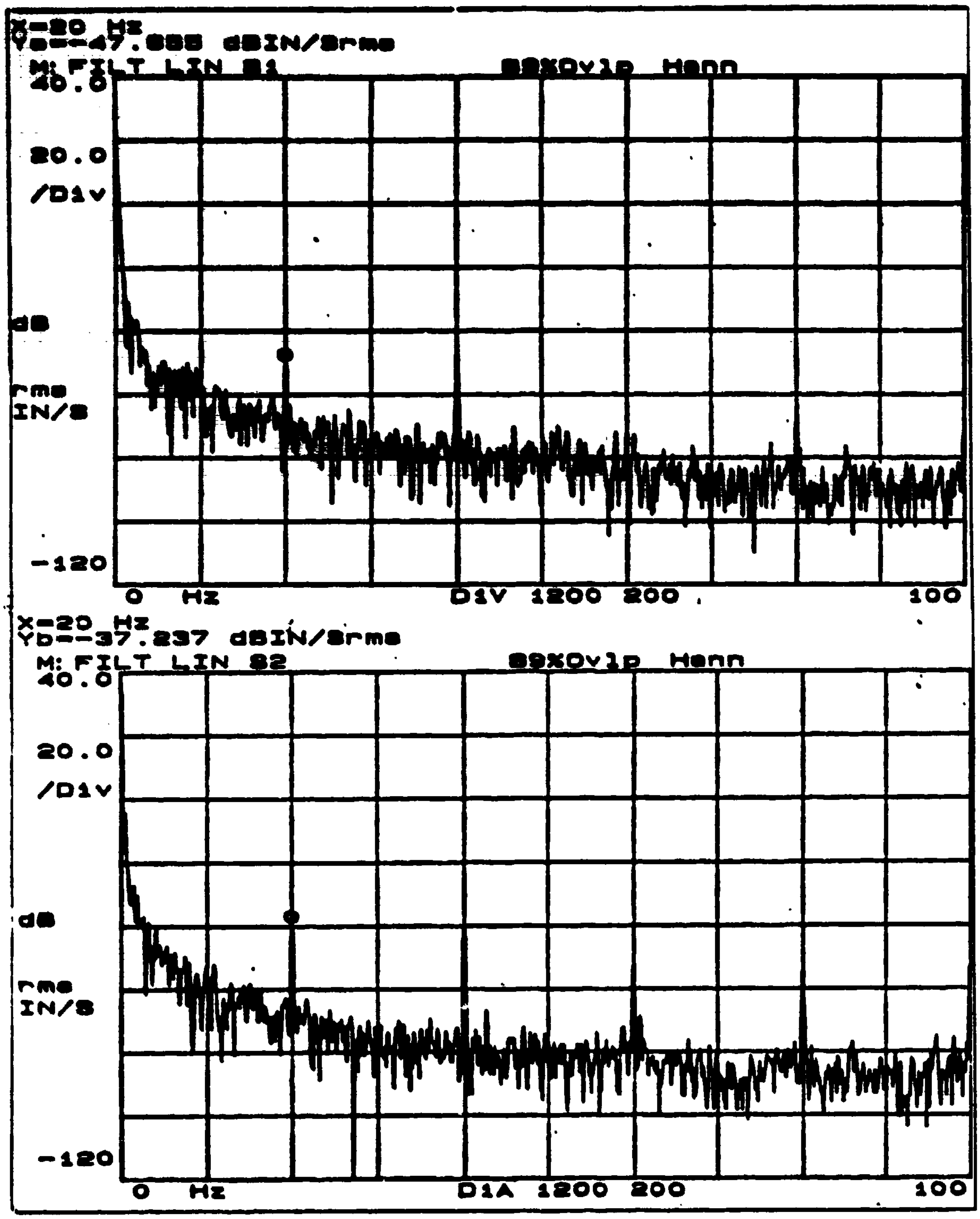

Figure 23. Misalignment signatures: vertical direction (upper) and axial direction (lower).

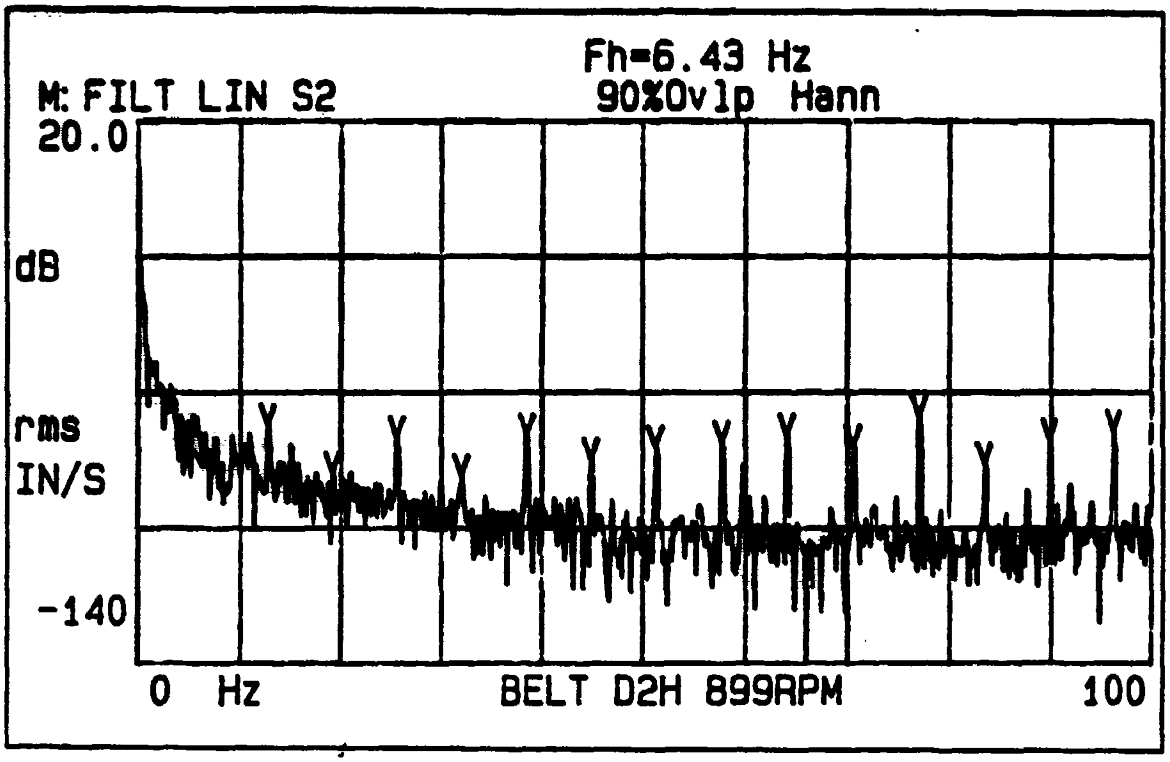

Figure 24. Drive belt defect signature.

3. Drive Belt Defect

A drive belt defect will appear in the spectrum as a line at the frequency of revolution of the belt, or sometimes at twice this value. For rapid diagnostics on an assembly containing belts and pulleys, a stroboscopic light with adjustable flash rate, or one which can be triggered by the vibration event itself (some analyzers have this feature) can be very useful in rapidly locating the problem component. The spectral line created will only appear in measurements taken in radial directions providing there is no misalignment, and phase readings are not needed.

The filtered linear spectrum display for the belt defect of the model is shown in Figure 24. The span between the pulley centerlines is eight inches which makes the belt frequency equal to $[4\pi / (16 + 4\pi)]\,(f_r)$, or 0.4399 times the shaft frequency. For the given shaft speed of 899 rpm, the shaft frequency was 14.9833 Hz which gave a calculated belt frequency of approximately 6.59 Hz. Using the special markers function, the fundamental frequency was adjusted until each of the chevron markers appeared on a peak, and this occurred when the fundamental frequency was 6.43 Hz. The difference between this value and the calculated value is due to the fact that the pulley outside diameter was used in the formula as an approximation whereas the belt actually travels in the groove which has a slightly smaller diameter.

The ability to retrieve a signal such as this that was intentionally implanted was not difficult and, perhaps by itself, not too interesting. However, a feature of this signature that is significant is one which appears in other fault signatures and worthy of special note. Many harmonics are seen to appear, yet the fundamental frequency is completely undetectable: in this case it is so low in level that even its marker arrow is buried in the noise floor of the trace. This same condition can occur with faults such as rolling element bearings [Ref. 10] and others as well. Being aware of this phenomenon helps in the analysis of signatures which may have many harmonic multiples of different events, especially if they may be overlapped in the spectrum display.

4. Mechanical Looseness

Mechanical looseness includes all forms of relative motion between components which are not designed to be present, usually found where bolts have been overlooked or improperly tightened, or have vibrated loose in service. It is characterized by a large number of shaft harmonics and sometimes subharmonics, and it is highly directional. Therefore, the amplitude will be greatest when the transducer is nearest the location of the fault and in alignment with the direction of the looseness. If flexible belts are in the assembly, the harmonics may be damped out and leave only the once per revolution spectral peak which appears as an imbalance [Ref. 10].

The tests conducted to detect mechanical looseness gave results which were identical to those obtained for misalignment in that none were able to be obtained which did not have axial components and, therefore, the appearance of misalignment indications precluded the ability to declare any specific traces as showing only mechanical looseness. Many traces obtained may actually have mechanical looseness as a part of them, but it was not possible to distinguish this fault separately and distinctly by itself in any trace.

5. Bearing Defects

Approximately 90 percent of all rolling element bearing failures are due to defects in one of the raceways, and 10 percent are due to defects in the rolling elements or the cages [Ref. 20]. The defects typically begin as microscopic pits where Hertzian stresses have caused local surface hardening and spalling of the material. As the small pits grow in size, the transit of the damaged region becomes an increasingly impulsive event. Consequently, both the frequency of the impact event and the bearing natural frequency may show in the spectrum. Knowing the geometry of the bearing and the

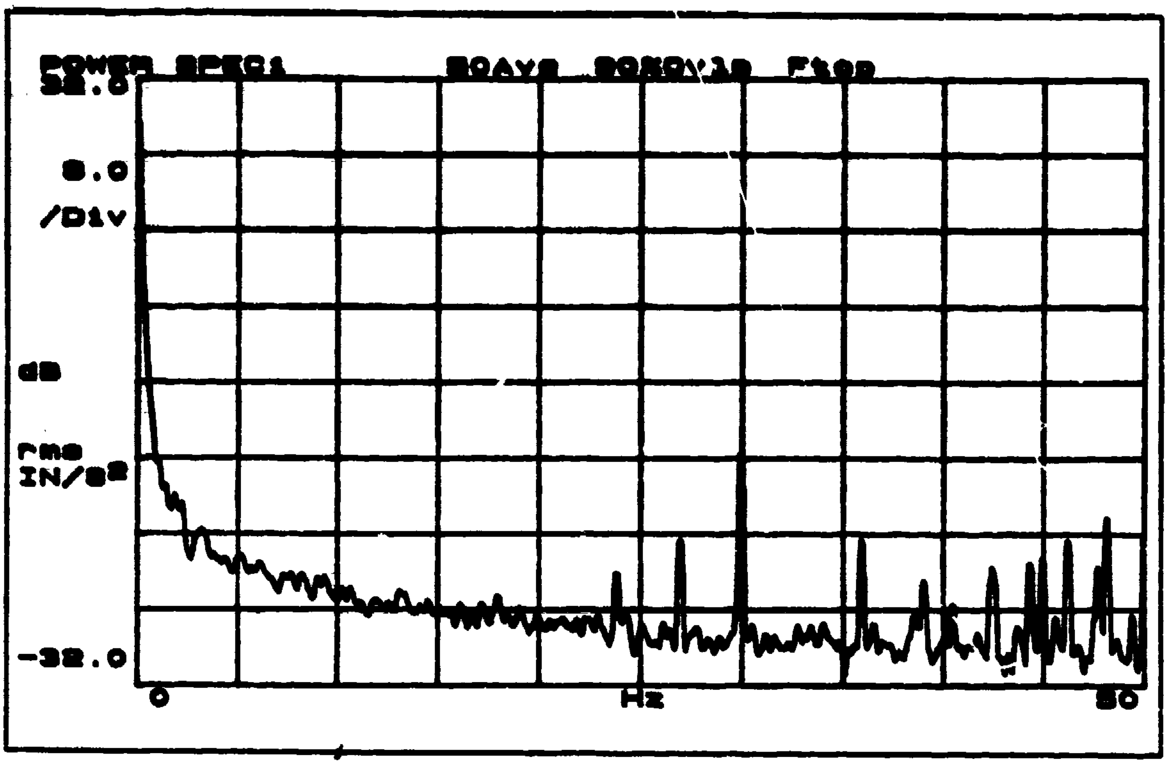

Figure 25. Signature of the defect-free bearing.

shaft rotational speed, there are four characteristic bearing frequencies that can be calculated: the outer ball passing frequency, equation (18), the inner ball passing frequency, equation (19), the ball spin frequency, equation (20), and the cage frequency, equation (21). The ball spin frequency formula has a factor of two already included in it to account for the fact that there are two impact events per ball revolution, one with each race.

$$f_o = (n/2)(1 - \frac{BD}{PD} cos\alpha)(f_s) \tag{18}$$

$$f_i = (n/2)(1 + \frac{BD}{PD} cos\alpha)(f_s) \tag{19}$$

$$f_{bs} = (n)(\frac{PD}{BD})[1 - (\frac{BD}{PD} cos\alpha)^2](f_s) \tag{20}$$

$$f_c = (1/2)(1 - \frac{BD}{PD} cos\alpha)(f_s) \tag{21}$$

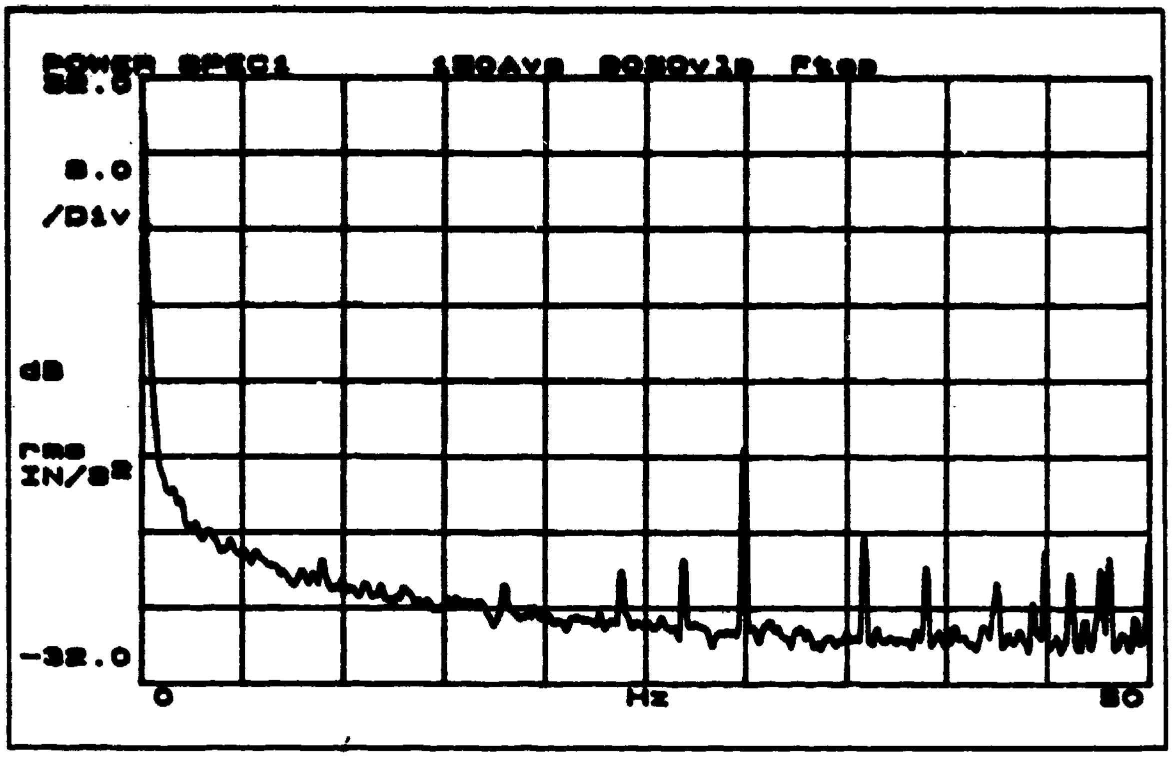

Figure 26. Signature of bearing with inner race defect.

where:

n = number of rolling elements in the bearing

BD = ball diameter

PD = pitch diameter of the bearing

α = contact angle

f_s = shaft frequency in Hz

The spectral signature varies depending on how far the damage has progressed. For an undamaged bearing, all of the above frequencies may be found, but they will be at extremely low amplitudes. As a defect sets in, the frequency associated with the defect location will begin to show. As the defect worsens or the number of small defects increase, the events become more impulsive and random and cause the natural frequency of the bearing to be excited which gives a small "area" of amplitude on a spectral display as opposed to one or more lines.

In hydrodynamic bearings, the main problems which arise are oil whirl and oil whip. Oil whirl occurs when the pressure difference (oil pressure) across the load region of the bearing causes rotor precession, and this will generate a spectral line at slightly less than half the shaft frequency. As the shaft speed increases and approaches twice its critical speed (meaning the oil whirl condition is now occurring near the shaft critical speed), the oil whirl changes to oil whip which may cause the oil to lose its ability to support the shaft due to the large vibrational forces generated.

The tests conducted to detect the bearing characteristic frequencies were successful in that the characteristic frequency for an inner race defect was found appearing in both the damaged and the defect-free bearings. Figure 25 shows the power spectrum reading taken for the good bearing. As can be seen there is a predominant peak at the shaft frequency of 30 Hz. Also showing are a number of other discrete peaks in the upper half of the baseband. As was the case in the drive belt tests, these upper range peaks are harmonically related to an event which is of too low an amplitude to appear. The signature obtained for the inner race defect bearing is shown in Figure 26 and looks almost identical to the signature for the good bearing, including the appearance of the harmonics. The main distictive feature is the very small peak just below 10 Hz. This peak is at a frequency of 4.5 times shaft frequency and it is the harmonics of this that are appearing throughout the spectrum of each. The calculated characteristic frequency for an inner race defect was 4.368 times the shaft frequency. These signatures indicate that both bearings have inner race defects.

6. **Gear Defects**

As with bearings, gears have characteristic frequencies which can always be found, but are not always high in amplitude. The predominant frequency is the gear mesh frequency which is equal to the shaft frequency times the number of teeth on the gear. The exact number and size of spectral lines vary depending upon the specific problem and its severity, but a general signature is one where the gear mesh frequency is sidebanded by a series of spectral lines at shaft frequency. The relative amplitudes of the lines are indicative of the degree of damage, but the amplitude of the gearmesh frequency peak alone is not meaningful since it may normally alter due to changes in operating conditions. Again, as with bearings, the gear natural frequencies may also appear in the spectrum which indicates that a problem exists of an impulsive nature.

For the first gear damage test done on the model, the good 50 tooth gear was run at a shaft speed of 540 rpm which gave it a calculated gear mesh frequency of 450 Hz, and a shaft frequency of 9 Hz. As expected, the initial signature, seen in Figure 27, showed the gear mesh frequency as a predominant peak sidebanded by peaks at 9 Hz spacings. The 50 tooth gear with the one tooth missing was expected to have

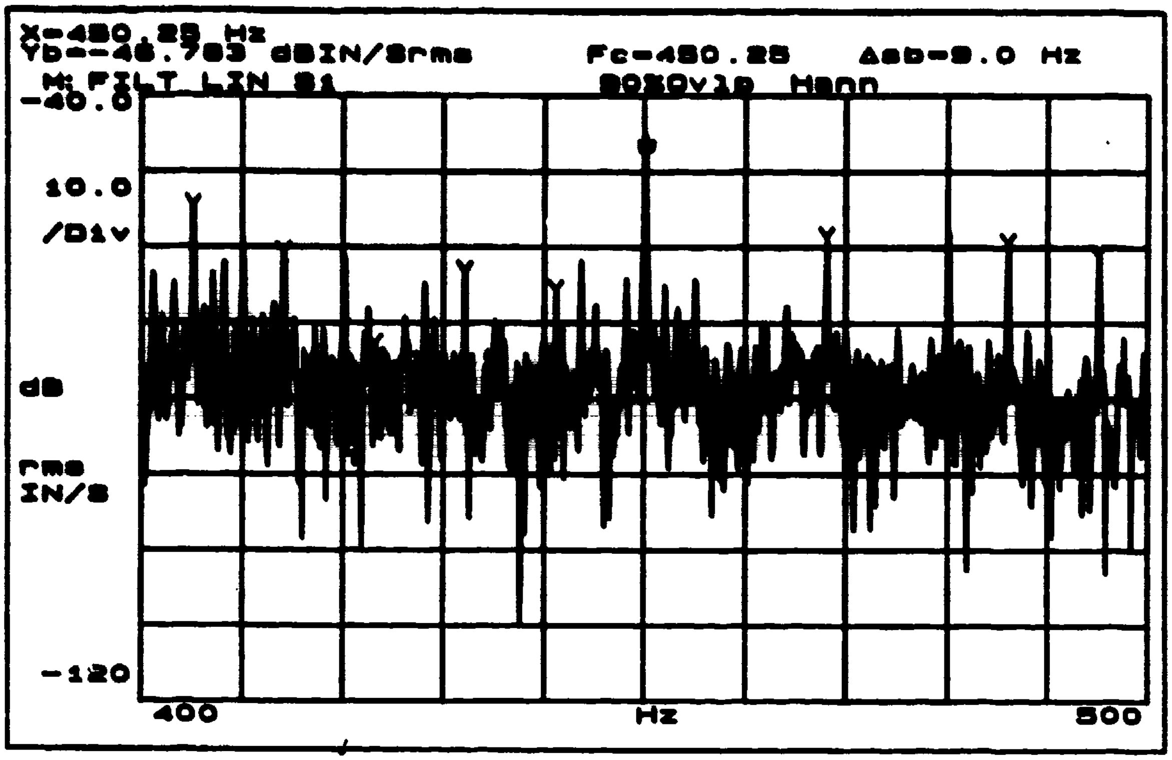

Figure 27. Signature of first reduction gear set, no damage.

a similar signature but with higher sideband amplitudes. As shown in Figure 28, this was not exactly achieved. Although most sideband amplitudes remained the same or increased slightly, two of the upper sidebands were found to diminish into the noise floor of the measurement. The difficulty in these, as in many other measurements, was the very low level of the signals and their strong dependence on the level and steadiness of the load.

Test results for the second part of the reduction set were closer to what was expected. Figure 29 shows the signature obtained when only one tooth profile had been slightly shaved. The gearmesh frequency was found to be of equivalent amplitude to almost all sideband levels. In Figure 30 where two tooth profiles have been filed, all levels have elevated slightly, and the gearmesh frequency begins to stand out significantly from the rest of the events. Figure 31 was taken after three tooth profiles had been filed, and the signature may be seen to appear more similar to the original trace. Finally, in Figure 32 where four teeth had been filed, the signature begins to appear as would be expected for a case of advancing gear wear.

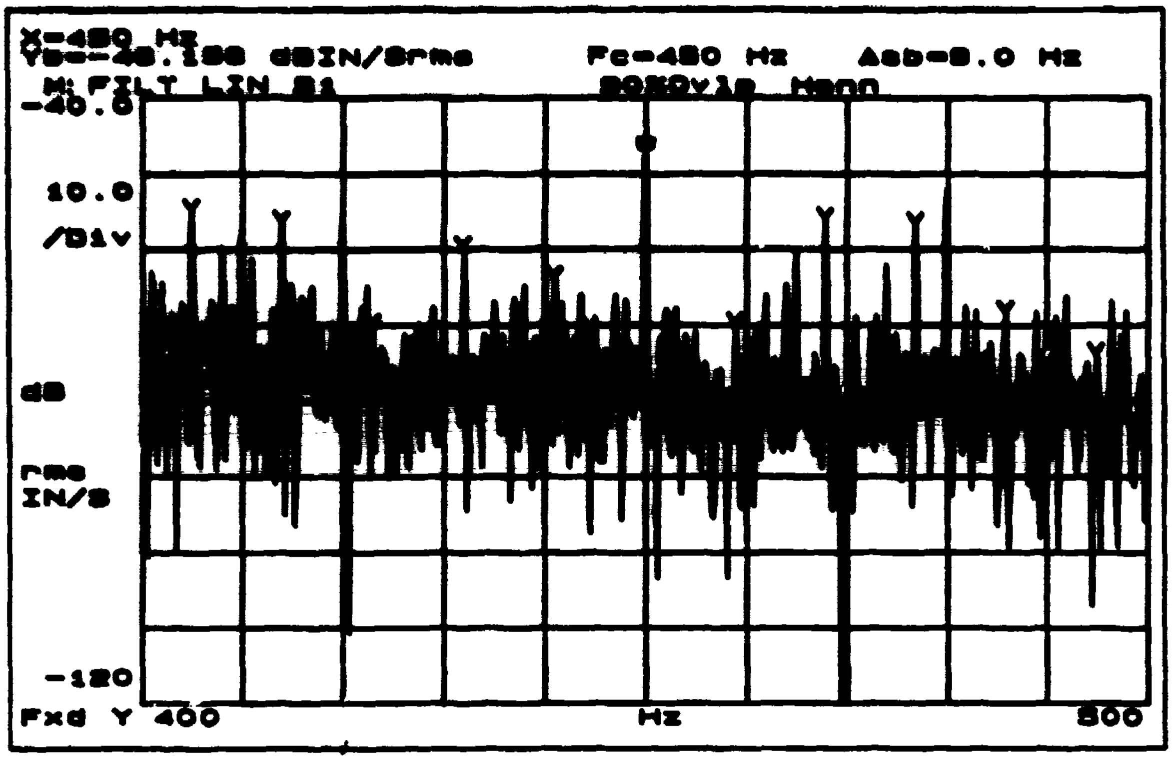

Figure 28. Signature of first reduction gear set, one tooth missing from gear.

One collection of gear train readings were of particular interest, not only because they appeared so predominantly, but because of the condition which created them. Figure 33 and Figure 34 show high resolution measurements of an event which at first could not be accounted for since it occurred at a frequency that was apparently unrelated to any component or condition known to exist in the model. All three signatures show the classic sidebanding pattern, but they are all centered about a carrier frequency of 1350 Hz. This frequency is not the gearmesh frequency normally calculated, but it is a gearmesh frequency of a particular meshing event; the frequency with which the 15 and 50 tooth gears become re-indexed to one another. The 15 and the 50 tooth gears cycle through 150 tooth meshes to become re-indexed; i.e., to have the same two teeth in mesh again. For the 50 tooth gear, this can be calculated to be a frequency of (50 x 540)/60 = 1350 Hz. As seen by the chevron markers in the figures, there are actually two sidebanding frequencies of this meshing event, one of 30 Hz and one of 3 Hz. The 30 Hz sideband is, of course, the frequency corresponding to the input shaft speed. The 3 Hz sideband represents the frequency with which one cycle of re-indexing is occurring. Since the gear teeth are all the same, the only outstanding

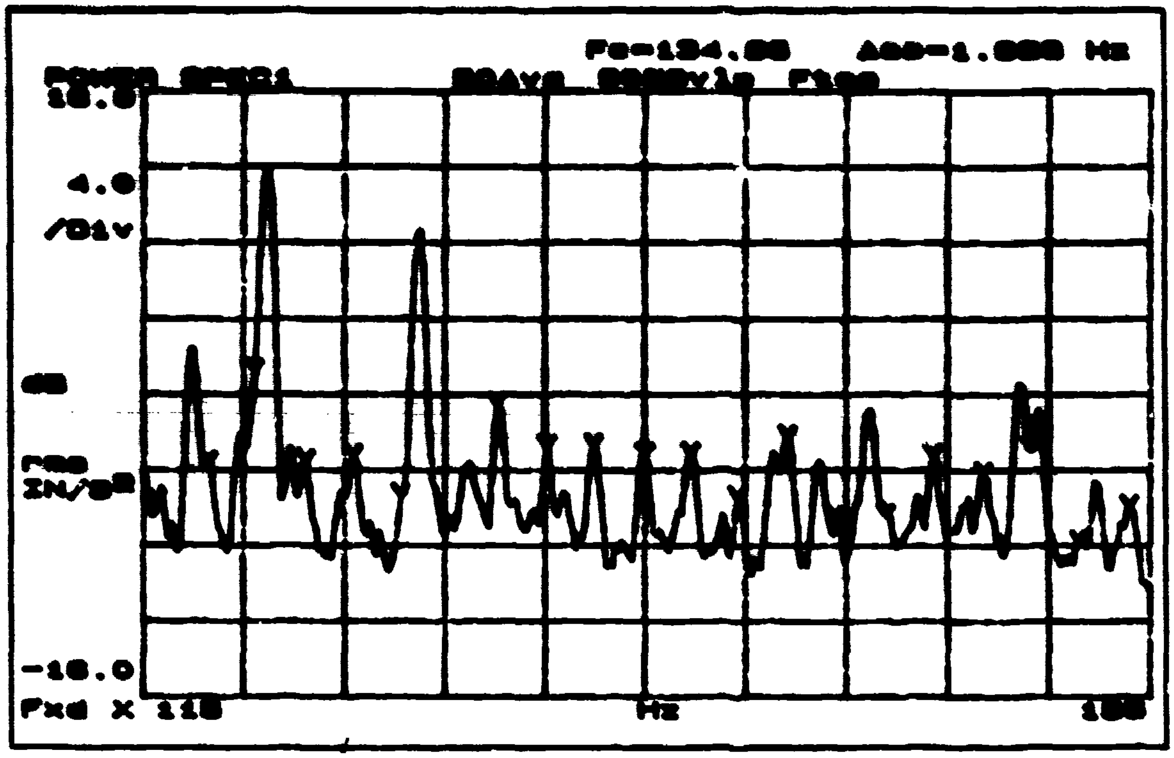

Figure 29. One gear tooth filed.

feature on the gears which was presumed to be involved with this indexing was their locating set screws. The use of set screws to fix the gears to the shaft results in their centerlines being non-coincident with the shaft centerlines which will cause the degree of mesh to cycle from some maximum amount (when their set screws are pointing towards one another) to some minimum amount (when the set screws are 180 degrees out). This will cause an amplitude modulation due to the change in surface contact area between tooth flanks in mesh, and would most likely be a very predominant feature if these teeth were to be heavily loaded.

In all the tests, the two most variable and uncertain parameters were the affect of the amount of load on the teeth, and the amount of noise contamination of the signal. Although the loads only varied very slightly (less than plus or minus 0.05 amperes on the ammeter scale), the actual value of the load may have been too small for the gears in question to show realistic relative amplitudes between carrier and sideband signals. The extremely low level of the signals also leaves doubt as to exactly how much noise was contaminating the signal.

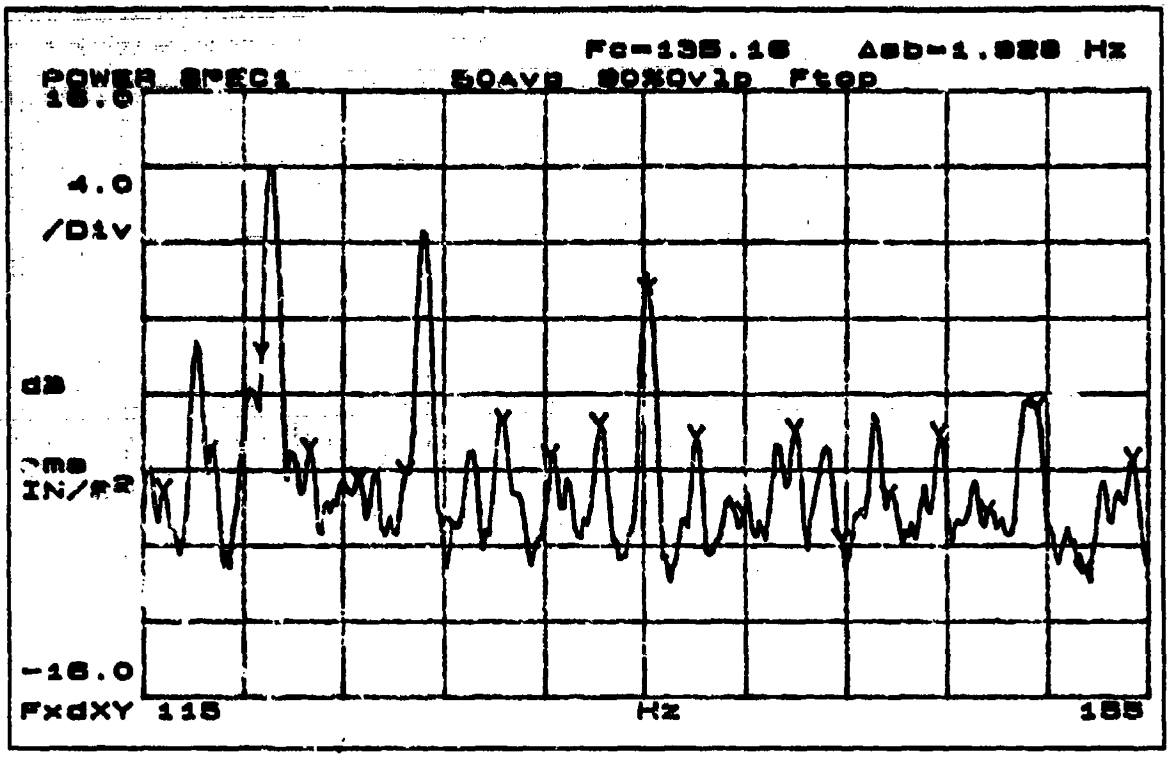

Figure 30. Two gear teeth filed.

C. SPECIAL GEAR STUDY

The result of the first step of the gear display technique is shown in Figure 35. Here the 10 Hz waves are seen to be identical except for the required 90 degree phase difference created by the 25 msec delay that was imposed on channel 2. Figure 36 shows the result of adding the 10 Hz and 500 Hz waveforms together. At this point, the waveforms are each still real valued functions. Waveform math is needed to make them be recognized by the analyzer as complex. To accomplish this, the upper trace was multiplied by (1,0) and the lower trace by (0,1). The result is seen in Figure 37 where the upper trace is now a complex waveform with only a real part, and the lower trace is a complex waveform with only an imaginary part. Note that when this multiplication is performed, the abscissa scales are automatically reduced to half their original value. This will limit the size of the frequency span selected for the initial measurement to some minimum value so that at least one full period of the waveform is retained up through this point in the process. The final result was obtained by summing these two waveforms to create a single waveform and then switching to a Nyquist coordinate system display; this result is shown in Figure 38. Although the display has some

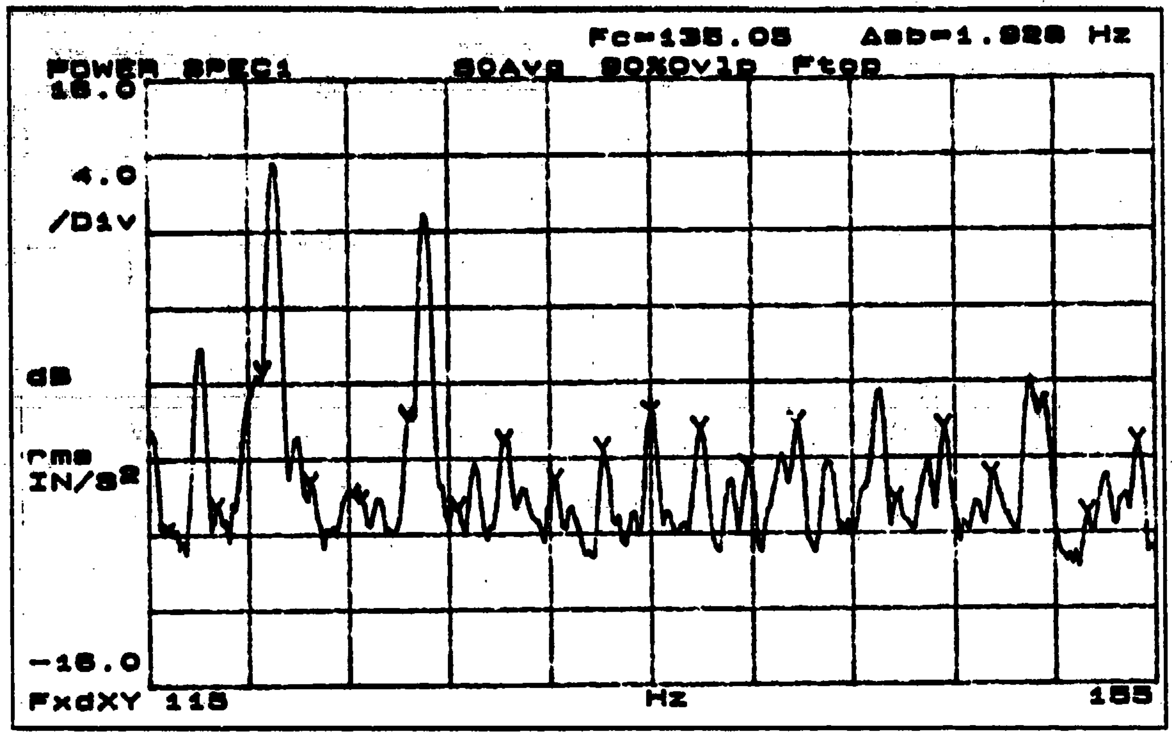

Figure 31. Three gear teeth filed.

distortion of the "teeth", the concept of the display technique is very clear. Variations in the degree of amplitude modulation (level of the 500 Hz source) on other repeated runs gave different depths to the cusps of the curve; but beyond source levels of about 35 mV the cusps turned into small loops and the definition of the "tooth" profiles began to be lost. The correlation between the source signal levels used and the actual amount of amplitude modulation experienced in a real system was not explored due to limitted time remaining to complete this thesis work, but would be the next logical step to investigate, along with the ability to seed a stray signal into the synthesized 500 Hz signal to see how the ideal display would appear with a simulated defect.

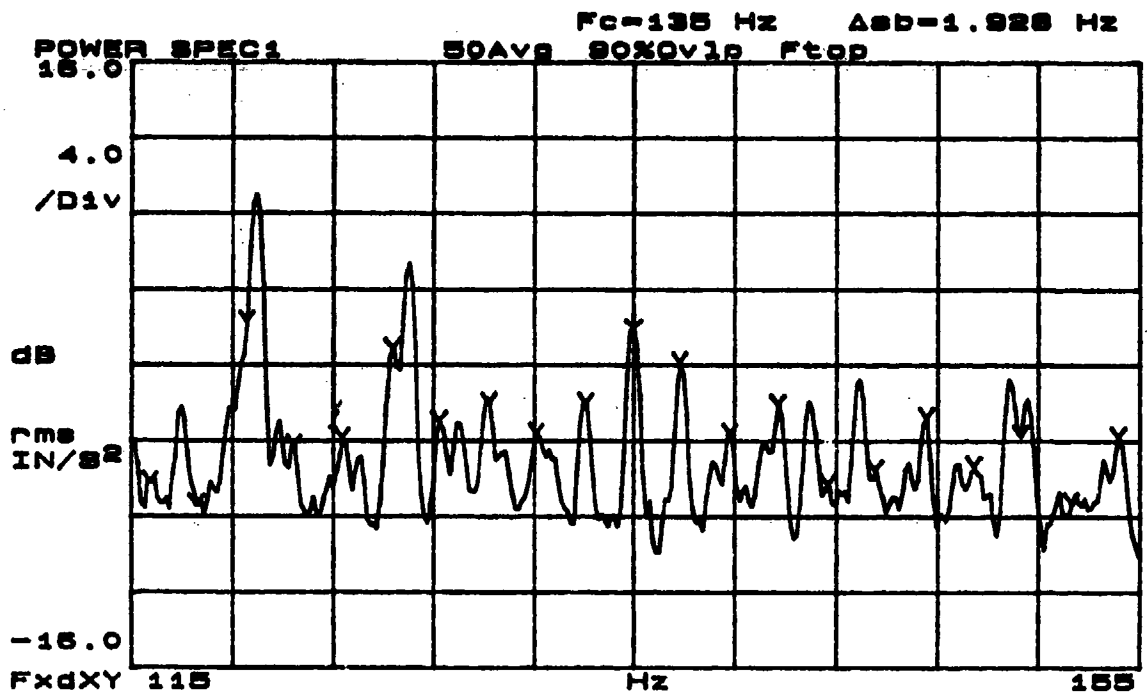

Figure 32. Four gear teeth filed.

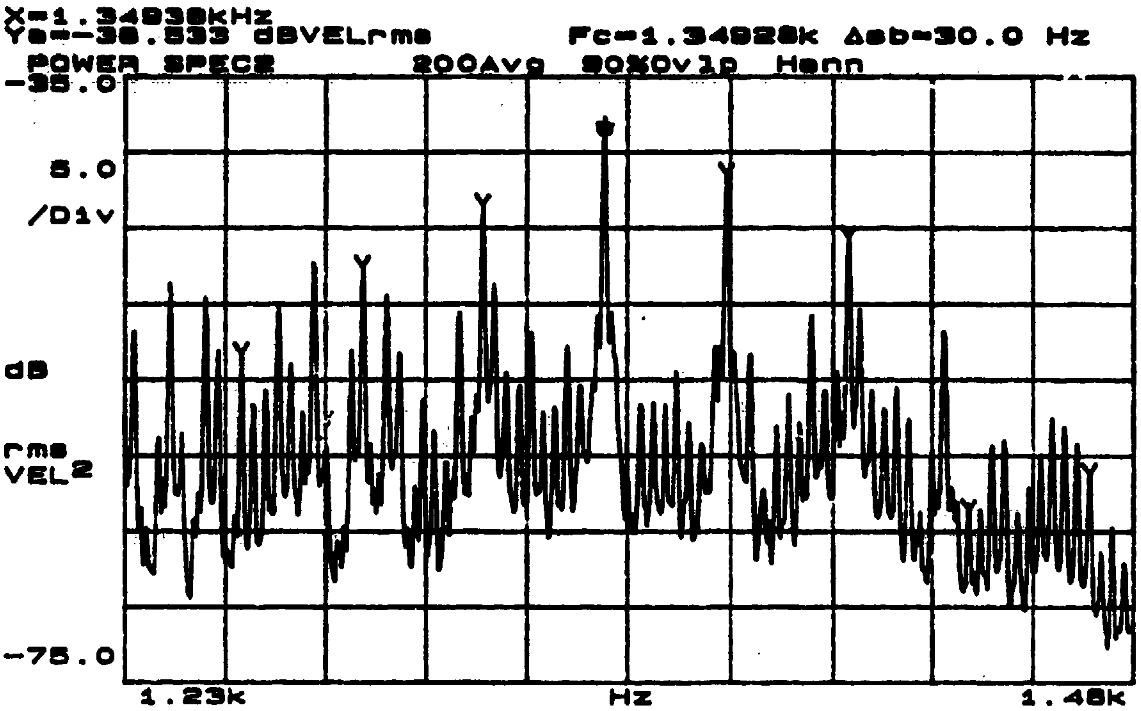

Figure 33. Major sidebanding by 30 Hz signal.

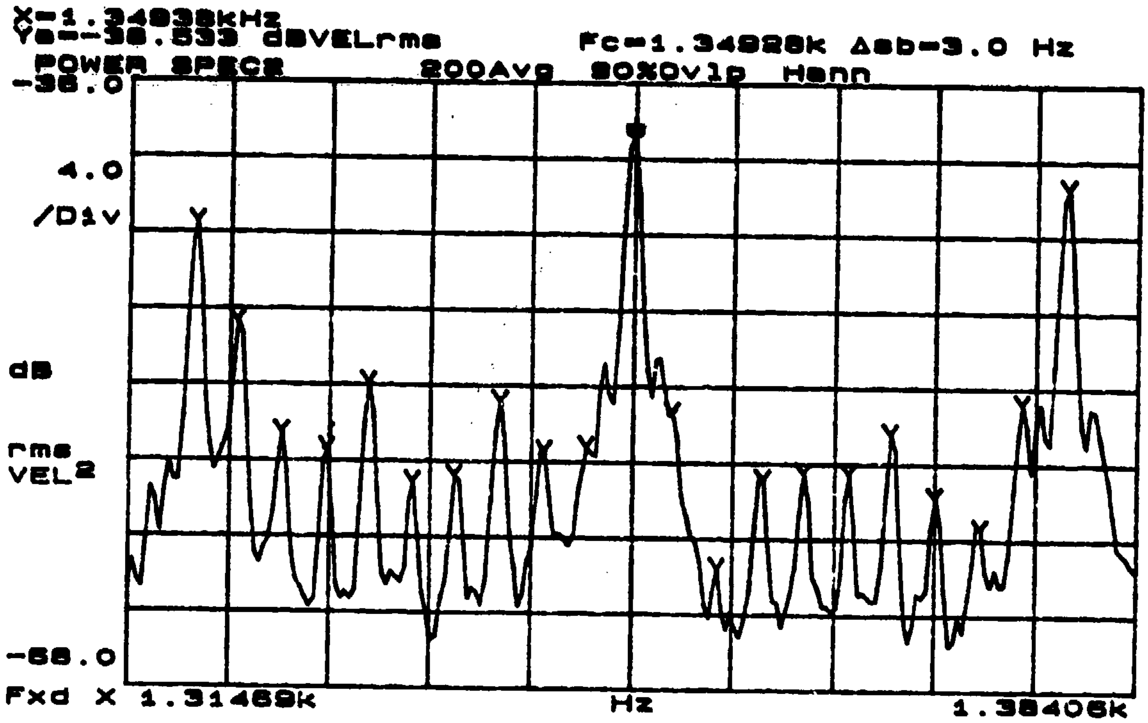

Figure 34. Secondary sidebanding by 3 Hz signal.

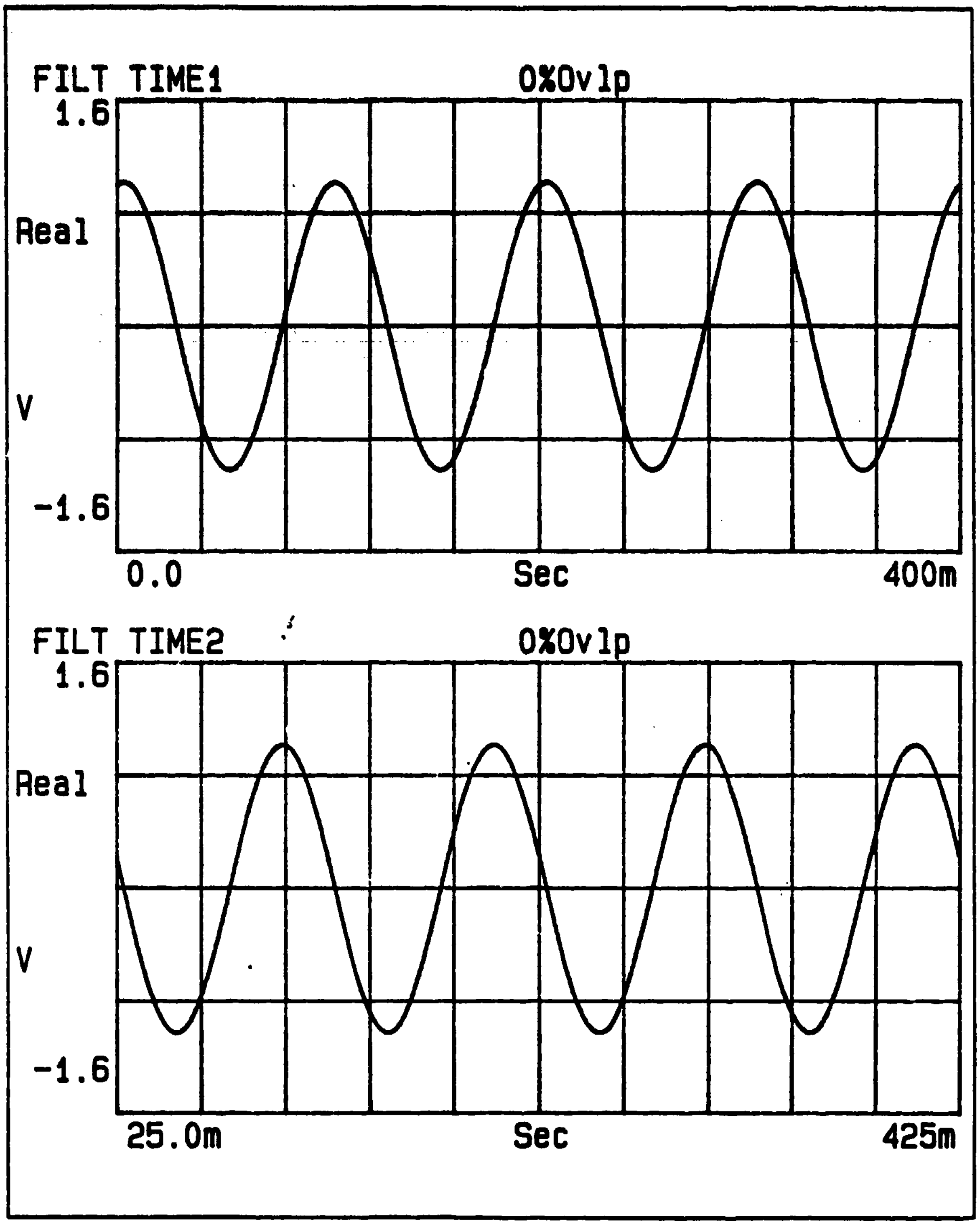

Figure 35. 10 Hz waveforms as measured.

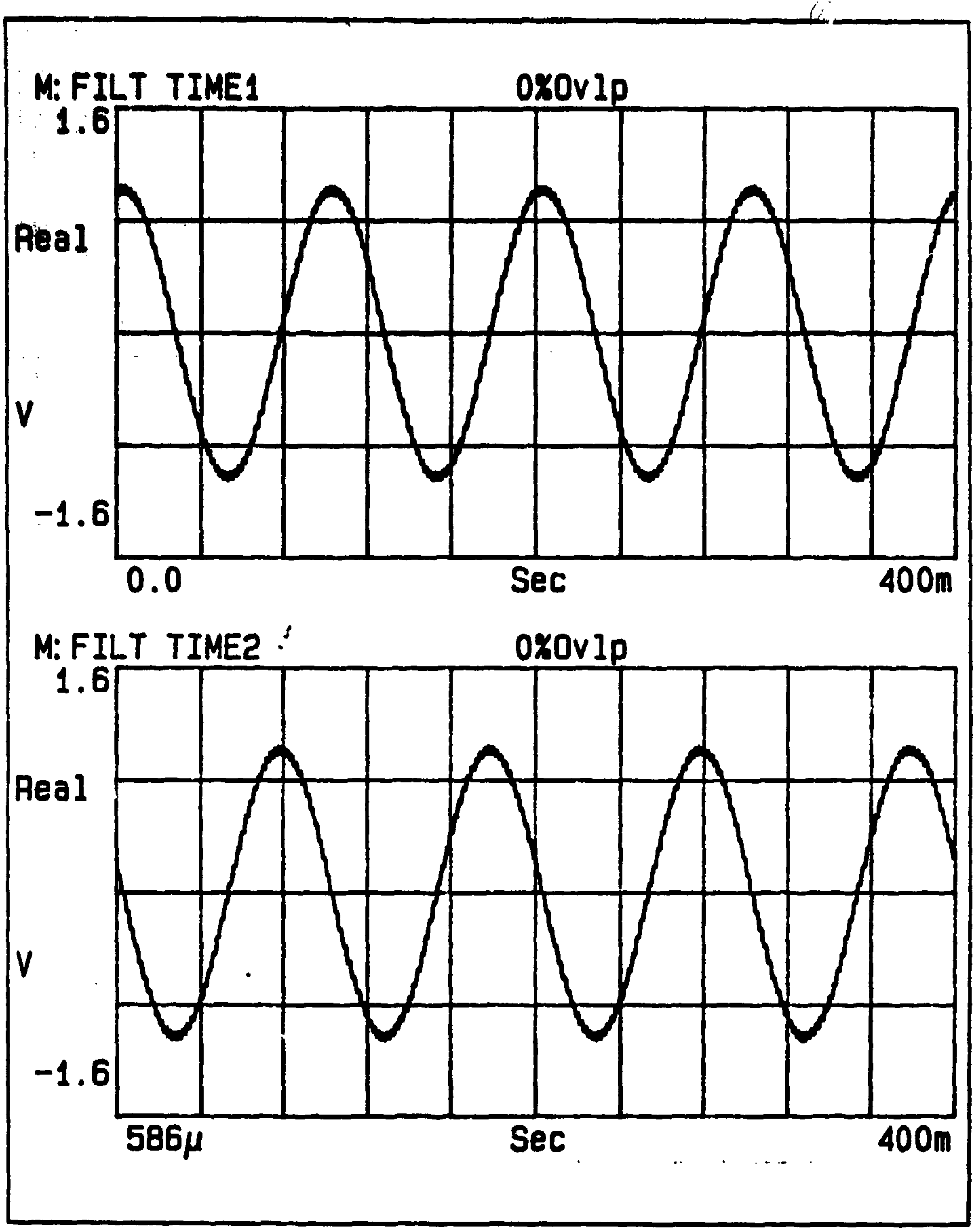

Figure 36. **Sum of 10 Hz and 500 Hz waveforms.**

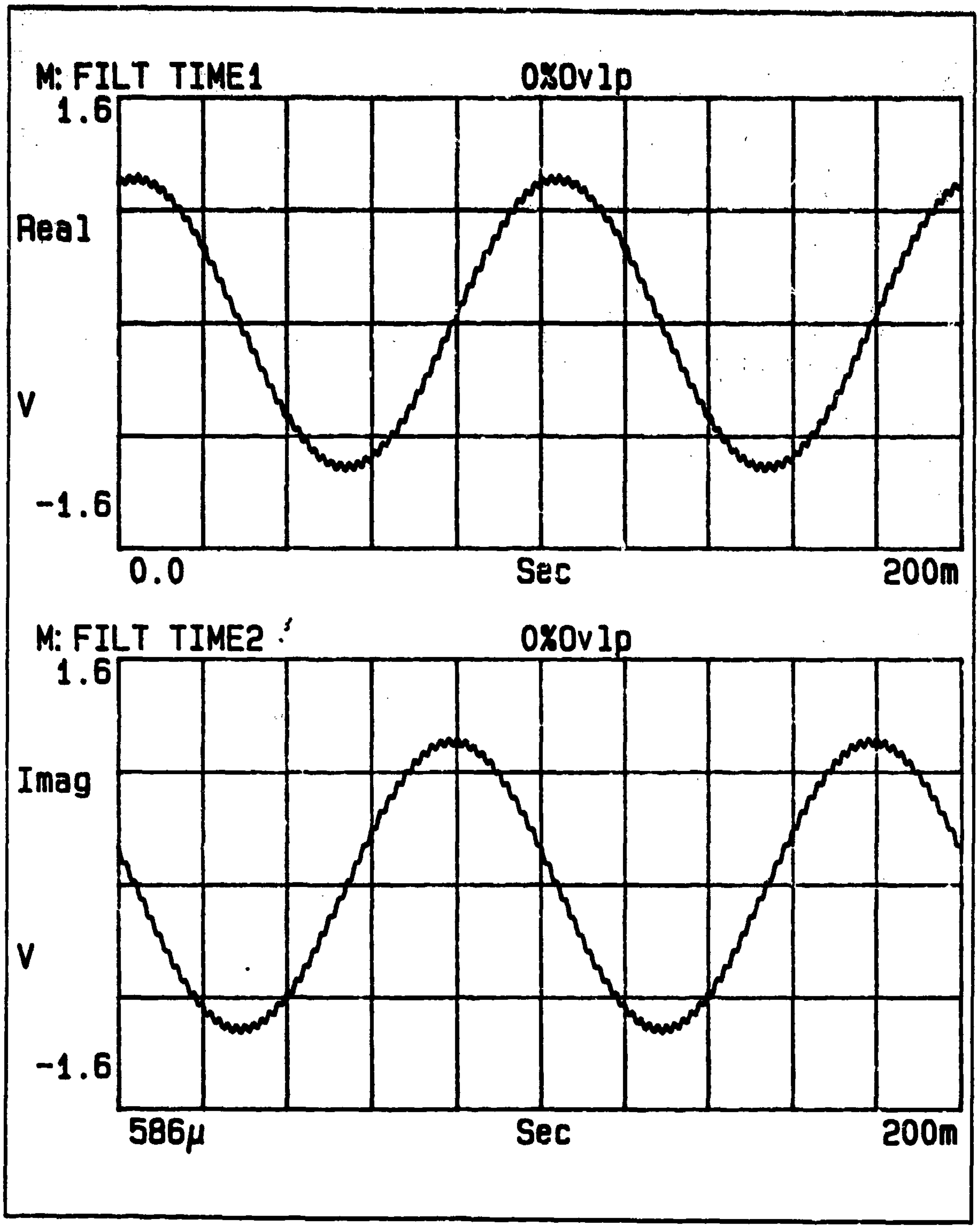

Figure 37. Summed waveforms after conversion into complex form.

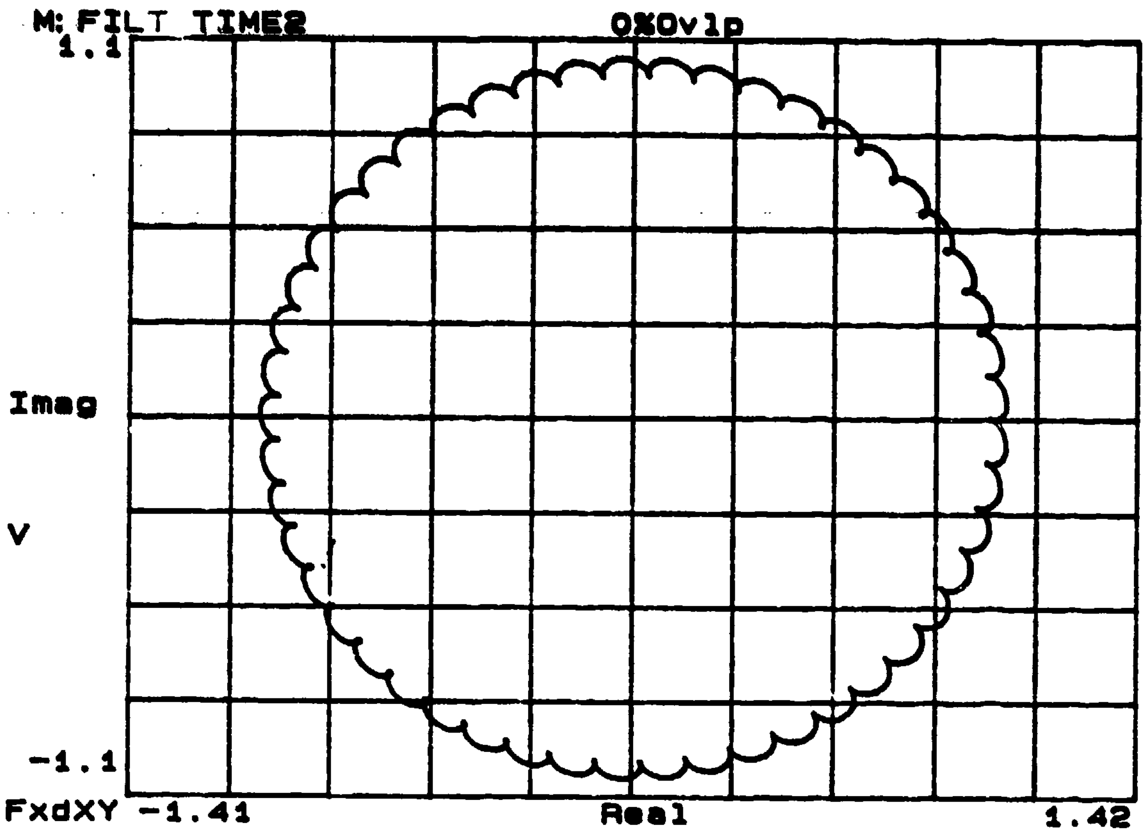

Figure 38. Final Nyquist coordinate plot of complex waveform.

VI. CONCLUSIONS

A physical machinery diagnostics model was developed that was designed to incorporate some of the more common machinery faults found in rotating machinery relating to shaft, bearing, gear, and alignment defects. Based upon the results of the model tests, the model was found to be effective in displaying designed machinery faults with the exception of mechanical looseness. The balance disc performed precisely as designed in that it was found to be very sensitive to mass addition. The results of this study clearly demonstrate that it is not always possible to distinctly identify all specific machinery faults which may be present by obtaining signatures using only a single transducer. The implication is that more complete and reliable machinery diagnostics might only be realized through the use of two channel measurements.

VII. RECOMMENDATIONS

A. MODEL IMPROVEMENTS

In view of the results of the model tests, the following recommendations are made with regard to model improvements.

For the balance disc assembly, a changeover to ball bearings might remove the appearance of the multiple shaft harmonics from the spectra which are thought to be caused by excessive clearance in the sleeve bearings currently being used. This would also remove the requirement to retain the aluminum collars on the shaft which account for a small protion of the residual imbalance, and this would preclude introducing extraneous signals (rubs) in the event that the collars were inadvertantly spaced too close to the bearing blocks. Also, even though the shaft axial position tends to track to the drive belt plane, there may some slight wander or jitter in its tracking which may be the cause for the axial readings that were seen in many displays.

The difficulties in working on the ball bearing assemblies shows the need for obtaining larger sized components for conducting specific damage studies. This would also create more working space on the assembly itself which would facilitate positioning of components and measurement devices; some bearing blocks on this model could not be fitted in any but the vertical direction with the transducers that were obtained (of course the option of smaller transducers exists, but this may involve greater expense). This directly eliminated the ability to take readings in other directions in order to gain phase information which would have been of benefit in many cases.

The difficulty with speed control points out the need for more refinement of this feature. It is possible that removing the motor controller from the base plate might help if the instability is due to controller vibration, but even if this is so, it is not expected that this will suffice since the problem seems to be more related to a deadband in the speed control knob. The use of the external sampling feature of the DSA is recommended as the best alternative method available to avoid errors induced by speed drift.

All bearings should be solidly pressed and/or crimped into place to firmly seat them in the bearing block bores and guarantee best transmissibility of signals. Also the proper method of installation must be observed. The last bearing that was installed was seen to be pressed into place by pushing down on the inner race, a practice which may be the reason why both bearings showed inner race defects and not only the one which was intentionally damaged.

All components to be shaft mounted should be pressed or shrink-fit in place to avoid the introduction of imbalances due to set screws, flats ground on the shaft, etc. Additionally, this is presumed to have had a pronounced affect on the final alignment of the gears, and so it is recommended that any components which similarly involve mating contact or close tolerances be specifically so installed.

A more suitable load should be acquired for the model. The load presently used was insufficiently steady for the long time periods involved with several types of measurements such as when time domain averaging was used where overlap processing was automatically set to zero percent which greatly increased the time needed to take high resolution measurements. A recommended alternative is a variable core magnetic brake type assembly. The assembly is driven by a small constant speed motor and the strength of the magnetic brake is varied by the degree to which the core is axially engaged in the field. The load from such a device should be very steady, easily controlled, and of sufficient capacity to provide what is needed to adequately test the components presently incorporated in the model.

B. FUTURE STUDIES

In view of the uncertainty of the amount of noise contamination in the signals, it is recommended that two channel studies be pursued which would then allow measurements of signal coherence as an excellent indicator of the amount of noise present.

It is recommended that phase readings be made a part of future experiments since this information is, in many cases, vital and necessary for effective diagnostic work. This need for phase information further underscores the importance and benefits of two-channel measurement studies.

In continued work on the gear display technique, it is recommended that efforts be directed toward further studies using ideal signals, especially their correlation to real measured events and parameters. Simulation studies which try to quantify the degree of amplitude modulation and its resulting influence on the gear display, and the result of an "ideal defect" in a simulated signal are two areas which may quickly prove or disprove the usefulness of continued efforts to perfect the technique. The studies done thus far have used a 25 millivolt modulating signal and produced a gear profile simulation of reasonable proportions, but levels of 35 millivolts and higher cause severe "tooth" distortion. If the actual ratio of carrier to modulating wave amplitude is far from this, there could be enough distortion in the display to render the technique useless. This is why continued efforts are recommended to remain in pursuit of more substantial theoretical groundwork to support the potential benefits of this display technique. This, of course, should then be followed by actual model tests if the technique still shows promise. In obtaining real signals, the main effort should be in trying to capture as "clean" a signal as possible since this is critical in the quality of duplication of the profile. Also, the signal sought might best be restricted to the gearmesh event alone which could then be superimposed on a simulated signal which represents the shaft frequency since the lower (shaft) frequency serves merely to create the "base circle" of the gear.

In general, the difficulties experienced in the model testing were primarily in the areas of speed drift, load stability, amount of load, and size of components. Future machinery models, or changes to this one, should consider these factors seriously in the early stages of the design. The load and power supplies must be very carefully sized to ensure that the device under test is able to be loaded to the desired or required levels which will give meaningful results, and the load stability is important because and the use of some of the more beneficial techniques in reducing noise and otherwise improving measurement content (such as time averaging) requires extremely tight control of speed.

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