NBSIR 73-252

MFPG

detection diagnosis and prognosis

Mechanical Failures Prevention Group
MEETING No. 18
NOVEMBER 8-10, 1972

PROCEEDINGS

Edited by
T.R. Shives and W.A. Willard
Institute for Materials Research
National Bureau of Standards
Washington, D.C. 20234

NBSIR 73-252

MFPG

DETECTION, DIAGNOSIS, AND PROGNOSIS

Proceedings of the 18th Meeting of the Mechanical Failures Prevention Group, held at the National Bureau of Standards,

November 8–10, 1972
Gaithersburg, Maryland,

Edited by

T. R. Shives and W. A. Willard

September, 1973

The 18th meeting of MFPG and these proceedings were sponsored by the Office of Naval Research, the National Aeronautics and Space Administration, and the Institute for Materials Research of the National Bureau of Standards.

U. S. DEPARTMENT OF COMMERCE, Frederick B. Dent, Secretary

NATIONAL BUREAU OF STANDARDS, Richard W. Roberts, Director

FOREWORD

The 18th meeting of the Mechanical Failures Prevention Group was held at the National Bureau of Standards in Gaithersburg, Maryland, on November 8-10, 1972. The program for this meeting was organized by the MFPG Technical Committee, Detection, Diagnosis, and Prognosis which is chaired by Raymond Misialek. There were four technical sessions, each of which dealt with an aspect of the topic. The program is a tribute to the committee, to the session chairmen, and, of course, to the speakers.

This volume consists of the papers presented at the 18th meeting as well as the discussions of these papers and the panel discussions. The authors' manuscripts are given as submitted to us except for minor editing in a few cases. The discussions were recorded at the meeting and were edited somewhat to improve readability. We are presenting these discussions as faithfully as practical and we sincerely trust that no one has been misquoted.

Special appreciation should be extended to Mr. T. R. Shives and Mr. W. A. Willard for their editing, organization and preparation of the "Proceedings"; Mrs. Sara Torrence of the Special Activities Office of NBS for the meeting and hotel arrangements; Mr. H. C. Burnett, NBS Metallurgy Division for general coordination, and the entire staff of the Metallurgy Division and the Institute for Materials Research for their assistance in many ways. Special thanks are due Miss Bronny Evans of the NBS Metallurgy Division for her diligent efforts in transcribing the recorded discussions.

ELIO PASSAGLIA
Executive Secretary, MFPG

Chief, Metallurgy Division
National Bureau of Standards

TABLE OF CONTENTS

SESSION I

BEARING

CONDITION

MONITORING

Chairman: Paul L. Howard
SKF Industries

"A STATUS REPORT ON SENSORS

AND

THEIR APPLICATION TO BEARING CONDITION MONITORING"

By

R. M. Whittier, Manager, Transducer Development

Endevco, Pasadena, California

INTRODUCTION

Vibration and noise emanating from a rolling contact bearing carries information about the condition of the bearing. The total noise environment is, however, extremely complex. Through analysis and experimentation certain features of the total vibration and acoustic environment can be found to vary with the bearing's operating condition. Experimenters and instrument designers have found that detection of such signals can be practical, and in application the result can be useful and quite repeatable. This presentation discusses the criteria for sensors and the technology employed to achieve this type of bearing condition monitoring.

VIBRATION AND ACOUSTIC ENVIRONMENT

In addition to the total vibration and acoustic environment which envelops the bearing, the physical characteristics of the bearing itself cause additional periodic and nonperiodic dynamic stresses. Periodic inputs include both harmonic forced vibration and repetitive force impulses. An example of a forced vibration input would be the oscillatory motion caused by dimensional imperfections, such as race waviness. Repetitive impulses occur from material impacts at surface discontinuities, for example, at a spall. Both of these types of inputs occur at rates which are higher than the rotating frequency of the bearing. They are a function of ball quantity and bearing geometry multiplied by the rotating frequency. A second type of physical change is the ball, race, and structure free vibration which is excited by the impulses. A third is the high frequency stress waves which are emitted from material failure points. This is called acoustic emission; it occurs even well before damage is apparent by visual inspection. In short, these various dynamic changes, which might be sensed, do occur at frequencies higher than the fundamental rotating frequency of the bearing.

The transmission of these inputs through the bearing structure and into the surrounding mechanical system significantly changes the input. Stress waves are partially transmitted, partially reflected and self resonances are set up — all this is a function of the total machine design. This means that placement of a sensor is a critical factor to the detection system design. The sensor should be mechanically well coupled to the bearing. This usually means that placing it physically close to the bearing is important.

SENSOR ALTERNATIVES

Many types of motion sensors have been used for machinery vibration measurements. A common application would be the measurement of machine unbalance at frequencies typically within the range of 1 Hz to 300 Hz. Noncontacting and contacting sensors are both in use for this. A non-contacting proximity/displacement probe operates best at low frequencies where the motions do involve displacements in the region of .01" to .1". Contact sensors, those physically attached to the machine, include all types of seismic devices for sensing displacement, velocity, acceleration, or stress. Basic to such a contacting sensor is the fact that its presence does affect the dynamics of the structure. In short, the structure is changed by the addition of the sensor. This can be expressed as an impedance loading problem at the contact point. On most bearings such considerations become important at frequencies above a few thousand hertz. When sensing dynamic motion, one usually keeps the sensor mass small in proportion to the local mass of the structure at the point of contact. More often than not, this desire results in a sensor of 50 grams or less. Because of this limitation, and because of the low displacements at high frequency, the velocity coil vibration sensor is not too useful for detection of bearing faults.

Vibration sensors designed to respond to acceleration inputs are of smaller size than the velocity coil devices. In general their use to 10-20 kHz has become almost routine. Some people even use them to 100 kHz. From 20 kHz to above 100 kHz contact sensors designed to sense the acoustic pressure in the structure have also been used some. Because of the high frequency response capabilities of these types of sensors they are desirable sensors for listening to bearings. A simplified vibration nomograph showing the applicable relationships, vibration levels, and types of sensors is shown by Figure 1.

PIEZOELECTRIC SENSORS

Piezoelectric transducers are desirable condition monitoring sensors since their sensitivities are high, sizes can be small, and their outputs are self-generating (no excitation voltage required). In the last few years new piezoelectric materials have been developed increasing the temperature capability from about 500° F to greater than 1200° F. This is very important since bearings and machines can often be quite hot. A plot showing temperature capabilities of some of these materials is shown by Figure 2. Long life and excellent reliability, much higher than that feasible with most sensors, can be achieved because these piezoelectric devices have few parts and essentially no internal deflections (not subject to fatigue problems of many other sensors). Also important to the overall reliability is the cabling and connection system design. These components, along with the sensor, must function in extreme environments. In service temperatures in excess of about 500° F result in special cabling and connector designs to eliminate organic insulation, to reduce surface oxidation on contacts, and to improve strength properties of the metal parts. Special designs have been produced to provide connection systems for at least 1000° F; however, for minimum cost and best reliability the electrical connector temperatures should be maintained below 500° F. To provide cabling at temperatures in excess of 500° F the metal sheathed, mineral oxide insulated cables are used. These have been produced in various electrical configurations and sizes from .01" diameter to .18" diameter. One interesting method to place the sensor close to the bearing and to provide fast installation has been to place the sensor at the end of a probe and bring the electrical leads through the probe to a connector at the other end. This connector end is exterior to the machine and the unit is simply inserted into place.

To condition the electrical output from these piezoelectric devices, circuits which sense the electrical charge, not voltage, are usually employed. By doing this the output is not attenuated by the cable shunt capacitance. In addition, the circuits for these types of systems do not require the high resistance-capacitance time constant that the voltage amplifiers require. Typical input resistance requirements are now 100,000 ohms. Both single ended and differential sensing circuits are available, the choice depending on noise rejection needs, shielding difficulties, and cabling/connector requirements. Sensor signals are now being transmitted through standard multi-pin MIL-type connectors into differential amplifiers. If the temperature at the sensor is below $+250^{\circ}$ F, conditioning electronics are sometimes placed within the sensor case. This can reduce cabling and connector complexity.

Many different styles and configurations of sensors are available. Some are designed for placement on bearing housings within machines and others for mounting external to machines.

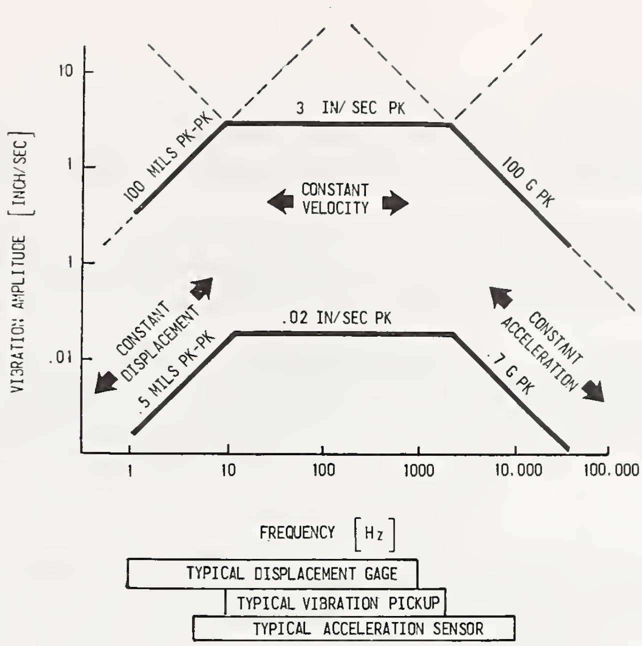


Figure 1. Simplified Vibration Nomograph Displaying Typical Machine Vibration Levels and Sensor Frequency Ranges.

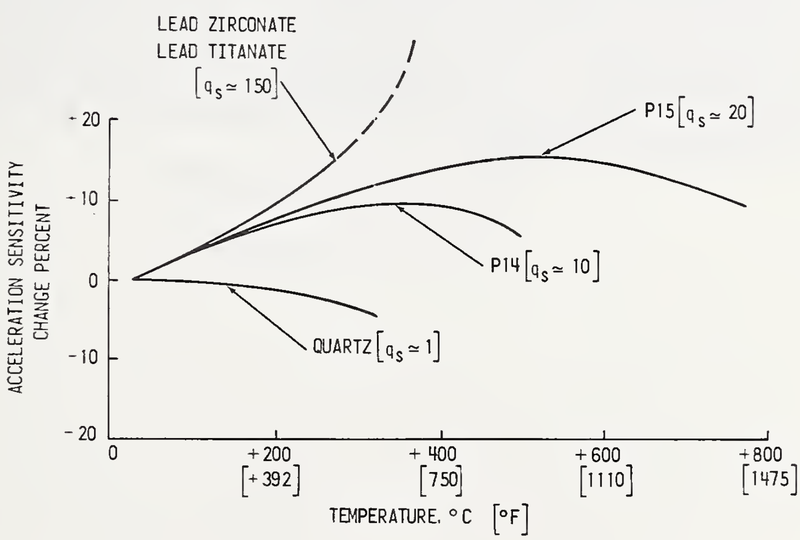


Figure 2. Temperature Ranges and Charge Sensitivity Changes for Several High Temperature Piezoelectric Materials.

DISCUSSION

C. Jackson, Monsanto: Are the P14/P15 materials for high temperature use in accelerometers of the lead zirconate or lead titanate construction?

R. Whittier: No, they are not. The information is proprietary. The materials are manufactured by our company and we don't reveal what the formulations are. They are ferro-electric ceramic materials.

John Sudey, Goddard Space Flight Center: Can you suggest a way to solve the problem when the phenomena is transient and could possibly excite the resonance of the accelerometer? How does one prevent reading accelerometer resonance instead of the failure of the bearing which is the thing he is after?

R. Whittier: A transient input may cause the transducer to resonate when the sensor is quite undamped, as most of them are. Some people actually use this in their system concept, and other people, by looking at it in a different way, simply don't look at that portion if they know what it is.

John Sudey: I would assume that the signal would be so high in magnitude that it would saturate the amplifier that follows the transducer, and, as a result, you would get a very big erroneous signal that does not say anything about the bearing.

R. Whittier: No, quite often that isn't the case. In that spectrum of energy, the frequency content is not white. At the high frequency, it is down, and therefore really the amount of noise, if you want to call it that, sometimes appears to be very small. Another technique that is used is input filtering. In the charge convertor, if you have any resonance, either structural or from the sensor itself, you can eliminate it by electronic filtering.

CREST FACTOR ANALYSIS

FOR

ROLLING ELEMENT BEARINGS

K.A. SMITH

GENERAL ELECTRIC CO.

SCHENECTADY, N.Y.

All mechanical systems generate secondary effects during their operation. These secondary effects, in contrast to primary effects such as pressure and flow in a pump, are not directly utilized in the operation of the system. Examples of typical secondary effects are vibration, acoustic noise, heat generation, etc. In fact, often these secondary effects are major operational problems, and one would rather see a system operate without them.

Luckily, there is a positive side to these effects. Generated inside the machine, they carry information about internal operation to the outside where they can be detected by a sensor and analyzed to provide insight into the operation of the system. Signature analysis of rotating machinery is one use of these secondary effects which has demonstrated great success in recent years.

DIAGNOSTICS OF ROLLING ELEMENT BEARINGS

Ball and roller bearings usually follow a characteristic wear/failure pattern. Since the sliding between metal surfaces is minimized, the wear rate is usually very low. But the rolling contact load between balls/rollers and the race is instead very high, and leads to subsurface fatigue. Therefore, rolling element bearings have a tendency to fail by fatigue rather than wear-out.

Obviously, the statements above are very general, and can be challenged in special cases. But they describe the typical situation for bearings which are properly designed and maintained. All bearings can fail through many other mechanisms if they are not properly used, for example, through corrosion or lack of lubrication.

Figure 1 shows a simplified but typical wear rate curve for a rolling element bearing. Early in life the wear rate is often at a maximum. This is the running-in period. The maximum may be absent for very high precision bearings. After this initial period comes a long period with very low-wear rate. This represents the major portion of the bearing life. During this period the wear rate is low, but both races and balls accumulate fatigue cycles.

The end of the low-wear period is marked by the appearance of some surface defects, usually on one of the races. This may be fatigue spalling but also corrosion spots or other effects. The surface defects are small at the beginning, but disturb the smooth rolling process, increase the friction, and therefore grow and increase the wear rate. When a sufficient amount of surface defects have accumulated, the bearing fails to perform its function. It may jam or fail to hold the position of the shaft, so that secondary damage occurs. It may overheat and burn the lubricant. This is the end of the bearing life. Many techniques are used to detect bearing malfunctions in operating machinery.

Temperature: The increased friction which preceeds bearing failure generates heat that increases the temperature of bearing and lubricant. Oil temperature is often measured.

Overall Noise Level: When the spalling has proceeded sufficiently the bearing gets noisy. The problem is that bearings are usually buried deep down in machinery, and there may be many other noise sources.

Oil Contamination: Spalling of bearing surfaces generates metal chips which can be collected or detected with spectroscopic oil analysis methods.

All these techniques and others have been tried with varying degree of success in many cases. The problem is generally that the malfunction indication comes too late, so that there is no time to take preventive action before total failure has occurred. Another major problem is that these techniques do not identify the failure mode. This is unsatisfactory since failure mode identification is often necessary to decide what corrective action should be taken.

In the following a few examples will be given to demonstrate how bearing malfunction signatures can be predicted and extracted from complex vibration signals.

DETECTION OF FATIGUE SPALLS AND OTHER LOCAL DEFECTS

A local defect in the inner or outer race of a bearing generates an impact every time a ball (or roller) rolls over it.

Likewise, if the defect is in a ball surface, an impact is generated every time it hits the inner or outer races. If the bearing rotates with constant speed, each defect generates a regular set of impacts with an impact repetition rate that depends upon bearing speed and location of the defect. Assuming that no sliding occurs in the bearing, those fundamental repetition rates can be calculated. The result is:

Outer Race Malfunction $$f_e = \frac{n}{2} f_r \left(1 - \frac{BD}{PD} \cos \beta\right) \quad \text{(cps)}$$

Inner Race Malfunction $$f_i = \frac{n}{2} f_r \left(1+ \frac{BD}{PD} \cos \beta\right) \quad \text{(cps)}$$

Ball Malfunction $$f_b = \frac{PD}{BD} f_r \left(1 - \frac{BD}{PD}\right)^2 \cos \beta^2 \quad \text{(cps)}$$

where f_r is the relative speed between inner and outer race (revolutions per second), and BD, PD, β, and n as defined in Figure 2.

The vibration pattern, or signature, which can be expected from a single local defect in a bearing then consists of a series of transients with a repetition rate as calculated above. Figure 3 shows such an ideal pattern.

The problems in practical cases are that the signature from the bearing defect is often hidden behind background noise and also that the repetition rate can only be calculated approximately. It has been discussed in an earlier paper(1) how a diagnostic system may be designed to extract this signature from a local defect, and thereby make the diagnostic system more selective. In addition to the mere transient repetition rate, there are many characteristic features associated with each individual transient. These details of the transient waveform depend on the load pattern for the bearings and on the structural response to the impacts between rolling element and defect.

Once a relevent signature has been extracted it is possible for an engineer to analyze the result and evaluate the condition suggested by the data. Often, however, it is desireable to have on-line continuous monitoring with an automatic warning if a failure is approaching. This requires a pattern recognition scheme to detect the failure parameter. For detection of fatigue spalls in rolling element bearings, the crest factor meter has been found to be a sensitive indicator. By measuring the peak level of the processed data and comparing that to the RMS level, an "impact index" can be defined which increases with increasing spall size. This measurement allows monitoring of spall growth rates as well as providing an alarm when the defect indicator exceeds a preset level.

This technique has been the basis of several pieces of hardware developed by General Electric and now in use by both government and industrial organizations. In fact, the sensitivity of this technique is such, that in cases of low noise and strong bearing signals (such as where the sensor is mounted immediately on the bearing housing), it is possible to detect small spalls without prior processing or signal extraction.

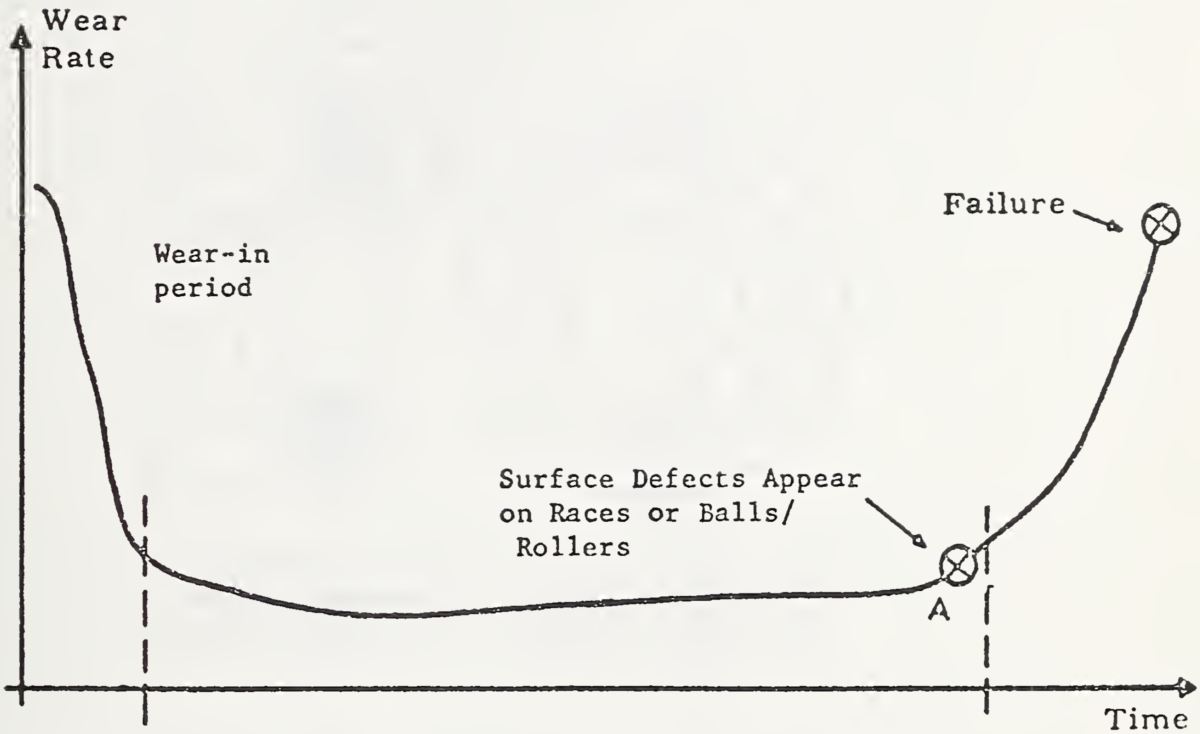
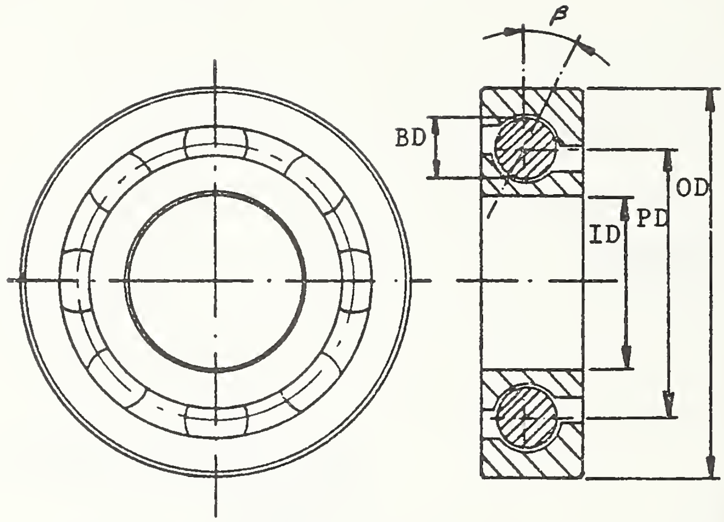


Figure 1 · Typical Wear Rate Function For Rolling Element Bearings



OD = Outer Diameter BD = Ball Diameter
ID = Inner Diameter β = Contact Angle
PD = Pitch Diameter n = Number of Balls

Figure 2 Basic Dimensions of Rolling Element Bearing

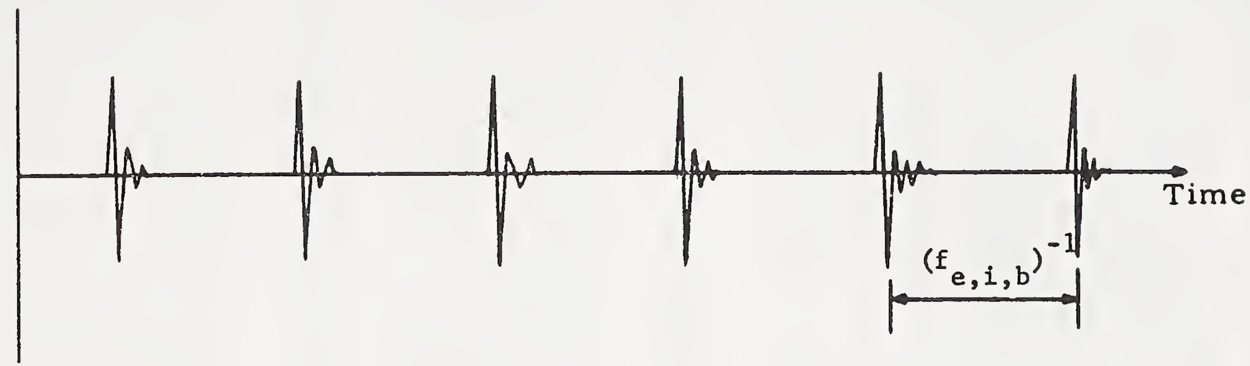


Figure 3 Ideal Vibration Signature From Local Bearing Defect

BIBLIOGRAPHY

1. Weichbrodt, B., "Mechanical Signature Analysis, a New Tool for Product Assurance and Early Fault Detection", presented at Fifth Relaibility and Maintainability Conference, New York City, July 18-20, 1966, sponsored by AIAA, SAE and ASME. Published in the Annals of Reliability and Maintainability vol. 5, Achieving System Effectiveness, American Institute of Aeronautics and Astronautics (1966).

DISCUSSION

Oscar Horger, Brenco: What degree of spall can be picked up in roller bearings by this method?

K. Smith: In some of our tests on roller bearings, we have inserted artificial defects of various sizes into the bearings. We have found in cases where the spalled area covered something on the order of 1/4 of the roller, there was no significant energy apparent in the analyzed data. Where the spall did cover a significant portion of the rolling element, the information was present just as in the ball bearing. Now the thought occurs to me that probably in a spall which develops as a result of operation, you are not going to have the artificial situation that we created where there is a very tiny spall on one end of the bearing, of the race, of the width of the race, unless there for some reason is high force or high loading at that point, and indeed there is contact and impact generated. So I think in applications where the defect is generated during the normal operation of the bearing, this same technique applies equally well to roller bearings as to ball bearings.

William Glew, Naval Eng. Test Establishment: Is the averaging technique on real time analyzer systems equivalent to signal summations?

K. Smith: The analysis that I'm talking about is strictly a time domain averaging process. However, you can draw very close correlations between averaging in the time domain and using a comb filter with very narrow band width which has response peaks at each of the harmonics of the periodicity that you are using for the summation process. In that sense, the techniques are complimentary, are similar. It's just that the averaging that I'm talking about is performed entirely in a time domain.

J. Sudey: If in fact you did get some answers by looking at the envelope, would you pursue envelope protection further? It seems to me it would be so much simpler than the crest factor.

K. Smith: The crest factor measurement by itself is a very simple technique. I think it's much simpler than doing the envelope detection, particularly where the envelope detection may require something like the summation processing in advance. It is very often possible to simply filter out the resonant response of the accelerometer. If you have a high enough resonance, you can get sufficient impact energy in a band width that extends no farther than 20 kilohertz, and you throw away the rest of the information above 20 kilohertz. One gets rid of the resonance of the transducer by that means.

Jerry Forest, Ontario Hydro: Your block diagram on crest factor instrument arrangements showed an integrator at the tail end. What is the purpose of this integrator?

K. Smith: The integrator is not absolutely necessary in the overall scheme of the crest factor monitor itself. In that particular case, the integrator was used to make sure the crest factor was remaining at a high level for a sufficiently long period of time to give you a reliable reading for that application, but in fact it is not necessary in most cases, and on the system that we have used on jet engines and on traction motor bearings, the averaging or the integrating done simply by the meter response itself is adequate to give you a stable reading.

RESONANT STRUCTURE TECHNIQUES FOR BEARING FAULT ANALYSIS

RICHARD F. BURCHILL, PROJECT ENGINEER
FIELD VIBRATION ANALYSIS, MECHANICAL TECHNOLOGY, INC.

This paper presents a rolling element bearing detection technique developed to define initial signs of failure in an angular contact ball bearing for a space gyro application. This was done by MTI under contract to Marshall Space Flight Center at Huntsville, Alabama. Goals were for a detector which could evaluate installed bearings with a minimum of installation difficulties, had minimum speed, load, and temperature dependance, and produced an output which varied with the state of bearing condition so end-of-life predictions could be made.

The development program was a rather straight forward experimentally based one which used a variety of sensing methods to define performance of new and damaged bearings which were operated in a simple test rig. The subject bearing was mounted on a stub shaft which was supported by a high speed precision spindle. A housing was pressed on to the test bearing outer race and floated on it. Thrust load was applied, from 50 to 1000 lbs., and the bearing was operated at 6000 and 8000 RPM.

New bearings provided base line performance. To simulate the onset of fatigue, a bearing was damaged by acid - etching a line across the inner race ball track, 1 mil deep and about 12 mils wide. This fault is similar in character to the first fatigue spall which appears in a worn bearing and guarantees that a defect indication will exist for any load situation.

The bearing fault indication frequencies for this particular bearing were computed from rolling contact geometry. For this 107 size bearing, they were 8.7 x inner race rotation for an inner race fault; 6.3 x inner race rotation for an outer race fault; and 6.0 times inner race frequency for a ball fault.

We looked at a variety of sensing systems to detect the bearing fault:

1. Sound - to 20 K Hz with a microphone near the test bearing
2. Narrow band frequency analysis of housing accelerometer outputs at fault frequencies (to 2000 Hz) using real time spectrum analysis techniques
3. Bearing torque
4. Bearing outer race strain measurements using strain gages
5. Ultrasonic translator output in the 36 K Hz to 44 K Hz range which several experimentors have used as a fault detector
6. Broad band accelerometer outputs to 50 K Hz using a miniature accelerometer mounted on the bearing housing

From this variety of sensing techniques, only the miniature accelerometer output provided the kind of output variation we felt we could work with. Narrow band frequency analysis of the faulted bearing at bearing ball defect passing frequency produced only very minor indications which were nearly lost in noise, yet one could easily see the 8.7 defect ball contacts for each revolution of the inner race. The first figure is of oscilloscope trace photos taken for three different bearings.

The upper photo is data from a new class 9 bearing.

The center photo is a similar bearing which has operated several thousand hours and is beginning to show slight wear indications which show up as a general roughness increase.

The lower photo is another new bearing which has an acid etched fault across the inner race. Note the distinctive impact and decay indications as each ball contacts the etched line. Pulses occur at 8.7 x inner race rotation frequency.

It was found that the character of the damaged bearing signal was retained when the data was passed through a band pass filter set of 28 K Hz center frequency. This filter has fairly narrow band characteristics with a "Q" of about 13 which produces a window 2400 Hz wide at the 3dB points.

Second figure -

The next set of photos are for those same three bearing accelerometers, but after being passed thru the 28 K Hz filter. Note that the lower photo more clearly shows the characteristic impact and decay phenomenon.

We concluded that the fault information we were after was being carried on the 28 K Hz response much like an AM radio signal, so a suitable demodulation technique was developed to get at that information.

The results of this demodulation are shown on the 3rd figure. Again, the new bearing on top shows very little discrete content, while the worn bearing and the discrete fault bearing show individual ball contacts. The scale of the bottom photo has been reduced to permit the data to be photographed, but at this point the fault indications are about 30 times greater in amplitude than those produced by the new bearing.

A broad range of conditions were evaluated to determine the effect of this bearing response, and we come to the following conclusions:

1st - the 28 K Hz response was characteristic of this bearing and did not change frequency with speed or load.

2nd - the amplitude of response at 28 K Hz was only weakly load dependent. A 20 times increase in thrust load produced only a 3 times increase in amplitude.

3rd - the amplitude of response at 28 K Hz varied almost directly with speed - a 25% decrease in speed produced a 20% decrease in amplitude.

4th - the bearing fault indications were only mildly attenuated when the bearing outer race was moved from a tight, tap-fit housing bore to one with 1 mil clearance. This turns out to be a real asset.

We took advantage of these characteristics to build a Fault Detection circut which included the 28 K Hz filter and envelope detection device. The output of the envelope detector was presented to a tuneable narrow band pass filter which could be set to inner race fault frequency, outer race fault frequency, or ball fault frequency to detect the type of fault present. An alarm system indicated when a preset level was exceeded.

We had built the required fault detector without fully understanding the mechanism of the indicator. My initial computation of resonance of an individual ball had come out at 28 K Hz, but I had neglected "G" in the equation and the actual value was 550 K Hz, much beyond the frequency response range of this equipment. A calculation of the response of the ball on its oil films as a spring-mass system gave values from 28 K Hz to 34 K Hz, but it was dependent upon ball load, and the experimental data did not follow that trend. Scratch another good possibility.

I had, very early in the program, computed resonant frequencies of the inner and outer races as free rings, but the first modes came out to be 6000 Hz for the inner race and 2800 Hz for the outer race, and I had assumed that the first modes would be the predominant one for each. Neither frequency was present in the Accelerometer Spectrum Analysis. I later found that indeed we were measuring race resonance but higher modes than I had expected. With a special fixture to resonate the inner and outer races separately, it was found that both responded strongly at 27 to 28 K Hz, apparently the third ring mode for the inner race and the 5th ring mode for the outer race. Exactly why these modes are excited so clearly we don't understand, but it may be that both being in resonance reinforces the output.

It was found during race resonance testing that low order modes of vibration were easily excited in the free ring condition, but these seem to be very sensitive to race mounting conditions and are not transmitted through normal bearing mount interfaces. It seems that the very small deflections involved in the acceleration output in the 28 K Hz region are not damped out by the energy dissipators one normally finds in a composite structure.

We have taken this test procedure and applied it to a number of different bearing configurations to define bearing condition, and have always been able to find a discrete fault frequency in the range from 20 K Hz to 45 K Hz which carries the ball passing information we need. Outer race faults in line and out of line with the sensor were evaluated without difficulty, and also individual ball faults were easily defined. Ball faults may be intermittent however, because they may not be in the ball rotation plane and may not show up under all operating circumstances.

On several occasions we have broad band filtered the demodulated output so that inner race, outer race, or ball fault will trigger the single alarm. Often it is not important which component has failed, only that one of them is defective. General wear in the bearing shows up as a random increase in response at the resonant frequency and an increase in DC level from the demodulator. We have proposed using logic to pass or fail units based on combinations of conditions of discrete faults and general roughness or wear. We have seen frequency components in the demodulated output which appear to be related to bearing mounting conditions, once and twice rotation, so it appears likely that the same techniques might spot problems due to housing machining errors.

We have looked at a number of automotive grade transmission and accessory bearings with good results. The class one bearings produce significant amplitudes at resonance from their apparently higher surface roughness, but the first fatigue fault still shows up an order of magnitude greater in amplitude than acceptable bearings, so a very reasonable range exists to permit limits to be set.

We worried that the etched fault we had been using to simulate a fatigue spall was too great in amplitude until an automotive manufacturer provided us with a sample bearing which had been removed because it was "noisy". Its level checked out about 8 times greater than our bearing, so we felt much better about it!

In summary, then, bearing fault information is available from a system which uses high order race resonance as a carrier for bearing defect information. With high frequency sensors and electronics, it produces a linear response output which is related to the magnitude of the defect within the bearing, so the rate of deterioration of the bearing can be monitored. The sensor can be attached to an outside surface near the bearing to diagnose the condition of operating systems. The frequency range examined is far removed from the areas in which general machinery vibrations are present, so sensitivity is maximized. General wear indications are present to permit study of system response to oil contamination by dirt, or by general bearing wear.

The system can be applied to new bearings as a quality control check. Expensive used bearings can be evaluated to determine reusability. It can be used to evaluate ball or race damage incurred during assembly or during non-operational shock condition. Finally, and most important, it can define the performance of a bearing in a complex machine and give warning of impending problems to allow replacement much in advance of catastrophic failure.

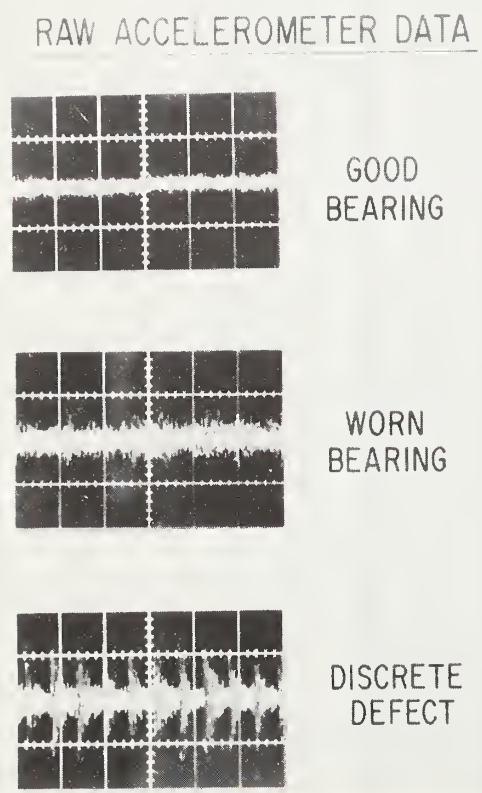


Figure 1.

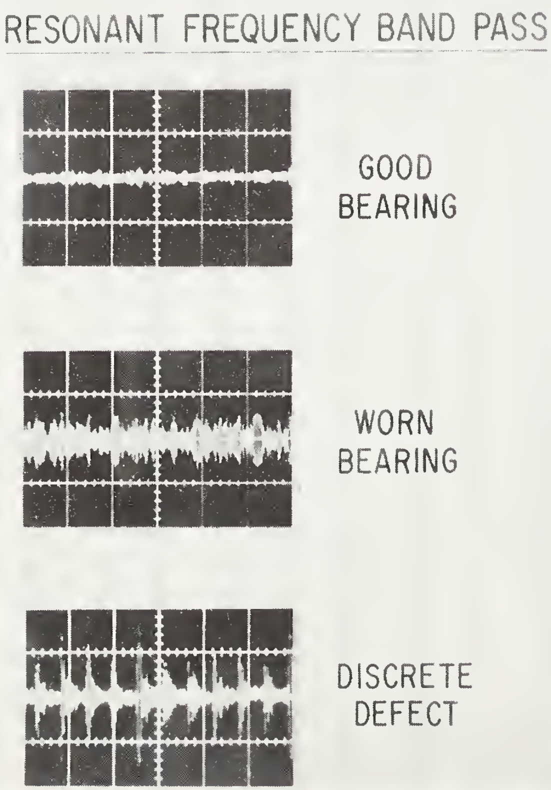


Figure 2.

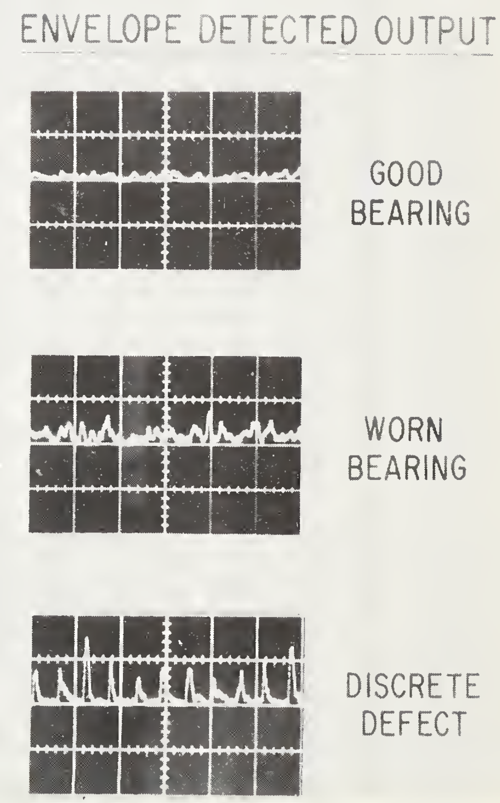


Figure 3.

DISCUSSION

William Glew, Naval Eng. Test Establishment: Have you managed to establish any amplitude criteria for good and bad bearings in these studies?

R. Burchill: What we use as a general guide is the fault indication we obtain from the first spall. When does one replace a bearing? That's a real serious question. We are in the process of working with an automotive manufacturer who is attempting to define the difference between the new condition and the rate at which the bearing fails, so we can look at it from the quality control standpoint. We expect that this program will extend also to attempting to run the bearing to the time of maximum useability and yet replace it before it fails. At this point we have not really established how far you can go.

William Glew: You are looking at a very high amplitude vibration measured in terms of acceleration. Are you measuring it quantitatively in terms of G levels or any other definite quantitative sort of dimension?

R. Burchill: The levels that we showed on that damaged bearing were in the range of about 10 G's peak. I haven't established that as any magic number, however.

William Glew: Yes, it looks to me as if potentially this is a very valuable tool in the early detection of bearing monitoring and the next stage would be to establish these high frequency vibrational mode G levels.

R. Burchill: With the bearing mounted in a real live structure, there are some impedance problems, or at least some impedance requirements, that one need keep track of, so that in establishing levels, one needs to look at the distance between the problem bearing and the sensor.

MECHANICAL SIGNATURE ANALYSIS OF BALL BEARINGS BY REAL TIME SPECTRUM ANALYSIS

By: A. S. Babkin, AVCO Lycoming and J. J. Anderson, Federal Scientific Corp.

Alexander S. Babkin

John J. Anderson

Mr. Babkin is supervisor of the acoustics group at AVCO Lycoming, Stratford, Connecticut, where he is responsible for all phases of acoustics, including design of silencers, noise testing, data analysis and predicting noise levels from engine design data. He received a BME Degree from McGill University in 1960 and an MSME Degree from Columbia University in 1967. He has 12 years of experience in sound and vibration testing and noise control. He is a member of the Acoustical Society of America and the Society for Experimental Stress Analysis.

Mr. Anderson is manager of application engineering at Federal Scientific Corporation, New York, N.Y., where he specializes in developing new techniques in the application of real-time spectrum analysis to machine diagnostics and production-line testing. He received a BEE Degree in 1954 and an MEE Degree in 1957, both from the City College of New York. He has 18 years experience in the design and application of instrumentation used in control systems and dynamic measurements. He is a member of IES, IEEE and ISA and a Registered Professional Engineer in the State of New York.

INTRODUCTION

Instrumentation for early-warning fault detection is critically needed in many industries, especially air transportation, where high reliability and safety standards result in large costs for quality control, expensive test programs, periodic overhauls and continuous inspection and maintenance. Although all of these activities are very important to the program to prevent catastrophic failures that could result in tragedy, some of the test procedures presently used are based on long-established methods that are tedious and often suspect. Disassembling a mechanical device to inspect internal parts can be a very thorough and meaningful method of preventive maintenance. However, it is costly and the potential for a fault caused by a poor reassembly sometimes outweighs the benefits of this type of visual inspection. If methods were available to determine the "internal health" of a device without disassembly, then maintenance and inspection costs could be reduced and reliability improved by lowering the potential for failures due to poor maintenance work.

To fully appreciate the value of such a method, one has to experience the frustration of having disassembled a faulty mechanism, and not finding a defect, and having the "fault" disappear after reassembly. How much better would it have been to have had a scientific "diagnosis" of the defect prior to disassembling the device for visual inspection or testing of internal parts.

Modern instrumentation techniques are now available to measure "mechanical signatures" of rotating devices. These signatures allow a diagnosis of internal health without having to disassemble the mechanism.

This report will discuss the ability of mechanical signature analysis to detect faults in high-quality ball bearings used in aircraft engines. Included in the discussion will be the technical concepts needed to understand mechanical signature analysis, the potential pitfalls to be avoided in performing the measurement and a description of the equipment selected for the test program. Mechanical signatures of a number of ball bearings will be illustrated and specific frequencies correlated with mechanical defects.

FIGURE 1. EXAMPLE OF HIGH QUALITY BALL BEARING USED IN TEST EVALUATION (SHOWN DISASSEMBLED)

DEVELOPMENT OF MECHANICAL SIGNATURES

A ball bearing represents a complex vibration system whose components are the balls, ball cage, inner race and outer race as shown in Figure 1. Like any other manufactured part, these components have degrees of imperfection and generate vibration by continually contacting each other during rotation. The amplitude of the vibration is dependent upon the energy of the impact, the point at which the vibration is measured, and the construction of the bearing. In ball bearings, the continuous contacting of the rolling elements results in a vibration pattern that can be mathematically correlated with the geometry of the mechanical parts of the bearing. The vibration signal will contain many frequency components due to the complex nature of the generation and transmission of vibration in mechanical structures.

Although bad bearings tend to generate vibration levels greater than the levels obtained from good bearings, the simple method of measuring peak vibration levels fails to detect a bearing with a defect that will shorten the life of the bearing and possibly cause breakdown failure. And yet, the defect, while not necessarily affecting peak vibration levels, is "crying" to be heard. Like one voice masked by the many voices of a choir, it may have a distinctive pitch that can be singled out if the ear is tuned to listen. Modern instrumentation techniques give us the means to listen properly to the wealth of information contained in the vibration of a bearing.

What is a Mechanical Signature? The vibration signal, as measured by an accelerometer or other motion transducer, can be electronically broken into its frequency components and their related amplitude levels. It is this plot of discrete amplitude vs frequency (narrow-band spectrum of the vibration signal) that is called the "mechanical signature" of the ball bearing, since it identifies the bearing and is unique to the unit selected. Figure 2 is a photograph of an oscilloscope display of the mechanical signature of a ball bearing used in this investigation. Many of the discrete frequencies contained in the mechanical signature can be related to specific mechanical occurrences within the bearing. The amplitudes of these frequencies are a measure of the energy transmitted in the occurrence and, therefore, the "smoothness" of operation. By measuring the amplitudes of specific frequencies, and performing relative comparisons, a decision can be made as to the "health" of a specific ball bearing.

Use of Spectral Orders. A comparison of the mechanical signatures of two ball bearings of the same type would require data obtained at identical speed, since most of the vibrational frequencies are proportional to speed. Rather than trying to hold speed constant, it would be better to have mechanical signatures independent of speed. This is accomplished by normalizing all frequencies relative to the fundamental rotational speed. This procedure is called "order normalization" where the fundamental frequency of rotation is called the first order. Applying certain assumptions, mathematical relationships can be derived between the rotational speed of the various bearing components and the rotational speed of the non-stationary race. When expressed as a relative speed ratio to the fundamental rotational speed, the component defect frequencies are identified in terms of orders rather than frequency.

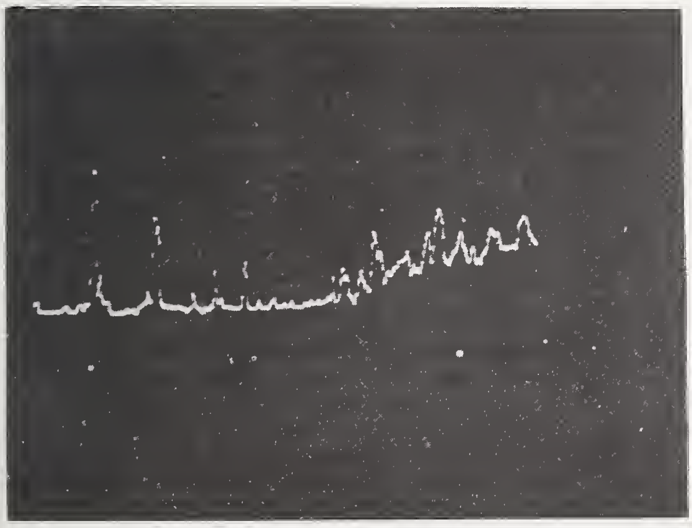


FIGURE 2. MECHANICAL SIGNATURE OF A BALL BEARING, AS SEEN ON AN OSCILLOSCOPE DISPLAY

Linear and Non-Linear Effects. The vibration spectrum of the ball bearing assuming rigid body motion is characterized by a large number of discrete peaks that can be divided into two categories--linear and non-linear effects. Assuming linear deformation at the contact point and flexurally rigid bearing races the frequencies due to linear effects can be theoretically calculated from the bearing geometry. The computed frequency orders can be readily found in the mechanical signatures of ball bearings. The non-linear effects are associated with the non-linear elastic behavior of the bearing structure. Frequencies associated with non-linear effects are primarily found at low speeds and with low loads. The source of the non-linear vibrational orders are large ball diameter variations, eccentricity and low-order waviness of the races. As the speed or the load is increased, linear effects predominate in the mechanical signatures.

Frequencies Related to Bearing Geometry. The vibrations generated by a ball bearing will cover a wide range of frequency. However, in deriving the equations for analysis of the mechanical signature the ball bearing was assumed to have rigid body motion. For this assumption to be valid, signature analysis will be limited to a frequency range including up to eight (8) orders of fundamental rotation. At higher frequencies bearing defects excite the elastic modes of the bearing races. Also, random noise due to friction dominates the vibrational amplitude levels at the higher frequencies. Preliminary studies showed that the analysis of low-order mechanical

From the geometry of the ball bearing, formulas can be derived to correlate mechanical defects with vibrational frequencies. The geometry of the bearing will be defined in the following terms:

ω = inner race rotational speed, rpm
d = diameter of ball element
D = pitch diameter of the bearing
α = contact angle of the balls to the raceway
Z = number of balls
n = 1, 2, 3...index number

The fundamental frequency of rotation assuming a stationary outer race is given by

$$f_r = \frac{\omega}{60}$$

The order of the rotating ball set (cage frequency) relative to the stationary outer race and normalized by the fundamental frequency, f_r can be calculated as: (Ref. 1)

$$f_c = \frac{1}{2}\left[1 - \left(\frac{d}{D}\right)\cos\alpha\right]$$

The order of the rotating ball set (cage) relative to the rotating inner race and normalized:

$$f_i = \frac{1}{2}\left[1 + \left(\frac{d}{D}\right)\cos\alpha\right]$$

The order of the rotating ball relative to the stationary outer race and normalized:

$$f_b = \frac{D}{2d}\left[1 - \left(\frac{d}{D}\right)^2 \cos^2\alpha\right]$$

Using these three orders of rotation, the orders associated with various bearing faults can be calculated. Table 1 lists the formulas for vibrational orders that would be found for various types of surface imperfections.

MEASUREMENT CONSIDERATIONS

In obtaining the mechanical signature of a ball bearing, there are four major sources of error that must be considered in the measurement. First, it is important to establish what frequencies are present in the measurement that are not associated with the ball bearing, but can be attributed to external forces or connecting drive elements. Second, speed changes will cause shifts in the frequency spectrum, and will cause inaccuracies in both the amplitude and frequency measurement. It is therefore important that speed be controlled very accurately or a means provided in the instrumentation to correct for speed changes. Third, the random nature of the vibration requires spectral averaging to enhance the signal-to-noise ratio and provide statistical accuracy to the amplitude measurement. Lastly, the time required to obtain the mechanical signature must be minimized, so as to reduce variations due to temperature, lubrication, wear and calibration. In selecting the equipment to be used in obtaining mechanical signatures, these major considerations become interdependent due to the instrument techniques applied to the measurement.

Drive Isolation - The measurement of the vibration of precision parts is obscured in many cases by vibration generated in the drive elements and transmitted through rotary couplings and mounts. Even when care is taken to isolate the drive vibration, the amplitude levels typically associated with the bearings are so low that many frequencies evident in the spectrum can be correlated with external forces. When these externally induced vibrations have sufficient frequency separation from the internal vibrations, and are of low amplitudes so as not to limit dynamic range, the drive frequency interference can be tolerated. This is not normally the case, however, since vibrations associated with the drive are often the same frequencies or very close to bearing frequencies. This is particularly true of the first 10 orders of rotation when direct coupling is used between the drive and bearing. Beat frequencies (those frequencies equal to the sum and differences of mixed frequencies) also

TABLE 1. VIBRATIONAL ORDERS GENERATED BY SURFACE DEFECTS ON A BALL BEARING

TYPE OF SURFACE DEFECT		VIBRATIONAL ORDERS	
BALL PART	IMPERFECTION	LINEAR THEORY	NON-LINEAR THEORY
Inner race	Eccentricity	1	—
	Waviness	$nZf_i \pm 1$	$nf_i \pm f_c$
	Rough spot	nZf_i	—
Outer race	Waviness; rough spot	nZf_c	$(n \pm 1)f_c$
Ball	Diameter variation	f_c	—
	Waviness	$2nf_b \pm f_c$	—
	Rough spot	$2nf_b$	—

tend to interfere with the measurement of mechanical signatures.

Therefore, great care must be taken to minimize vibration not directly associated with the rotation of the bearing. The test equipment selected in this study to measure ball bearing mechanical signatures de-couples the drive prior to the start of the measurement. The ball bearing is allowed to coast down in speed, while loaded with a flywheel that rests on the inner race of the ball bearing while the rotational axis is oriented in the vertical plane. The kinetic energy available in the flywheel at the moment of de-coupling from the drive will be slowly dissipated in the torque disturbances occurring within the bearing. These disturbances generate the vibration sensed by the accelerometer mounted to the fixed outer wall of the ball bearing. A "rough" bearing will dissipate more energy and, therefore, the time to coast to a stop will be shorter.

Speed Stability - The selection of the flywheel drive imposes an additional requirement on the spectrum analysis equipment; i.e., the spectrum must be obtained independent of speed. A technique must be applied to generate spectral orders or amplitude components associated with mechanical occurrences per revolution (i.e., 1, 2, 3, 4.6, 5.2, 7, etc.) and independent of the rotational speed. A technique known as "order normalization" can be applied when a tachometer is available that generates a pulse train, the frequency of which is proportional to the speed of the ball bearing. Order normalization was incorporated in the instrumentation and further description of this technique will be presented in the equipment discussion under test techniques. In this way inaccuracies in frequency measurement due to variations in speed were eliminated.

Signal-to-Noise Enhancement and Repeatability - In addition to detecting faults, it is also desired to establish a criterion for setting quality levels for ball bearings. Since each bearing varies in quality and no bearing component is without some imperfections, the measuring instrument should detect rotational frequencies associated with bearing defects. However, these frequency amplitudes may be obscured by the broadband noise levels associated with bearing friction, external vibrations, and instrumentation noise. A spectrum averaging technique can be applied to enhance the signal-to-noise ratio of the periodic discrete frequencies generated by the ball bearing under test. In addition to increasing signal-to-noise ratio, the technique of spectral averaging increases the statistical accuracy of the measured amplitudes. Exploratory tests were conducted to determine the number of averages required to obtain repeatable results within the accuracy of the instrumentation and the limitations of the mechanical test equipment. A total of 64 averages was determined as the optimum number to obtain a repeatable spectral density. This number of averages could easily be obtained by a real-time spectrum analysis system during the two minutes it took to coast from 3600 rpm to 2400 rpm.

Evaluation Time - The selection of a flywheel drive imposes limitations on the time available to measure a mechanical signature. Statistical accuracy and signal-to-noise enhancement require sufficient length of data (time) to allow spectral averaging. However, the slowing down of the flywheel speed also varies the amplitudes of the spectral frequencies. This amplitude variation introduces another limitation to the measurement. A compromise is required in selecting the maximum speed range, therefore limiting the test time available to measure the mechanical signature. The selection of a Ubiquitous© real-time spectrum analyzer provided the high-speed analysis to allow sufficient spectral averaging within the limited test time.

INSTRUMENTATION AND TEST TECHNIQUES

The instrumentation and test fixturing had to be selected with the intention of optimizing accuracy and minimizing the potential errors discussed in the preceding section. Figure 3 is a photograph of the test set-up selected to obtain the mechanical signatures of the ball bearings investigated in this program. The equipment consists of a Kostron Bearing Tester that provides the motive force necessary to get the bearing/flywheel combination up to speed before allowing it to coast. The vibration of the test bearing is sensed by an accelerometer mounted to the outer casing of the bearing.

The vibration signal is conditioned by a charge amplifier and then processed by a real-time, 500-line spectrum analyzer to develop the mechanical signature. A 500-line spectral averager provides signal-to-noise enhancement and statistical reliability (repeatability). A photoelectric pulse generator provides a speed-proportional signal that is used by the spectrum tracking adapter to normalize the frequency spectrum in terms of rotational orders independent of speed. For this trial program, mechanical signatures were continuously observed on the oscilloscope display and permanent records were obtained by plotting X-Y graphs of the mechanical signatures digitally stored in the spectral averager. The block diagram of the instrumentation package, showing signal interconnections is illustrated in Figure 4.

Kostron Bearing Analyzer - The mechanical fixture used in these tests to impart the rotation to the test ball bearing was developed by Bendix under the trade name Kostron. The device is used to measure bearing vibration characteristics in such a manner that the rotating energy source does not introduce significant spectral components into the mechanical signature of the test bearing. This is accomplished when a high-inertia flywheel is brought up to speed by an electric motor and then lowered to engage the inner casing of the test bearing. The drive motor is disengaged from the flywheel and the kinetic energy of the flywheel is allowed to dissipate itself in the friction and torque disturbances of the test bearing supporting the flywheel. Once free-wheeling, the flywheel takes approximately eight minutes to dissipate its energy in a typical sample from the type bearings used in this program. The Kostron Bearing Analyzer incorporates a built-in

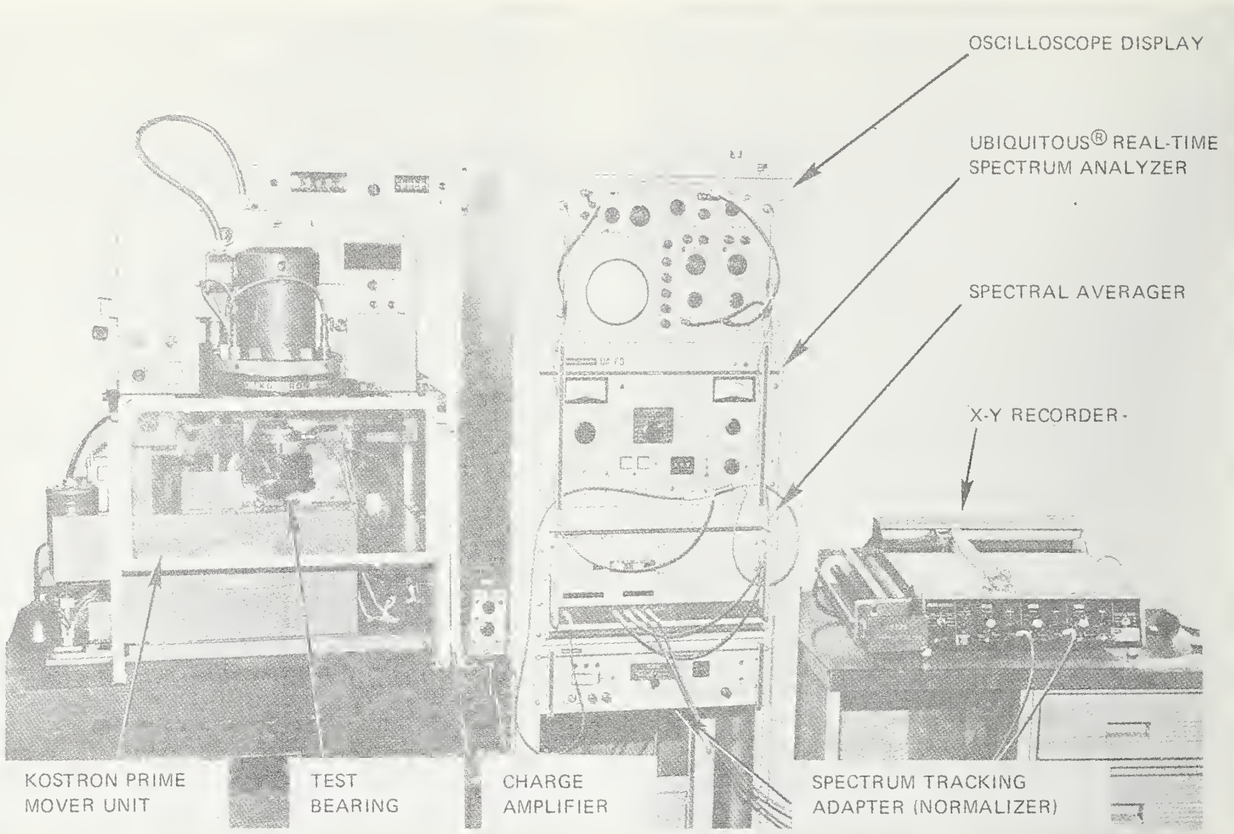


FIGURE 3. BEARING DEFECT DETECTION SYSTEM

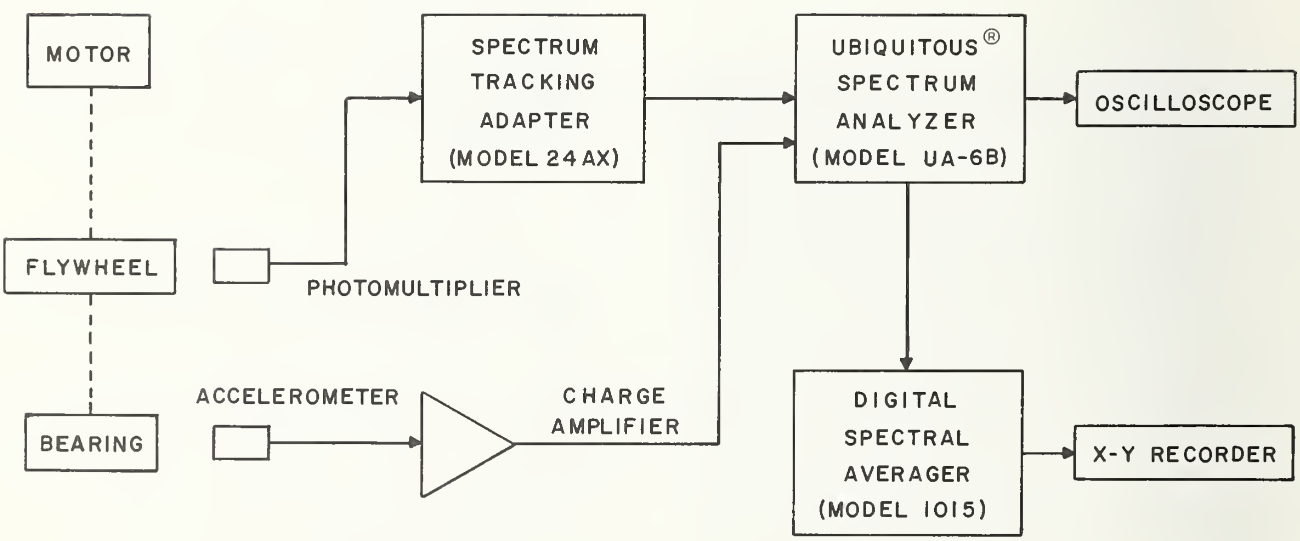


FIGURE 4. BLOCK DIAGRAM OF INSTRUMENTATION

optical tachometer to produce a digital display of bearing speed. This tachometer, generating 60 pulses per revolution, is used to provide order normalization of the spectral data.

Vibration Transducer - The vibration generated by the motion of the ball bearing is sensed by a piezoelectric accelerometer, rigidly attached to the outer casing of the test bearing. The sensor must have a frequency response sufficient to cover the data range of interest. Limiting analysis to a range where rigid body motion of the bearing is a valid assumption, the measurements were made over the first eight orders. Data is processed while the bearing coasts from 3600 rpm to 2400 rpm. This is equivalent to a first-order frequency ranging from 60Hz to 40Hz. The eighth order would then range from 480 to 320Hz. The lowest order of interest is 0.4, this being equivalent to a frequency range from 24 to 16Hz. Therefore, the sensor and instrumentation must have good low-frequency response and be essentially flat from 16Hz to 480Hz to provide the required signatures.

The accelerometer selected for this test program was a B & K type 4332, having a flat frequency response from 0 to 6000Hz $\pm 2\%$. Low-frequency cut-off is generally determined by the conditioning amplifier and high-frequency response is limited by the accelerometer mounting characteristics. To maintain good low-frequency cut-off, a charge amplifier was selected for conditioning the accelerometer output signal before processing the voltage in the real-time spectrum analyzer. The charge amplifier provides low-frequency cut-off below 1Hz. The resonance of the accelerometer mounting used was assumed to occur at approximately 2000Hz. The frequency response of the input circuitry to the real-time analyzer has lower and upper frequency break points at 1.7Hz and 600Hz.

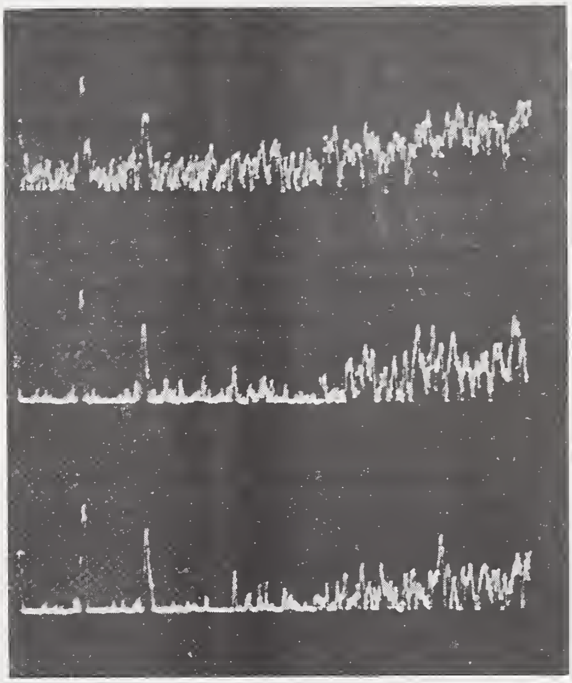
Therefore, the instrumentation selected for the test is essentially flat over a range of 5Hz to 500Hz and accurate to within $\pm$1dB over the 16-to-480Hz range used in the determination of mechanical signatures.

Real-Time Spectrum Analyzer - The heart of the instrumentation is the Federal Scientific Ubiquitous® Spectrum Analyzer. The accelerometer signal is analyzed for frequency/amplitude components by the Model UA-6B, 500-line analyzer. A real-time spectrum analyzer was selected for this test to enable the high-speed spectral processing required in the study. The UA-6B is a hybrid spectrum analyzer (analog and digital) that uses the digital time-compression technique to expand the low-frequency input signal to a high-frequency waveform equivalent. This high-frequency signal is spectrum analyzed using a fast-settling broadband analysis filter. (A technical description of the performance of this type spectrum analyzer is contained in References 2 and 3.) The selection of a real-time analyzer in this test program is based on its ability to process 100% of the vibration data available in the short period of slowdown between 3600 and 2400 rpm. In less than two minutes the real-time analyzer generates 64 statistically independent spectra in the frequency range of interest to this study.

Spectrum analyzers using analog sweeping methods would take 10 minutes to generate only ONE independent spectrum, or over 10 hours for a 64-spectrum average. Not only is the slowness not acceptable to this program, but earlier analyzers processed very little of the available data and many of the identifiable frequency components were lost.

Spectral Averager - The mechanical signature of the ball bearing is obtained by averaging a number of statistically independent spectra using a Federal Scientific Model 1015 Spectral Averager. Figure 5(a) is an oscilloscope photograph of the frequency spectrum obtained from a sample ball bearing. The spectrum was obtained at 3400 rpm using signal data lasting 0.85 seconds or one real-time period (i.e., sufficient time to generate one statistically independent spectrum). Figures 5(b) and 5(c) show spectra obtained at 3000 rpm and 2600 rpm, respectively, using approximately 0.96 and 1.1 seconds of time data. In each photograph, the spectrum is speed-normalized, so that each trace represents 8.3 orders of rotational speed.

The effect of speed variation on the vibration amplitudes of the major orders can now be compared. The upper trace clearly shows the increased, broadband noise present at higher speeds. This noise


(a)

(b)

(c)

FIGURE 5. FREQUENCY SPECTRA OBTAINED AT 3400, 3000, AND 2600 RPM

tends to obscure the stable discrete frequencies present in the mechanical signature. Figure 2 is the mechanical signature of this same bearing obtained by averaging 64 spectra within the speed range of the spectra shown in Figure 5. The study proved that to generate repeatable mechanical signatures of ball bearings, both spectral averaging and order normalization was required. A total of 64 spectrum averages was selected as a compromise between increasing statistical accuracy and minimizing the range of decaying speed for the test data. Since vibrational amplitudes become very low at low speeds, test data were restricted to speeds above 2400 rpm.

Spectrum Tracking Adapter - Spectral averaging is required to enhance the vibrational signals that are buried in the random noise and to establish repeatability by statistical averaging. Figure 6 shows the time history of spectra during coastdown from 3400 rpm. The sliding frequency pattern, due to speed slowdown, is evident in the time history. When the spectra are normalized with respect to speed, the frequency sliding is eliminated, as shown in the time history of Figure 7. The spectra are said to be "order normalized."

The Federal Scientific Model 24AX Spectrum Tracking Adapter accomplishes order normalization by using the tachometer pulse signal available from the Kostron equipment. The tachometer pulse rate, which is proportional to bearing speed, is used to control the sampling rate of the Ubiquitous® Spectrum Analyzer. In this way, the sampling rate of the vibration signal varies with speed and, therefore, automatically compensates the spectrum for speed variations. (This technique is discussed in greater detail in Reference 4.) The use of order normalization is even more important when averaging is required. Figure 8 shows the spectrum time history without normalization and Figure 9 with normalization. In each photograph every 16th spectra only is displayed for comparison.

When averaging the sliding frequency spectra, the energy of the orders is distributed over more than one spectral address of the digital averager. The effect is to smear the peaks and this is especially evident in the higher orders. For example, the energy of the eighth order will be distributed in eight times as many addresses as the energy of the first order. Figure 10 shows the same spectra as Figure 8; Figure 11 shows the same spectra as Figure 9, but, instead of showing every 16th spectra, each trace represents the average of the previous 16 spectra. Whereas the normalized time history shows consistency in the spectral amplitudes, the time history not using order normalization broadens the spectrum peaks and does not maintain the shape of the mechanical signature spectrum.

MECHANICAL SIGNATURES OF BALL BEARINGS

The bearings selected for this study were split inner race, angular contact ball bearings used in jet aircraft engines. Forty bearings were tested, six were new, 28 used and six were reclaimed bearings. The new bearings were purchased and inspected for defects both visually and by measurement. The used bearings were obtained from the AVCO Lycoming overhaul facility and the load/life history was not available. The reclaimed bearings are used bearings that had defective parts replaced by new parts. Present inspection methods for these bearings, including visual inspection and probing with styli are slow, tedious and, in many cases, unreliable. A faster automatic inspection system, not depending on operator judgment is preferred. An automatic high-speed system proposed for this measurement is shown in Figure 12. The tests of this program were carried out to determine if bearing defects could be sensed and identified by the selected instrumentation.

Bearing Frequencies - The bearings used for analysis had the following geometry:

Ball diameter	(d)	=	0.50000 inch
Pitch diameter	(D)	=	2.75565 inch
Contact angle	(α)	=	28.5 degrees
Number of balls	(Z)	=	14

Using these measurements in the formulas for bearing frequencies we obtain:

$$f_c = 0.42$$
$$f_i = 0.58$$
$$f_b = 2.69$$

With these basic rotational orders, the formulas for surface defects contained in Table 1 are used to calculate all possible frequencies that may exist in the mechanical signature of the ball bearing. Table 2 lists all vibrational orders up to 8.54 using both linear and non-linear theory.

Test Results - The mechanical signatures of four of the bearings selected for discussion are shown in Figures 13, 14, 15 and 16. The amplitude of the spectrum is plotted in log scale to provide the greatest dynamic range on one chart. This will allow the detection of small defect frequencies in a measurement containing a large frequency component. The dynamic amplitude range of the instrumentation is better than 54dB.

A total of 8.3 orders was selected for plotting as this range includes all major linear orders. Above this range, order-related vibrations tend to excite the elastic modes of the bearing races. Also, random noise due to friction dominates the spectrum, making it difficult to measure frequencies that can be correlated with bearing defects.

The mechanical signature of a good bearing is shown in Figure 13. The amplitude is calibrated for 90dB equals 0.26 g and this scale is used on all graphs. The noise floor for all mechanical signatures is approximately 50dB or 0.0026 g. The first order is the only frequency evident in this mechanical signature. Although the first order can be correlated with inner race eccentricity, the 1, 2, 3, and 4 orders obtained on graphs are assumed to be caused by the imbalance of the large flywheel used in the Kostron.

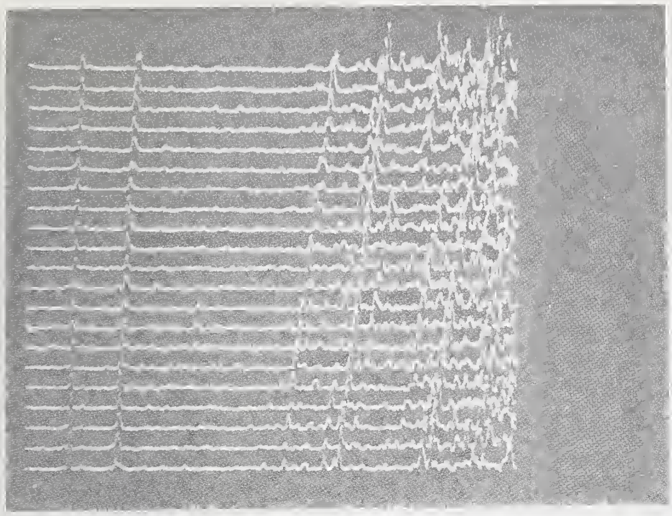


Figure 6. Spectral Time History - No Order Normalization

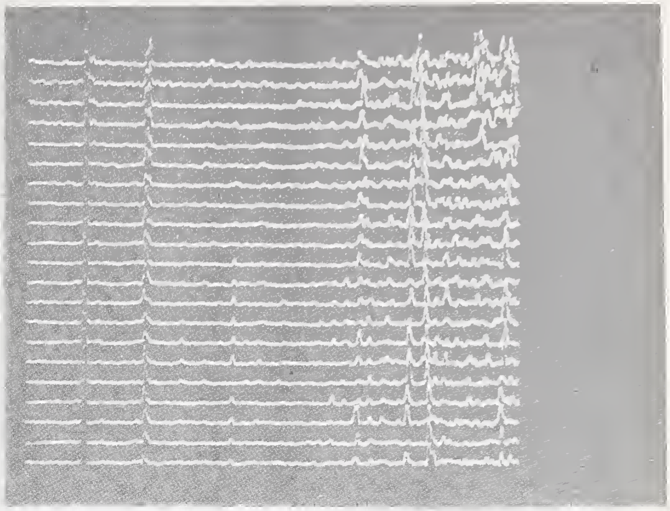


Figure 7. Spectral Time History - With Order Normalization

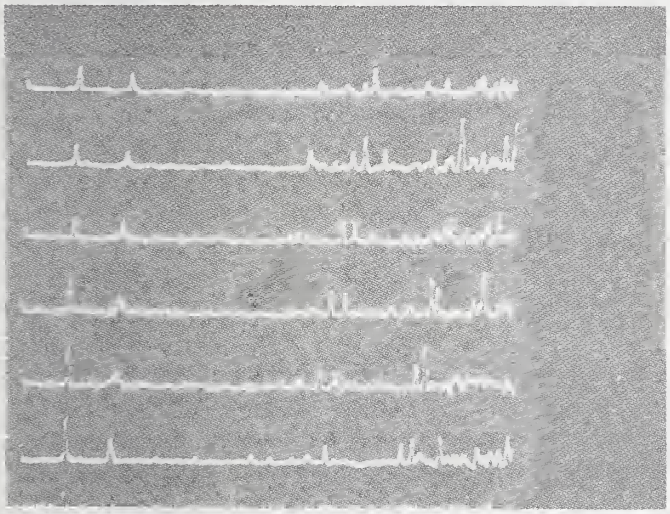


Figure 8. Spectral Time History - No Order Normalization - Every 16th Spectra

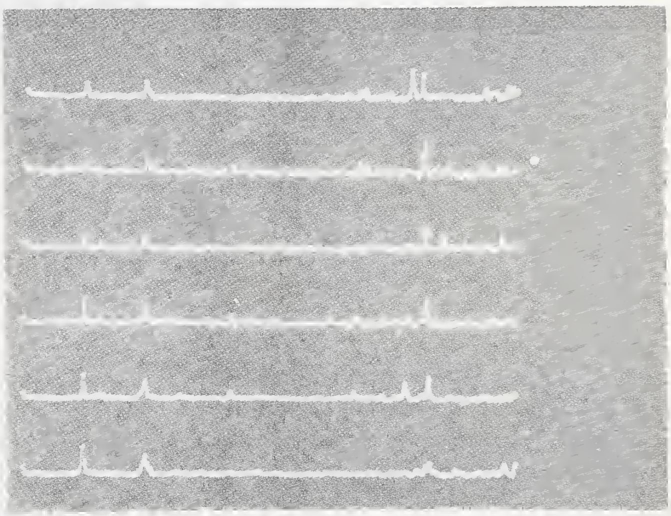


Figure 9. Spectral Time History - With Order Normalization - Every 16th Spectra

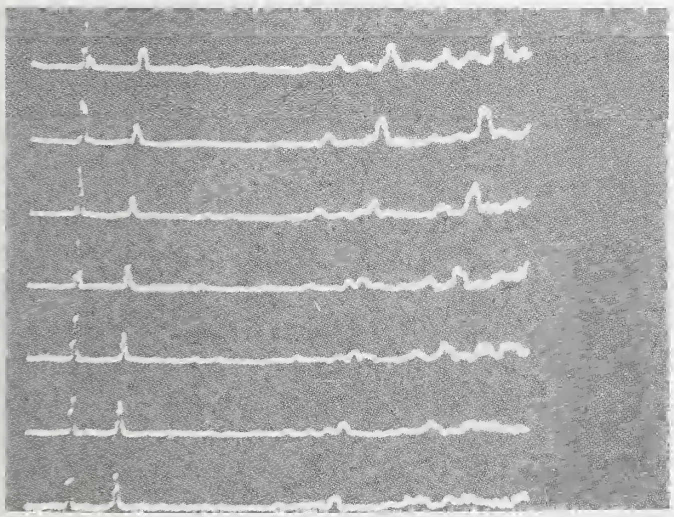


Figure 10. Spectral Time History - No Order Normalization - 16 Spectra Averaged

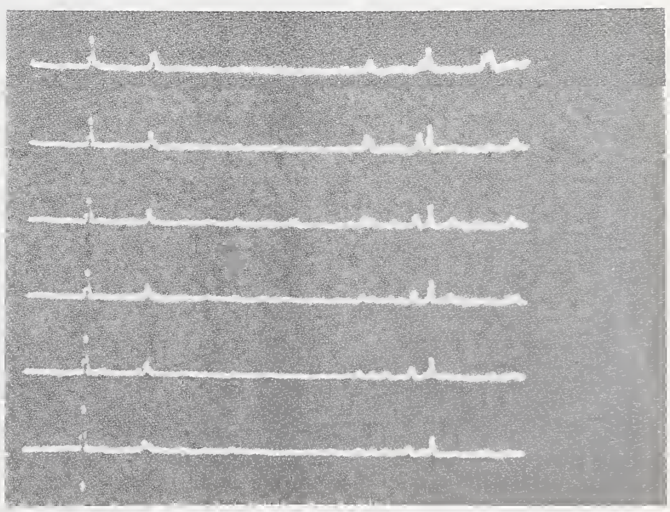


Figure 11. Spectral Time History - With Order Normalization - 16 Spectra Averaged

TABLE 2. VIBRATIONAL ORDERS ASSOCIATED WITH A BALL BEARING HAVING 14 BALLS, 0.5" BALL DIAMETER, 2.76" PITCH DIAMETER AND 28.5° CONTACT ANGLE

LINEAR THEORY			NON-LINEAR THEORY				
BALL	INNER RACE	OUTER RACE	INDEX	INNER RACE		OUTER RACE	
–	–	–	1	0.16	1.00	0	0.84
0.42	–	–	2	0.74	1.58	0.42	1.26
–	1.00	–	3	1.32	2.16	1.68	DUPLICATED ORDERS
–	–	–	4	1.89	2.74	2.10	
–	–	–	5	2.48	3.32	2.52	
–	–	–	6	3.06	3.90	2.94	
4.96	–	–	7	3.64	4.48	3.36	
–	–	–	8	4.22	5.06	3.78	
5.38	–	–	9	4.80	5.64	4.20	
–	–	–	10	5.38	6.22	4.62	
5.80	–	5.88	11	5.96	6.80	5.04	
–	–	–	12	6.54	7.38	5.46	
–	7.12	–	13	7.12	7.96	5.88	
–	–	–	14	7.70	8.54	6.30	
–	8.12	–	15	8.28	–	6.72	
–	–	–	16	–	–	7.14	
–	9.12	–	17	–	–	7.56	
–	–	–	18	–	–	7.98	

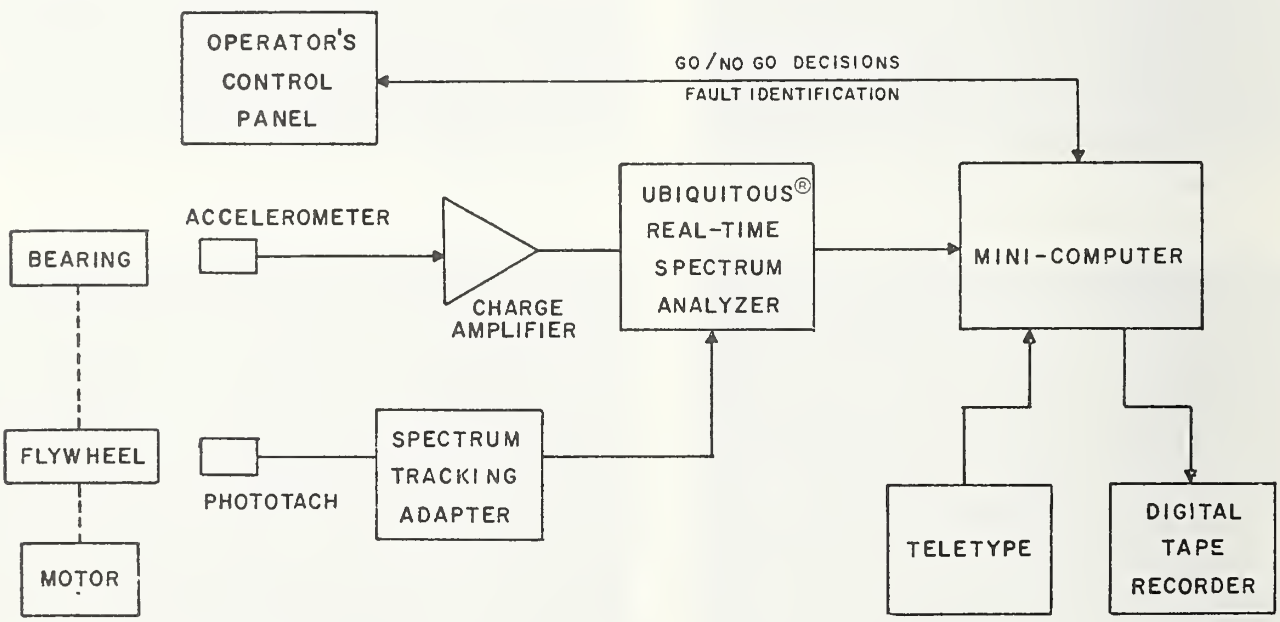


FIGURE 12. BLOCK DIAGRAM: PROPOSED INSTRUMENTATION FOR HIGH-SPEED EVALUATION OF BALL BEARINGS

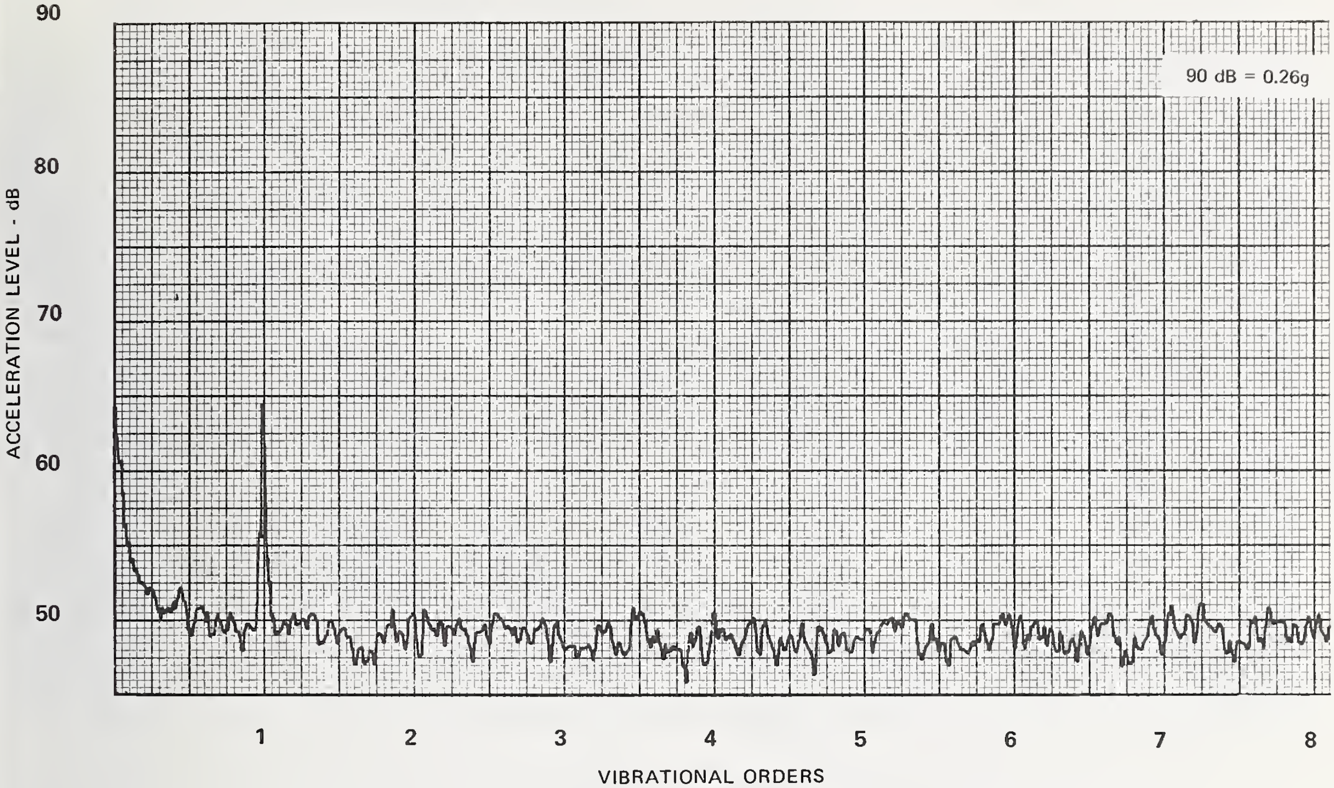


FIGURE 13. MECHANICAL SIGNATURE OF BALL BEARING WITH NO DEFECTS

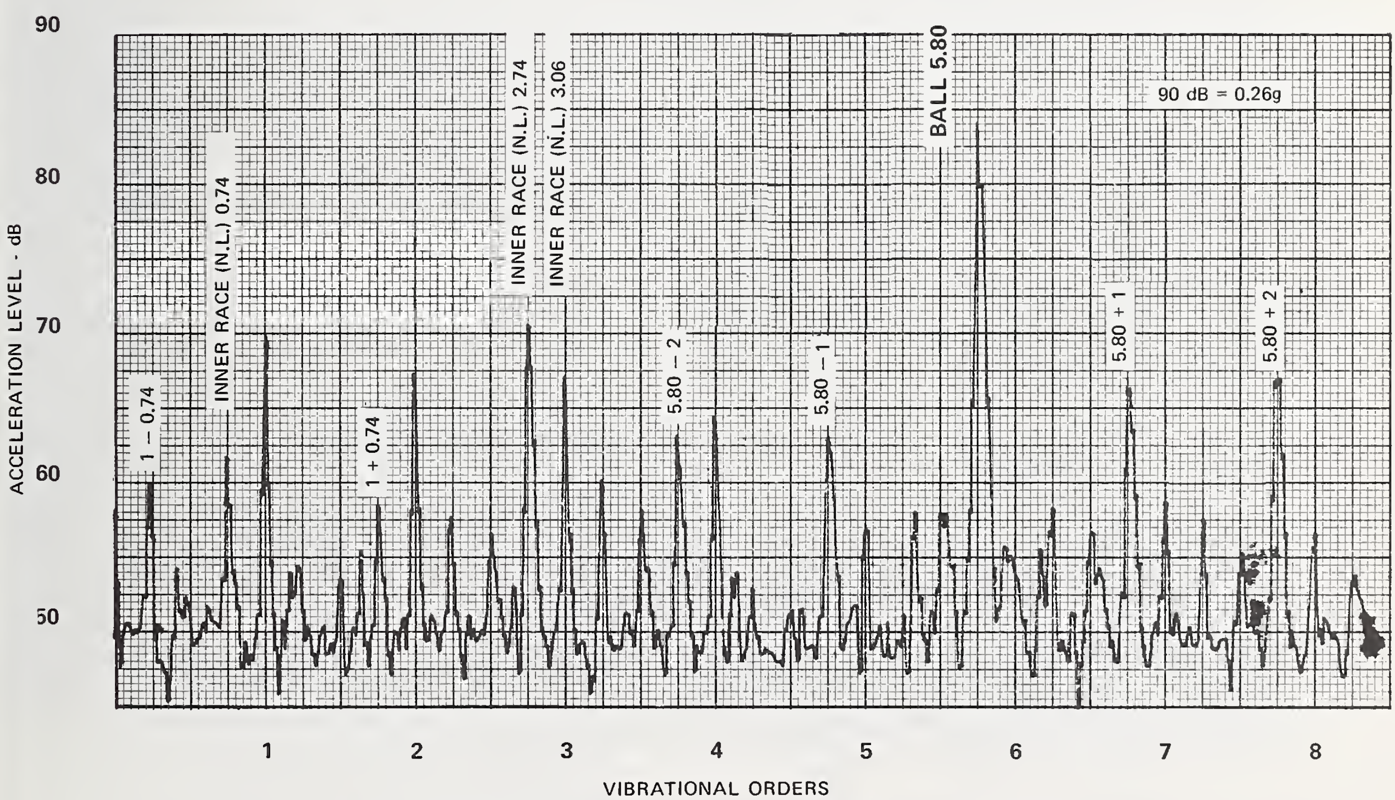


FIGURE 14. MECHANICAL SIGNATURE OF BALL BEARING SHOWING BALL DEFECT

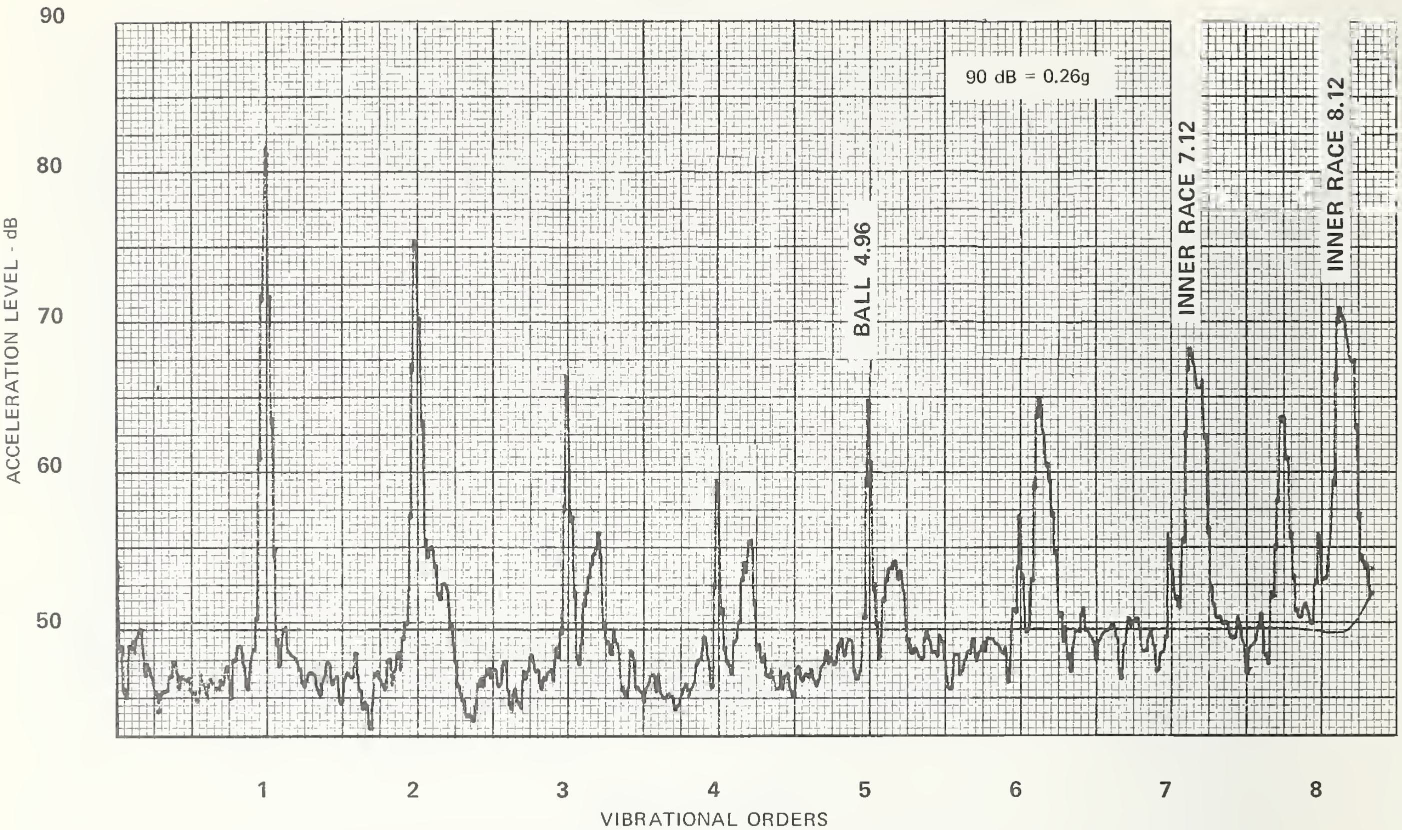


FIGURE 15. MECHANICAL SIGNATURE OF BALL BEARING SHOWING INNER RACE DEFECT

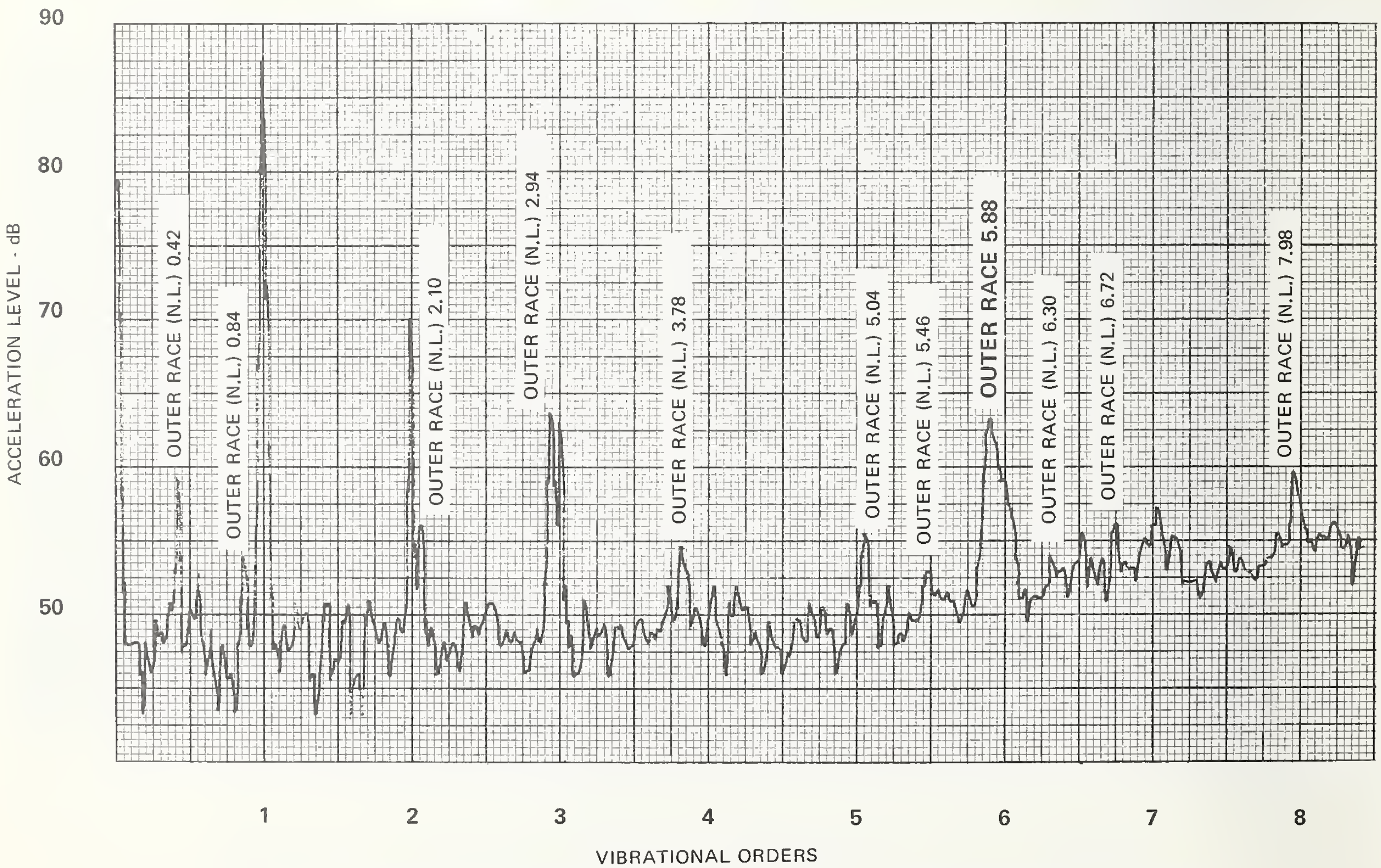


FIGURE 16. MECHANICAL SIGNATURE OF BALL BEARING SHOWING OUTER RACE DEFECT

A mechanical signature showing a ball defect is illustrated in Figure 14. The presence of two large orders (5.80 and 1.00) will normally generate beat frequencies that can be identified (5.80 $\pm$ 1.00, 5.80 $\pm$ 2.00). This bearing also shows orders associated with inner race defects as determined from non-linear theory.

Figure 15 illustrates the mechanical signature of a ball bearing showing an inner race defect. This signature also shows beat frequencies spaced one order apart with lesser amplitudes. The presence of a slight ball defect is assumed since the signature also shows a 6dB increase of the fifth order over the fourth and sixth orders.

An outer race defect is illustrated in the mechanical signature shown in Figure 16. Linear theory correlates the peak found at the 5.88 order. Numerous other peaks can be correlated with the outer race defects calculated using non-linear theory.

Correlation With Defects - Bearings having mechanical signatures containing discrete frequencies that could be equated to imperfection frequencies were disassembled for inspection. In all cases, defects indicated by large amplitude orders could be found in the inspection process. However, many of the defects indicated by the lower amplitude orders could not be found by present inspection procedures. It is felt that the instrumentation developed for this study has greater ability to detect small bearing imperfections than present inspection procedures. It remains for further studies to determine how much these lower order imperfections will reduce bearing life.

Test Time - For the results obtained in this investigation, mechanical signatures could be recorded at a rate of one bearing every five minutes, with more than one-half of the test time expended in setting the bearing in the test fixture. Recordings of mechanical signatures stored in the averager can be plotted simultaneously with the changing of test bearings. The time required for analysis is not easily determined because the factors to be evaluated (i.e., the number of spectral orders, amplitudes, go/no-go reference levels, and multiple weighting factors) have not been determined.

The instrumentation is capable of faster measurement and evaluation by introducing a mini-computer to the set-up. The computer is programmed for calibration and transducer equalization modes. Data will be obtained for peak amplitudes at selected spectral points and recorded on high-speed digital magnetic tape or slower-speed paper tape for processing statistical data off-line. Immediate decisions can be made for quality acceptance testing. The mechanical signature is compared to a reference signature stored in the mini-computer and a go/no-go indication given to the operator. With reduced handling of ball bearings by improved fixture design, the rate of high-quality testing and evaluation can be reduced to less than two minutes. Incorporating a mini-computer into the instrumentation system not only provides a new technique for high-production quality control but also allows for a statistical data bank of bearing defects that can be used to revise designs or alter manufacturing processes.

Figure 12 is a block diagram of the instrumentation proposed for high-speed evaluation of ball bearings.

Figure Studies. It is planned to extend the investigation beyond the objective of the present study. With the ability to correlate vibrational orders with bearing imperfections, future studies will attempt to develop a quantitative relationship between spectral amplitudes and severity of the defect. Also, a criterion will be sought for predicting the remaining life in a bearing showing a lower amplitude imperfection.

These studies will require a large sample of new bearings, all tested and accepted for high quality and displaying "good" mechanical signatures. The bearings will be subjected to controlled service in test rigs and periodically tested to obtain a time history of mechanical signatures. Bearings will be disassembled to obtain corroborating inspection data when variations are found in the mechanical signature.

CONCLUSION

This study was undertaken with the knowledge that earlier attempts at mechanical signature analysis of ball bearings were unsuccessful, or at best, not practical for high-speed test evaluations. The majority of the earlier studies suffered from limitations inherent in the instrumentation used for the measurement of the signature.

The present investigation was successful in developing test techniques using standard commercial instrumentation to obtain repeatable mechanical signatures of ball bearings. The frequencies found in the mechanical signatures were correlated with vibrations related to specific parts of the ball bearing.

Within the limits of the statistical sample of ball bearings tested for this study, a trend was indicated in the ability to measure bearing quality. The amplitude of each frequency is assumed to be a measure of the severity of the part roughness or bearing defect. The detailed correlation of spectral amplitudes with the severity of defects, and more important, the remaining useful life, will be the subject of a future study.

REFERENCES

1. Gustafson, O.G., et al, Dec., 1963, STUDY OF VIBRATION CHARACTERISTICS OF BEARINGS, SKF Industries, AD432-979, 177 pp.

2. Monograph #3, June 1971, REAL TIME SIGNAL PROCESSING IN THE FREQUENCY DOMAIN, Federal Scientific Corporation, 21 pp.

3. Bickel, H.J., March 1971, "Real-time Spectrum Analysis", SOUND AND VIBRATION 5(3):14-20

4. Anderson, J.J., Dec. 1971, "Real-time Spectrum Analysis in Vibration Testing", ISA TRANSACTIONS, 10(3):269-276

DISCUSSION

Lee Doubleday, Naval Air Systems Command: Do you operate the bearing under any load at all? You would have to know what kind of loading caused the failure if it is loaded against the race.

A. Babkin: Loading in this case is a fly wheel which is about 25 lbs. in weight. This is a thrust load applied to the bearing. The load is constant.

John Jamieson, Kennedy Space Center: Doesn't the fact that you are using a thrust loading on that bearing make the track to one side of its normal operating point, if that bearing wasn't thrust loaded in the machine it was used in? Doesn't that change what balls run?

A. Babkin: This angular contact bearing, as any split race bearing is, was loaded in the same direction as it is loaded in the engine. It has to be, otherwise it can't operate. The whole thing is the fly wheel. It is suspended on air jets so that we don't get any frictional effect and we have a constant thrust load on the bearing.

Jerry Forest, Ontario Hydro: During the run down of the fly wheel, won't all the forces acting on the bearing change? Centrifugal loads will change. I assume, like you say, there is some unbalance. The magnitude of the signals at the higher orders should be somehow related to these forces. How does this affect the analysis?

A. Babkin: As the fly wheel is coasting down, the magnitude of the disturbing forces also decreases. You could see this from the slide that showed repetitive average spectra. We based our analysis on 164 or 128 seconds. So basically, most of the information was derived in the beginning when the bearing was under relatively high speed. As it slowed down, useful information was lost. A time limit of 2 minutes was used, because beyond that point, there was no information left.

J. Anderson: Data was obtained between 3 thousand and 2 thousand RPM's. It took anywhere, depending on the quality of the bearing, from 8 to 15 minutes for that fly wheel to come to a halt. We limit ourselves to an area of 2 minutes.

PANEL DISCUSSION

Panel Members: R. M. Whittier, K. Smith, R. F. Burchill, J. J. Anderson, A. S. Babkin, and R. Misialek

Panel Moderator: Paul Howard

Jerry Forest, Ontario Hydro: It seems to me, in comparing all 3 basic systems of diagnosis or detection that have been presented, that the crest factor approach presents a very simple and straight forward technique. Perhaps this is a good point to discuss because some of the other problems that were mentioned, like changing frequencies, at least the way I understand it, should be taken care of by the crest factor approach. Mr. Smith mentioned that crest factor can peak at 5 to 7 times that which is considered normal or satisfactory. Is this a universal number? If it is, I think it seems to be a very good approach to failure detection.

K. Smith: In the majority of cases that we have looked at where the signal is detected from the accelerometer placed somewhere in very close proximity to the bearing, the results have been very repeatable. The normal bearing runs at something around 2 to 3, and the crest factor runs something around 6 to 7 for a small spall. So in that sense, it's probably repeatable. If you modify the signal in some way be adding large signal components on top of it, you're going to dilute that crest factor measurement, so it is important that you have a good signal to noise ratio. It is a simple technique, but of course like anything else, you never get something for nothing. First of all the crest factor monitoring technique does not discriminate all by itself whether the defect is located on the inner race or the outer race, or on the ball, or someplace else. If you want to determine where it is located, you have got to look at some of the other parameters that have been measured such as the impact rate or the calculated resonance response of the structure that's being excited. In terms of just detecting that a particular bearing has a fatigue fault present in the very early stages, it's a very simple technique, and apparently from all our results, a very repeatable one.

Paul Howard: Would each of the gentlemen on the panel highlight the similarities and differences of your systems?

R. Burchill: Mr. Smith has commented that the background noise is a very important facet in the crest factor approach. If the level is too high, other conditioning is required. The advantage of the resonant structure technique is that, by using a very selective frequency range, we've been able to reject a majority of the background noise. The information that we are looking at then is related to the bearing itself. By using a very high frequency range, we are also isolated from many of the problems that are experienced by attempting to detect bearing faults along with other machinery problems. This has been either attempted or done by the Federal Scientific people.

A. Babkin: Basically, the technique that we described was more selective. It not only indicates that there is a defect in a bearing, but indicates which part the defect is in. Now the crest factor technique works but, as Mr. Smith pointed out, it indicates only that there is a defect present. You can do the same thing by looking at the amplitude distribution functions. If you plot the distribution of amplitude as a bell chart you can see if there is a fault in a bearing, the plot will be very flat with a lot of peaks, and if the bearing is acceptable the plot would be much narrower, so your signal image will be further apart. In our technique we use a normalization method to get away from a speed dependency which other techniques utilize. In Mr. Burchill's case he is using a higher mode resonance of a race to detect a peak. This will be specific only to a particular bearing. If he changes the bearing, the resonance will appear different. In our technique, peaks will occur in different places if the geometry is changed, but our technique is also independent of the way the bearing is held to a great extent because we don't depend on the structural resonance. As you pointed out, you can damp out the first few modes of ring resonance just the way you hold it.

K. Smith: It's a little difficult for all of us to point out the differences because we are all looking at the same basic technique, the same basic event. The fact is that if a ball rolling over a defect generates a transient vibration which is then propagated outside the structure and which, at the same time, excites various structural resonances, there are a number of features that can be looked at and different ways of looking at them. I think that the pertinent fact is that there are various levels of sophistication to which you can go. If you simply want the lowest level of detection, then you can monitor a spall by looking at the peaks of energy that are indeed present in the signal. If you want to know where the defect is located, then you use the knowledge about the impact rates, and you can use the normalization procedure that has been discussed with averaging. You can use something like a tracking filter to zero in on the particular rates that you want to look for. If you increase the problem by having a much lower signal to noise ratio, you'll have to go even deeper into the signal in order to pull it out. I think the differences among the techniques attempt to resolve various complicating factors involved in looking at a particular bearing system. It all depends on how you wish to handle those complicating factors.

Dwayne Fry, Oak Ridge Nat. Lab: Our group at Oak Ridge has been involved for several years in analysis of noise signals from reactors to try to detect incipient failure or component degradation in the system. Has anybody on the panel or in the audience had experience in the application of continuous monitoring of these techniques and if so, how has it helped you in the diagnosis of incipient failures in your systems?

J. Anderson: I am not going to answer your question, but there is someone else within Federal Scientific who is experienced in this area, and there is an application note available from Federal.

R. Burchill: Since we have gotton into ball bearing activities, we have been using these very high frequency response sensors more often to look at varying kinds of machinery. We have been able to look at resonant responses of other structures, using similar demodulation techniques to the ones we've applied to the bearing. In geared systems, we have seen resonant modes of gears where the gears are being impacted and they ring like a bell. The repetition rate on this ringing frequency can often be related to the forcing function, and very often to a fixed structural member that is vibrating. By using this modulation component, one can often lead back to the source.

Dwayne Fry: I am wondering if anybody has installed any of these sensing devices on actual operating machinery and used them to diagnose failures on line. Most of what I've seen here this morning have been, I believe, laboratory type applications and on known failures. I am wondering if you can detect these in the presence of the normal background noise in an operating system?

K. Smith: We've been monitoring bearings in jet engines for some time. At Randolph Air Force Base, bearing monitoring of J-85 engines is being done in test cells, but it's an internally monitored sensor so it's always there. During each test period the defects are monitored. In one case there was a defect detected on a #2 main bearing ball in the presence of the operating noise of the jet engine and all the rest of the noise from the test cell. It was identifiable as a ball defect from the periodicity of the data that was taken. There are other applications being made within General Electric's Jet Engine Dept. There are a number of places where it is being used continuously.

William Glew, Naval Eng. Test Establishment: I think the point is well taken that these are apparently laboratory tools that we have been talking about. The GE system is very expensive. Our experience with the real-time analysis system on the outsides of typical machines is that it's not too good at telling bearing faults inside the machines and any Pratt & Whitney guy around here will say that they're no good on jet engines. However, we have recently had quite a bit of luck on simple machines with an ultrasonic analyzer and this is why I was very interested in Mr. Burchill's presentation. We have a quite old Hewlett Packard analyzer and Dave Watson, who is with me here today, started trying to tell us 5 years ago that he could tell when a bearing was bad one minute after it was installed. We recently carried out a comparitive check between this instrument and the fairly new SKF shock monitor and found that

Dave with his analyzer can tell a defect very much earlier then the SKF shock monitor. This is basically because until you actually have got a defect on your bearing, your shock monitor, or your transient pulse indicator, will not tell you that anything is going to go wrong. The ultrasonic analyzer appears to operate in the 34-36 kilohertz range. By the loudness in the audio frequency produced from these ultrasonic waves, you can tell whether a bearing is misaligned or whether there is any undue pressure on the inner race or the outer race right from the word go. This is the only practical application I've seen of bearing monitoring. Most of the systems are very much too complex and too sophisticated and require a very high standard of interpretation. But there is this ultrasonic monitor which I would like the panel to comment on.

R. Burchill: I have some experience with this ultrasonic monitor. In the program we ran, we were evaluating that instrument because it has been used in several cases and the manufacturer is very cautious about his claims for it. I believe the range is from 36-44 kilohertz and that data is folded back from 0-4 kilohertz. With sensors that can respond in this range, we didn't find exactly what was happening. I think that the overall levels in this high frequency region do tend to go up with the defect, and the unit does respond to it. We were not particularly enthused about linearity and the skill required to operate it.

K. Smith: I would like to make just one comment since my name was taken in vain. The GE system, the basic system of crest factor monitoring as I have tried to point out, is very simple and consequently very inexpensive. For the spectrum analyzers or ultrasonic translators, we're talking in the range of 100's of dollars, not 1000's of dollars. That system will tell you when you have an initial spalling if you have a sensor located very close to the bearing itself.

R. Whittier: There are a lot of different applications. I just wonder if one of the differences between these techniques relates to the fineness of the resolution of the detection. Some people are not interested in detecting minor faults such as a little etching or a little spalling, but they are more interested in knowing whether the balls are still in there. A very high frequency detection or sensing system tends to pick out the smaller items, the small irregularities or imperfections. A question for these other fellows relating to their technique is how do they resolve bearing faults from other faults in a machine, such as a blade problem or a foreign object damage.

J. Anderson: I would like to defend realtime analysis and echo what Bob Whittier just said. All these techniques have advantages and disadvantages. You have to look at the application involved to see what is best for you. To use realtime analysis in a bearing failure application, I'm going to repeat what I said at a meeting of this society a year and a half ago. In the initial application of real-time analysis, everyone expected too much. They went into the system expecting to solve all problems. The comment I made a year and a half ago was, we had to walk before we ran. No one took it upon himself to take a discrete component and to see whether the technique was applicable to that component before we went into a complete machine and all the inherent problems in many forcing functions. Our paper is showing you an application of realtime analysis where it was good for that application. It was a discrete component where we isolated all the other vibrations normally associated with a complex machine.

K. Smith: I want to fully concur with that. There are a number of demonstrations that we have seen in our work where realtime spectrum analyses have given a lot of information about the condition of a bearing immediately off the assembly line, whether there is any built-in out of roundness or any built-in eccentricities or ball size defects. All these things show up as spectral components, and indeed it is possible and was done on a program for the Mariner Space Craft evaluating long life rate gyros. It is possible to rank a group of gyros in terms of how many of these built-in defects are present, and thus perhaps make some evaluation as to which of the group is the better bearing. We are talking about different applications, different monitoring goals and it's a question of how early in the game you want to know about it, how good you want the bearing to be when you finally pull it out and do something about it. There are a lot of questions that I think only the user can give an answer to. It's not really our place to answer these questions for you. We are presenting techniques that work in a variety of situations, and you are here to shop amongst them.

Paul Howard: There is one thing that I hope all this brings out to you, especially the first time users, before you get hooked on diagnostics. If there was only one way to go there would be one rich guy and nobody else would be in the business. The thing that makes this a horse race is that there are a lot of different opinions and every one of these guys can point to a success. So you know their technique works on something, and it's very important to consider how close your application is to that something. I would like to state that bearings do not come off the line with defects. It's the users who smash them up on installation.

A. P. Brackney, Monsanto: We are believers that bearings are great when they come out of the factory. But in addition, we might have a pump and a reducer and a motor. What are the individual speakers' plans to help us out in the trenches where the problems are? I think realtime analysis is very close.

Paul Howard: I would like to give a short overview in answer to that and then turn it over to the panel. I think one of the problems that everyone of us who is in this business has faced is the inability, usually due to lack of data, to get a good description of the machine and of the problem that you are trying to solve. You go in and try your particular approach. If it works, you are a hero and if it doesn't work for lack of data or any reason, you're shot down.

R. Burchill: In general, the kind of work that I do is diagnostic work in machinery. It seems very often that you come up to a machine looking for a problem that might be apparent from vibration data one can measure. Or the operator might say, "Oh by the way, we ran a little water into it one day and this is what happened to it". This is part of the problem. It's often a very complex situation. I think the realtime spectrum analysis is an extremely valuable tool in getting at some of these particular facets. If you can identify a cause and effect, then often you can use less expensive, simpler equipment to monitor that particular situation. If you can use the analyzer to determine what kind of a monitor that unit needs, you've won.

Alan Duguid, Huntington Industries: We think in terms of the bearing in any system. It is the weak link in the whole system and any of the idiosyncracies that exist within the system are going to be concentrated in the bearing. My interpretation of this morning's discussion is that basically you are talking about fatigue failures in a rolling element bearing. Fatigue failures account for only a very, very small portion of the failures. My method for checking bearing operation is very simple, and that's to use a heat sensor very close to the bearing. When you start up a new system you will find that the heat output of the bearing will increase very rapidly, and then it will flatten out and it will stay that way. If it doesn't, it will increase very rapidly and I would say anywhere in the period of 15 minutes to an hour, you are going to know whether it is going to stabilize. I have seen some systems where the heat generated within the bearing has been so rapid from the time you push the button to start it till the time you could react to turn it off, the bearing was completely shot. I have seen needle bearings that have generated sufficient heat that they actually fused together and it was difficult to determine the difference from one roller to another. Sensing the heat that is generated in a bearing is going to create the means by which you can check it for a lot of different reasons. The user specifically is not interested per se whether a ball has a pit in it, or the race has a pit in it, or what the failure is. All he's interested in is that there is the possibility of a failure. The failure in a rolling element bearing, whether it be a ball or a roller, can be dependant upon just the intermittant skidding of the rolling element which is a heat generator. I also found that a very excellent tool to use is the torque that's required to drive the system. This is not going to react as fast as the heat is, but it will react. One of the major causes of failure of rolling element bearings is the failure of the retainer. Maybe it's possible to instrument the retainer, but I personally have not found a way. If anybody has, I would be interested to hear about it. This is my first experience where I have been to a meeting of an organization that really could be the voice of the user. One of the basic problems is the fact that the bearing manufacturer produces a product that consists of inner and outer races, balls, rollers and the retainer. When you the user put that bearing in your equipment, that's your responsibility. I don't care what the bearing manufacturer has done to the bearing, you can ruin it. And the problem is that most of the time the user has a bearing failure and he blames the bearing manufacturer. I was affiliated with a bearing manufacturer out in the wilds of Michigan and they are very peculiar people. I have seen bearings that had chips and dirt in them and I swear the only way they could have gotton in there was for somebody to put them in there purposely. If I can say anything to leave you with one basic thought about the use of rolling element bearings, it is that each one of you is going to have a separate problem and there isn't going to be anyone else who is going to answer it for you but you. Engineering is an art and you only learn it by doing it.

Paul Howard: I do have to give you a little bit of flak on temperature because we've run a lot of grease tests for people and there isn't any really good way to tell when a bearing is gone. There are a lot of good ways to tell when the grease is gone and we have determined this by temperature measurements. We can tell a grease manufacturer whether grease A or grease B is better under a given set of conditions, and we do use the rate of increase of the temperature and change in the rate of increase (the second derivative).

Alan Duguid: I have seen cases where the oil was running at temperatures around 150° and the bearing burned up.

Paul Howard: The oil must not have been in the bearing.

Alan Duguid: That's right. Just because lubricant is alongside the bearing doesn't mean that it's in the bearing.

J. Anderson: The torque approach is just another way of sensing, and when you are talking about the torque required to turn a bearing, you are talking about driving that bearing off line, not on line. So immediately you are faced with the same techniques. The use of torque is just another sensing technique. Dave Seneca from GE, Valley Forge, has a paper that was published in Nature Magazine which was specifically on that technique. But after the torque is sensed by strain gages it is spectrum analyzed. It turns out that the overall torque level is not an indication of incipient failure. But once spectrum analyzed, lower orders down as much as 10, 20, 30 dB below overall were an indication of incipient failure.

Jim Reis, Northrop Corp: I would like to ask a couple of questions pertaining to the ultrasonic device and the crest factor measurement. Has any study been done by you people to find out how these measurements relate to the actual remaining useful life of the component, that is, using these techniques in a prognostic sense as opposed to a diagnostic sense? Also do you have any experience in separating or do you know the effects of gear meshes in obscuring the techniques from working properly?

K. Smith: There is some indication, not yet fully confirmed, but I think that there is every reason to believe that it will bear fruit, that the impact level will change as a function of the life of the bearing once a spall has developed, growing with the spall up to a certain level where the spall is covering a significant portion of the race or the ball. At this point you begin to increase the overall vibration level of the bearing significantly. Thus, the crest factor, after reaching some peak level, will tend to level off and then start to be driven down as a result of this overall vibration level occuring. So the thought is that by monitoring this bearing over a period of time, after the initial indication of a fatigue spall, it may be possible to extrapolate something about the projected failure time by looking at the shape of this curve and the rate of change of the crest factor.

Jim Reis: Currently you've made no such measurements, is that correct?

K. Smith: Very little.

J. Reis: Is the rate of change when it's failing an exponential function, or is it a linear function?

K. Smith: I really couldn't tell you at this point. I think it's probably an exponential function but we don't have enough information to tell you.

Paul Howard: There are a number of techniques that keep track of damage growth. That's one that does. Shock pulse does, and in the shock pulse case it's usually linear.

C. Beachem, NRL: What is the size of the smallest bearings which you have listened to or analyzed by your acoustic emission listening devices? Is there some small or minimum size below which it is not worthwhile doing?

K. Smith: The smallest bearing we have looked at is a bearing that is in an x-ray tube and is about 3/4 of an inch in diameter. The largest was a 40 foot diameter antenna drive bearing that the technique worked equally well on.

R. Burchill: At MTI, our program has been a little shorter term than at GE. So far we have looked at bores from 15 mm to 55 mm. This is general gear box and engine range.

Bruce Baird, Boeing Co.: In answer to a previous question we have had a system installed at a chemical plant in Baton, Texas since last March. It operates 24 hours a day and we detected one failure about 8 days before there were any other signs of failure. We use a system that is similar to the systems being shown here. I have a question for Mr. Burchill. I found that I can put a spall into a bearing or a flaw in a bearing and detect the peak signals out anywhere from very low frequencies to very high frequencies, and not just at the resonances of the outer race. I've run them from down around 20 kilohertz up to the megahertz range. I feel that it's mainly the resonances of the transducer that I'll pick up rather than the resonances of the races or the balls.

R. Burchill: The transducers that we have been using have advertised mounted resonances of at least 80 kilohertz and above. The calibration data that I have shows that it's flat through the region in which I am operating, so I haven't been concerned. I've felt that I have been far enough away from that resonance that it hasn't been a problem. By picking a particular resonance that we have demonstrated to be quite independent of load, our system becomes more selective, and it's true that there are many resonances one can use, and they all have this same pulse character. But they have other facets that control their amplitude, and if you are going to make any sense of amplitude, not having to normalize the data in some fashion, there are some real bonuses in being selective about what you look at.

Bruce Baird: Transducers are used resonantly at 120 kilohertz, and that is where I made my measurements, and I did duplicate the data that you have shown there with the spherical roller bearing.

R. Burchill: You detected your signal and got that periodic wave shape? You just did an amplitude detection on that signal?

Bruce Baird: Right. Essentially we filtered out the high frequency band and envelope detected it to produce this modulation.

R. Burchill: I did this also, and then you can do a realtime signal analysis to increase your signal detectability. Also, you can detect your high frequency peaks relative to your background friction noise. I think that might tie your system in with the crest factor approach.

Bruce Baird: I think we've been frightened somewhat of using a system that depended upon the resonant characteristics of the sensor because of the problems that have been indicated in the range of the resonance. One may have a very pure and very large amplification, or you may have other distortions in that region. We have stayed away if we could from the sensor resonance itself.

R. Burchill: We haven't had any trouble. We band limit right around the resonance of the transducer, but we are using transducers that are usually used with acoustic emission techniques. That is really the earliest sign of detecting a bearing failure because the bearing failure starts beneath the surface and propagates to the surface as Mr. Whittier mentioned. It generates acoustic emissions as it propagates to the surface, and these can be picked up using higher frequency techniques instead of lower frequency techniques.

L. F. Sturgeon, GE: I want to respond to the question on bearing size. We are using techniques which are related to all the ones which were described this morning. Our primary emphasis has been on the spectral analysis approach. The application has been primarily to small bearings, for the reason that the smaller the bearing, the larger the effects of mounting and internal geometric variations will be on the loading. We are primarily interested in predicting failure down stream. We were doing this on our 165 bearings and we are routinely using this on bearings of the R2, R4 size.

David Board, Boeing: You discuss the advantages and problems associated with applying the crest factor analysis techniques to the high frequency acoustic emissions of the short duration and rapidly attenuated type that are generally associated with initial bearing defects. Could you also discuss the cost aspects of implementing this in realtime, in a real world.

K. Smith: The basic technique of crest factor measurements can be used on any kind of a signal. If short duration high energy pulses are the indication of failure, then in theory the crest factor technique can be applied to those kinds of signals. It will indicate the presence of those kinds of signals in otherwise random or background noise types of situations. The limitations are primarily those of instrumentation of a sensor. The problems in getting a sensor that is useable over the range which you want with the required sensitivity are (1) getting the sensor mounted in such a fashion that you have good response over that range, (2) obtaining reliable signals at those higher frequencies and, (3) having electronic circuitry that can handle the higher frequencies.

Robert Boole, General Radio: I would be interested in any comments on the band width that users have found to be desirable, either the band width of a tracking filter or the effect of band width of a realtime analyzer.

J. Anderson: Once again this comes in the area of applications. In the case of bearings, when you go through the formula there are approximately 8 to 10 frequencies in the linear domain that all appear within the first 10 orders. Now, depending on the geometry of the bearing, some of these may be very close to each other. This predicates what band width is required. Normally you try not to have too close or too narrow a band width. It becomes more critical as far as speed stability.

Robert Boole: I am assuming a constant band width is required. I was just wondering about some comments on how narrow they have to be for certain applications?

R. Burchill: I think there are some real problems with a very narrow band analysis technique, in that if you have data that is of the impulse nature, the energy involved is very, very low, and so one can be fooled. If you have some data that has a high crest factor, a spectrum analysis of that information does not give you anything near the amplitude that you would measure from an oscilloscope. One need be cautious because you can get in trouble even with the rapid data production capabilities of realtime analysis.

J. Anderson: The wider you make the band width, the less benefit you get from signal annoyance reduction.

Peter Kamber, Boeing: The fact that three Boeing people have so far spoken up here should indicate that we have a very profound interest in this business. My interest specifically is in seeing equipment coming forth that can be put on airplanes and used to do realtime analysis. I have a feeling that we cannot do this without totally miniaturizing the normalizing equipment that has been presented here. One of the panel members has mentioned the magic words, the concept of tracking filters. Does anybody have applicable experience in electronic tracking filters, specifically the types that use miniaturized phase lock loops?

D. H. Biggs, Bently Nevada Corp: As far as the tracking type phase lock loops in general are concerned, there are two types, the digital and the analog. The digital has the advantage of being limited to a very small range which you can track. The analog has the disadvantage of a slow rate capability for the time it takes to get where it wants to go. The kind of system you deal with depends upon how much time you have to take the data.

Paul Howard: Will each panel member give a short summation of what has been discussed.

J. Anderson: What I have learned from this panel is that there are more similarities than dissimilarities among our systems and basically we all have the same problems. Our technique was one of looking for a system to give early detection. It is a screening type system. In spite of the fact that it is on line analysis, the component is tested off line. Taking this to an on line test of a large, complex device, still has to be proven. The problem is one of economics. It is not economically feasible for a supplier who is selling a $200 accelerometer or a $5000-$10,000 instrument to embark on a long term program to prove the capability in complex machinery that would involve possibly a $100,000 to $200,000 study project. I have tried and the other speakers have tried to outline techniques. It is up to you to look at the techniques and the specific application. Listen to the various advantages and disadvantages and judge for yourself what technique is most applicable for your needs. I have found no magic panacea enabling you to sell a piece of equipment that goes on line without embarking on some sort of a preliminary program to study the approach.

R. Burchill: We've taken some very advanced accelerometer designs that have been working adequately in frequency ranges that have not been ignored, but let's say considered only in the noise spectrum, and found that there are indeed some rational components that can be used to define the performance of a bearing. By being selective in this frequency location, we've been able to retain the capability of evaluating minor degradation due to wear or to lubricant problems or dirt ingestion. It also gives a progressive increase as the bearing deteriorates. They can be applied to simple machines as a quality control system, or to complex installations with multiple bearings continuously or periodically monitored.

K. Smith: I can only echo what's been said already. There are techniques available at varying levels of sophistication, but there's no one technique that's going to solve everyone's problem. It's incumbent upon anyone who wants to use these techniques to evaluate his diagnostic goals and then through direct contact or through meetings like this, to communicate those goals to the people who are doing the work developing these techniques. In that way usable systems may be provided that will satisfy your requirements for diagnostics monitoring.

A. Babkin: Basically as pointed out, there are different levels of sophistication. In certain bearings you cannot do any maintenance. In certain types of bearings you can replace components. It also depends on the economics of a system. Our system is applicable if the bearing cost is above $80 a piece. Below $80 the system will not be economical.

R. Whittier: From a sensor standpoint, there are many ways of signal enhancement that have been discussed. I see applications from low frequency to high frequency. In the lower frequency areas, up to 20 kilohertz, we are talking about measuring the motion in the structure. At frequencies above that, in general I see people actually making acoustic measurements on these structures. A sensor may be resonant in that area, and if the sensor is being used in its resonant mode, there is an inherent limitation, basically because it is now a resonant system more susceptible to changes. I will second Burchill's effort here in that there is a good deal of effort going on in the higher frequency areas in nonresonant systems or in systems with some band width to them.

SESSION II

DIAGNOSTIC SYSTEM TECHNOLOGY

Chairman: Jack L. Frarey
Mechanical Technology, Inc.

PREVENTIVE MAINTENANCE MEASUREMENTS ANALYSIS AND DECISION/ACTION PROCESSES

Charles Jackson, Monsanto
MFPG, Gaithersburg, Maryland - November 8, 1972

This is a summary of a 72 slide program presented to MFPG relating the Petrochemical Industry in terms of measures, decisions, and actions taken to prevent failures on rotating turbomachinery.

While one cannot relate to the entire industry, the rotating equipment is representative. The following types of turbomachinery operate at speeds up to 52,000 rpm, horsepower to 15,000, temperatures from plus 1750 to minus 340 degrees F, with conventional trains in three cases of equipment at 10,000 rpm.

- Steam Turbines
- Centrifugal Compressors - Horizontal and Vertical Split
- Expanders

The maintenance cost for our one plant is one and one-half million dollars on rotating equipment. Machine weights to rotor weights will vary in a ratio sometimes from 20:1 to 175:1. Examples of various problems with rotating machines are recorded including rotor fouling unbalance, rubs, oil whirls, misalignment, blade salting, thrust overloads, loose components, base plate resonance and other sub-harmonic resonances plus many forms of bearing instabilities.

DESIGN PRACTICES:

Many of the practices outlined in the 1973 revisions of API 617 which refer to centrifugal compressors for refinery services plus the soon-to-be issued API 614, seal and lube oil standard, are applied throughout our plants. A review of various practices are listed.

1. Two eddy current proximity probes are mounted 90° apart at or on each bearing observing the relative vibration of the rotor to its bearing. These outputs connect to permanent monitors for two levels of alarm. The first level is important because it warns the operating people that the normal operating level has been increased and that one of two things generally could happen. The first would be a shutdown because the second level may soon be reached, or secondly, the machine may maintain this higher level which would indicate that one of the mechanical technology personnel should be called to investigate the problem for a possible correction with-

out shutting down. A single level alarm really doesn't give you too much choice. If you put the maximum threshold level,then you must shut down. If you did not, you hardly know what to do except call someone and wait, risking serious failure if levels increase quickly.

2. Seismic sensors are used on slower moving equipment and generally equipment with exposed bearing caps. These sensors sense the absolute vibration measure of bearing caps to space.

3. A key phasor probe is installed on driver equipment so that a phase reference might be obtained on each train of equipment, i.e., a compressor will be related to its driver phase as would the driver be related to its own phase. Installation of this phase marker on the driven equipment would not allow a reference when the driver is run on solo test.

4. All vibration instrumentation can be viewed on oscilloscopes, x-y recorders, octave band or fourier type spectrum analyzers, tracking filters, phase meters, and other supporting instrumentation such as a digital tachometer.

5. Thrust displacement of rotors is sensed in both directions by eddy current probes preferably viewing the thrust bearing collar but occasionally viewing the rotor shaft near the collar when it is impossible to install a probe looking through the thrust bearings at the thrust collar as on some barrel type compressors.

6. The oil systems are fairly large consoles using stainless steel piping from the oil filters to the bearings with dual five micron, nominal, full-flow filters in the main supply. One-half micron, nominal,filters are used in the supply to the continuously lubricated coupling drives which, in operation, are effective 8000 G centrifuges. If oil or solids of a density other than turbine oil are present they will separate in the coupling in the form of a sludge which may become a critical problem promoting the sticking of these couplings which will severely affect the smooth performance of rotating machinery.

7. Steam turbines are equipped with vacuum leak-off glands and large condensers along with 10,000 G centrifuging at the oil reservoirs to maintain control on water and dirt accumulations in the oil system.

8. Thrust pad embedded thermocouples, or RTD's, are used on turbine thrust bearings. Generally the centrifugal compressor will use a static pressure gauge in the balance chamber and an oil exit temperature sensing the immediate discharge from the oil control ring of the thrust bearings. At severe situations, we have embedded thermocouples in the thrust pads of the compressor as well.

9. Oil exit temperatures and/or embedded couples are provided on radial bearings and other instrumentation or design, too numerous to list, are provided in the oil circulating system of both turbine and compressor.

INSTRUMENTATION:

The portable instruments used by our department now total $80,000 and are used for surveillance of approximately 100,000 horsepower in turbomachinery. The by-line becomes MACE (Measurement, Analysis, Correction, and Engineering). This equipment was obtained piece by piece over the past 10 years. We tried to obtain equipment that we need and understand so as not to get ahead of ourselves. Each piece of equipment has its place. Often the less expensive equipment may do a particular job better in certain situations. A good example of this is oil whirl which might occur at 43 to 45% of the operating speed, that is, the frequency measurement may be 43 to 45% of the synchronous frequency. A hand-held vibragraph, which is a rather inexpensive piece of equipment, prints out on pressure sensitive tape showing the waveform with a timing mark that occurs every half-second. A four foot trace from this instrument will very clearly show the non-periodic vibration form of an oil whirl frequency.

ALIGNMENT:

Sixty percent of the problems in rotating equipment were attributed to misalignment during the first 5 years of turbomachinery analysis. As a result, original development of techniques in proximity measure from water cooled reference stands started in 1965, later accompanied by company-owned optical alignment tooling, in 1970. Prior to this we had contracted our optical alignment work and there are several good firms performing this service.

BALANCING:

Use of conventional seismic-strobe equipment in field balancing has been used for many years. The current x-y probes with key phase markers allow orbital balancing reported earlier in an ASME paper. The most recent field balance was on a 14,000 horsepower, 5200 lb. steam turbine rotor, operating at 11,000 rpm which was balanced from 2.6 mils peak-to-peak to 1.1 mils peak-to-peak.

SEVERITY:

Severity and action points are the most difficult part of any analysis system. No one criteria is absolute. No one machine operation is identical to another. However, within the framework of rough guidelines and confining the limits to high speed turbomachines, the listed values can be considered conventional threshold values. I feel it safe to say that when these values are reached, strong study and consideration will be given toward shutting down the equipment; trying, of course, to obtain the most information possible in order to make a decision as to the problem. When the no-go limits are reached in many cases the shutdown is committed by automatic systems. Other times it is based on operator judgement or on mechanical technology personnel judgement. One exception is that all machines shut down automatically when the second limit of thrust deflection occurs on the operating rotors. The thrust bearing is the only mechanism separating the rotating elements in the machine from the non-rotating elements and when 20 mils deflection from the original design deflection point has occurred, these machines shut down automatically. The consequences are too great, in our opinion, to await the normal time to make a decent decision. A thrust bearing conventionally fails in about 30 seconds. I have examined over forty failures.

LIMITS

Description	Go	No-Go
Seismic Bearing Cap Measure	0-.2 ips o/p	0.5 ips o/p
Proximity Relative Shaft to Cap Vibration	0-2.2 mils p/p	4.2 mils p/p
Thrust Deflection-Kingsbury Type Thrust Bearings - Measured from Float Zone or Normal Deflection Point	15 mils	20 mils
Misalignment-Center Line Offset per Inch of Coupling Spacer Length in Inches	1/4 mil	1/2 mil
Thrust Bearing Babbitt Temperature - Bronze Backed High Tin Babbitt - Oil Control Chamber or Ring - Pivoted Pads - Turbine Oil Viscosity at 150 ssu @ 100°F and 43 ssu @ 210°F	170-210°F	240°F

Within limits, we have found the guidelines presented by the following persons or companies to be good. We do not necessarily adhere to them sacredly, but the values indicated in these guidelines approximate those used by Monsanto.

- IRD's modified Rathbone chart for vibration severity based on bearing cap readings only.

- Dresser-Clark's severity chart for vibration on their centrifugal compressors as measured by proximity sensors giving relative measure, shaft to bearing.

- Koppers high speed coupling misalignment limits.

- Safe thrust bearing temperature limits, as published by the Centritech Corporation in Houston, Texas.

BIBLIOGRAPHY:

Jackson, Charles - "Successful Shaft Alignment", Hydrocarbon Processing, January, 1969, Vol. 48, #1 (Ref. ASME paper 68-Pet-25, "Shaft Alighment Using Proximity Transducers", by Charles Jackson, September 1968).

Jackson, Charles - "How to Align Barrel-Type Centrifugal Compressors", Hydrocarbon Processing, September 1971, Vol. 50, #9, Pgs. 189-194.

Jackson, Charles - "Monitor Points on Turbomachinery", Twenty-Sixth Annual Symposium on Instrumentation for the Process Industries, Chemical Engineering Department, Texas A & M University Proceedings dated January 20, 1971.

Jackson, Charles - "Using the Orbit (Lissajous) to Balance Rotating Equipment", ASME paper 70, Pet-30, September 1970.

Jackson, Charles - "An Experience With Thrust Bearing Failures", Hydrocarbon Processing, January 1970, Pgs. 107-110.

Jackson, Charles - "Vibration Measurement on Turbomachinery", Chemical Engineering Progress, Vol. 68, #3, March 1972, Pgs. 60-65 (AIChE CEP Technical Manual, 1972, Vol. 14.).

Dresser Industries - Clark Turbo Compressor Division, Olean, New York. "General Guidelines for Vibration on Clark Centrifugal Compressors", (chart on last page).

Koppers, Inc., Baltimore, Maryland, "Instruction Sheet 1900-59", Fast's Self-Aligning Coupling Recommended Limits of Misalignment for Fast Couplings.

Herbage, Bernard S. - "High Speed Journal and Thrust Bearing Design", Centritech Corporation, presented at the first Turbomachinery Symposium, Texas A & M University, October 1972, Proceedings, Pgs. 56-61, Figure 8, Page 61.

Sohre, John S. - "Turbomachinery Anaysis and Protection", First Annual Turbomachinery Symposium, Texas A & M University, October 1972 Proceedings.

DISCUSSION

David Biggs, Bently Nevada Corp.: Can you give at least one example of putting together information on deciding when to shut down a machine? You gave us real quickly two levels. I would like a more specific example with a few more details.

Charles Jackson: That's difficult to answer. It's kind of like a domino game. It's like the fellow with the still. When he loses his mash, the whole unit goes down. On some of this turbo machinery, we consider 2.2 as a warning level and at 4.2 we bring it down. We have two different types of people to address this situation or problem to: one is the guy operating the unit who is out there on the No.3 shift and he doesn't really know what to do. The level all of a sudden hits 4. We try to pick that level on a basis of no matter what's happening it's too high, and he must bring the unit down, if the system itself checks out. There are some instrument checks. If the detecting device, is malfunctioning or not functioning, then of course he would not make a decision to bring the unit off. If he can check another point on the same bearing and it agrees with the first point, the system is valid. He is doing the redundant checking very fast, in a matter of a few minutes. There are other inputs such as increasing temperatures in the bearing, or changes in load that can be checked before actually shutting a unit down. But if the system is actually functional, we want that man to bring the unit off if it has reached the shut down level. The alarm level is about twice the level for normal operating conditions. We accept a unit on test standards on less than one mil peak to peak at trip speed which is 15% over rated speed. If the detection system is functioning properly and the alarm comes on, we start our analysis. We may decide to bring a unit down at the warning level if the system is erratic and only getting worse. We like to bring a unit down safely. We may want to abort a pump in favor of the boiler. Let the pump fail, for God's sake, save the boiler. We want to drive as long as we can until we get a spare unit on. It now becomes the "sacrificial anode". We like some redundancy. That's why our radial units are running more like 40% automatic shut down, 60% not automatic. But on thrust, we have found you can't play that game. If it moves, you have to come down with it. We'll then make a decision on it.

Alan Duguid: How many man years went into the development of this system?

Charles Jackson: It's taken 10 years and we have tried to come up from vibrometers to handhelds to proximities, accelerometers. We do not have a realtime analyzer. We are being criticized every day for it and I think they are right. The first accelerometer I used, as an example, we put on a steam turbine that had a 48 KHz resonant frequency. The turbine was running maybe 1/2 G. But all of a sudden, it went to 10 G's. The accelerometer was in resonance with the blade pass frequency.

THE ARMY'S AVIATION DIAGNOSTIC TECHNOLOGY EFFORTS

G. William Hogg

Eustis Directorate
US Army Air Mobility Research and Development Laboratory
Fort Eustis, Virginia

A review is made of the Army's aviation diagnostic exploratory development programs. Current efforts being pursued by the Eustis Directorate, USAAMRDL, are discussed and the areas of diagnostic research which should be pursued in the future are cited. The discussion includes analysis techniques, sensor technology, oil monitoring, and prognostics.

THE ARMY'S AVIATION DIAGNOSTIC TECHNOLOGY EFFORTS

INTRODUCTION:

Records show that Army operational helicopters experience a downtime of 25% to 30%, downtime being the ratio of out-of-service time to total available time. About one-fourth of this downtime is caused by supply problems, and about three-fourths is in waiting for maintenance actions. These large periods of aircraft unavailability obviously have a serious impact on military operations, from both an effectiveness and a cost standpoint. The maintenance performed on the Army's helicopter fleet is of such a magnitude that the average annual helicopter maintenance cost is 40% of the original purchase price of the aircraft. Included in this cost are overhaul costs, which are substantial. As an example, it costs $3,000 to overhaul a UH-1 transmission and $10,000 to overhaul each of the CH-47 transmissions.

Although there are a large number of factors which come into play in these high maintenance costs, there are a few that seem to stand out as large and identifiable items. Inaccurate and untimely fault identification processes are a large contributor.

Another identifiable item is what might be called "the supply system problem". Inasmuch as no advance warning of a malfunction is received, no advance notice is given to the supply system that a replacement part is needed, so one of two conditions must exist -- either a large inventory of replacement parts must be maintained at maintenance sites, or long delays will be experienced in the acquisition of replacement parts. At the present time, the former situation is attempted, but the latter is often experienced.

The third factor which appears to stand out is the overhauling of components on a time basis only. This is not to condemn the time-based overhaul procedure, because in the absence of other information upon which to make a determination relative to overhaul, time-basing that decision appears reasonable. That fact notwithstanding, a large number of components are removed from helicopters for overhaul unnecessarily. A recent survey of transmission removal data showed that of a total of 251 transmissions removed for overhaul, 198 were removed for no reason other than the fact that they had reached their established time for removal. It is reasonable to expect that some of these 198 transmissions did have distressed components in them but there was no indication of deterioration sufficient to cause them to be removed.

The ability to diagnose the mechanical conditions of aircraft components and establish a prognosis of that condition will have a great effect on the cost and usage factors. Accurate and timely fault identification determinations will eliminate most of the uncertainties and the large number of unwarranted component removals. Full use of diagnostic and prognostic equipment will allow the removal of components on a condition basis, as opposed to the currently used time-based removal basis. The use of equipment having these capabilities will lead to more valid prognosis and will greatly facilitate the logistics problem of having spare parts on hand when needed.

The universal use of diagnostic and prognostic equipment will have a most significant effect on virtually every aspect of aviation. The extension, and possible elimination of, time-based removals of aircraft components by the concept of removal for cause has pervasive implications. The air crew will continually be aware of the mechanical condition of their aircraft. Maintenance personnel will have a substantial portion of their unnecessary workloads reduced or eliminated. These factors will have considerable impact by reducing the required replacement parts inventories and other associated logistics problems. The advanced information on forthcoming component removals and required replacement will facilitate having the right parts at the right place at the right time. Knowledge of forthcoming maintenance requirements will also allow higher aircraft effectiveness through scheduling. All of these attributes will contribute to assure mission-readiness, flight safety, and conservation of materials, personnel, and finances.

In addition to the direct payoff of reduced maintenance costs, an additional benefit of a diagnostic system will be the identification of material failure mechanisms. With this knowledge, the designers should be able to design future systems with longer life and better reliability. The development of diagnostic and prognostic capability in the Army will have a major impact on the maintenance concepts which will be employed in the future.

OBJECTIVES:

The objectives of the Army's exploratory development program are to provide an adequate technology base for the advancement of Army aviation diagnostic systems and to investigate new diagnostic concepts and their feasibility for more effectively diagnosing the mechanical condition of Army aircraft components.

This presentation is a review of the efforts currently being undertaken by the Eustis Directorate of the US Army Air Mobility Research and Development Laboratory located at Fort Eustis, Virginia to meet those objectives.

Diagnostic Technology Investigations: Past, current, and planned efforts to advance diagnostic technology can be broken into four broad categories: (1) sensor technology; (2) signal conditioning and analysis technology; (3) prognostic techniques; and, (4) diagnostic techniques.

Sensor Technology: Sensor technology is not sufficiently advanced to provide all of the needed usable diagnostic signals from flight-type hardware. In some cases, acceptably accurate signals can be obtained from existing sensors, but these sensors are generally the more bulky and heavier laboratory type of equipment. Quite often, critical characteristics are sacrificed in the achieving of flight-sized hardware. Efforts are being pursued to provide sensors possessing the necessary characteristics, such as accuracy, repeatability, reliability, stability, light weight, long life, etc., which can meet the needs of diagnostic equipment. In addition to advancing the state-of-the-art of the measurement of parameters which can now be measured, work must be put forth to devise ways of measuring parameters which now cannot be measured; for instance, the important parameter of turbine inlet gas temperature cannot be directly measured at the present time. Present day determinations of the turbine inlet gas temperature are made by inference only.

We currently are involved in sensor efforts on flow, pressure, and torque sensing. Panametrics, of Waltham, Massachusetts, is investigating a concept of using ultrasonics to measure mass fuel flow. It is envisioned that this will result in the accurate measurement of this parameter using a non-intrusive, no moving parts transducer.

In the area of torque sensing, two contracts have been awarded; one to the Garrett Corporation and the other to Avco-Lycoming. Both of these efforts are directed toward improving the accuracy of torque sensors.

A contract has been completed with Hamilton Standard where an effort was conducted to develop a pressure sensor capable of measuring pressures up to 250 psi. The transducer used a vibration cylinder concept for determining pressure.

The thrust of the sensor technology investigations is toward the derivation of technology which will produce meaningful signals which relate to mechanical component condition. These investigations must include not only the advancement or improvement of existing sensor concepts to improve accuracy, reliability, weight, etc., but must also include investigations of new concepts of sensing parameters of interest. An example of such an investigation to be conducted is the application of acoustic emission sensing techniques for the determination of incipient metal failures in the aircraft environment.

Signal Analysis: The heart of a diagnostic system is the analysis technique and the decision-making criteria which are employed. The ability to accurately record a signal related to a component is meaningless if the analysis procedure used and the decision-making criteria upon which this analysis procedure is based are erroneous. Basically, there are two methods of signal analysis now being employed.

One of these, the logic matrix, is generally used where a norm exceedence type of concept is employed with several different parameters being involved. For instance, information related to the exceedence of some preestablished norms for hydraulic system components, such as pump pressure, oil temperature, relief valve position, control lever position, and actuator position, could be fed into a matrix to allow inferences relative to the hydraulic system condition. Implicit in the successful use of a technique such as this is the establishment of norm exceedence as an acceptable procedure and the establishment of appropriate norms.

The second type of analysis procedure is that of spectrum analysis. In employing this technique, some characteristic of the signal, as it varies with time or some other domain, is analyzed for its content. This type of analysis technique is mostly applicable to vibratory or acoustic signals, but conceivably it could be applied to such signals as hydraulic pressure vibrations. Work done to date on spectrum analysis investigations has generally been on a fragmented basis, and thus, a continuing coordinated effort to devise logically- and technically-based analysis techniques is planned.

Analytical and experimental investigations are to be conducted in signal conditioning and signal analysis technology, with the intent of deriving techniques of spectrum analysis and pattern recognition suitable for diagnostic use. Analysis techniques which have been used successfully in other fields appear to warrant investigation. Although it is probable that these concepts can be applied to both the vibrations and noise emanating from mechanical components, it has not yet been determined whether they can be used to identify a specific component experiencing failure.

A contract with the Ohio State University Research Foundation has been awarded to assess techniques which are applicable for vibration signal analysis of Army helicopter power train components. It is anticipated that this effort will result in the identification of techniques which show the greatest potential for this type of application.

Prognosis: Prognostic investigations will concentrate on forecasting the progression of a malfunction once detected and the life remaining of the component in question. Since mechanical components do not fail totally without any prior warning, it appears that the prognostic capability is a big pay-off area. The capability of determining the seriousness of a malfunction and its prognosis will have impact on all phases of Army aviation. The entire field of prognosis is so broad and deep, and has been penetrated so slightly that large amounts of work are called for. Investigative efforts must be accomplished to establish the factors, both internal and external to the component, which have effect on the failure progression rates. These factors must be verified experimentally. The investigations conducted must cover the range of the aircraft components' subsystems (e.g. hydraulic system, power train systems, structures, etc.). Failure progression rates and factors associated must be established for items such as bearings, gears, pumps, and mechanical connections.

These two contracts represent the beginning of a long range effort.

Other Diagnostic Investigations: Other diagnostic techniques which do not fit in the above categories are being or will be investigated by the Army. Included in this category are such items as engine power available studies, air density measurement techniques, fatigue crack detection, oil monitoring and analysis, and acoustic emission techniques, which are actually combinations of the previous technologies and would likely serve as subsystems in a diagnostic system. These concepts and others of this type must be pursued to establish feasibility and usefulness in Army aviation.

Identification of Failing Mechanisms Through Vibration Analysis

JAMES L. WOTIPKA **RICHARD E. ZELENSKI**

IBM General Systems Division, Rochester, Minn.

INTRODUCTION

Vibration and acoustic signals have been found to be valuable indicators of the general operating condition of mechanical equipment. Thus, their measurement and analysis is often part of a comprehensive maintenance program. Methods have progressed from subjective judgments based on listening to noise, to the use of microphones, accelerometers, and ultrasonic detectors for signal measurement, and waveform and frequency analyzers for signal analysis. The types of equipment being monitored have also become increasingly complex, e.g., from large gears and bearings to the valves in internal combustion engines. This increased complexity has placed severe demand on failure detection and identification techniques.

Two approaches to failure detection are commonly considered (1).[1] The first is statistical, where many samples of operation are taken to establish a characteristic signal. Components with known defects are used so that specific failures can be categorized. The second approach is analytical in that the possible failure modes are predicted along with their resultant effect on vibration signals.

The analytical approach has gained the widest acceptance. Detection of failures in large rotating equipment has been accomplished by observing increases in vibration levels. For more complicated mechanisms and when monitoring the functional performance of a machine, a primary method of failure detection has been comparison of machine timing charts and signal waveforms. This established the normal signal contribution of individual components and mechanisms during a cycle of operation. Any change in the waveform can then be attributed to a specific component failure based on correlation with the functional timing chart. Unfortunately, many mechanisms and components may be operating simultaneously, making it difficult to identify a specific one. Therefore, a study was initiated to determine if some other method for identifying malfunctioning components could be found. The results of the investigation showed that the natural frequencies of components making up a mechanism were of significant value as failure indicators, particularly on high speed, low inertia equipment. Previous investigators have primarily considered frequencies corresponding to rotational speeds and caused by occurrences such as the balls of a bearing rolling over the pitted surface of a race, or tooth contact frequencies in gears.

The study took the approach that to understand the operation of any machines, it is necessary to know the function of basic mechanisms making up that machine. Also, to gain an insight into the nature of vibrations of a machine, it is necessary to understand the signal contributions of its basic mechanisms and components. To accomplish this understanding, the general vibration characteristics of several basic mechanisms and the effect of their typical failure modes on these vibrations were studied.

A unique digital computer signal analysis system that provides a flexibility needed for thorough analysis of vibration data was used ex-

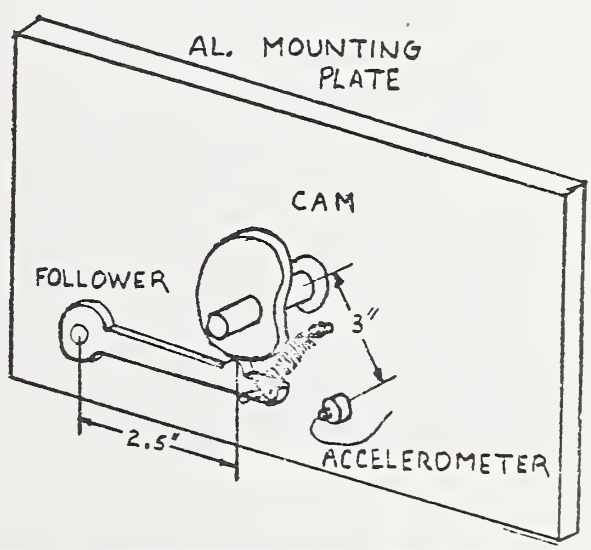


Fig. 1 Spring-loaded radial cam, oscillating radial follower, and accelerometer used as test vehicle in the study of mechanism and component vibration characteristics.

[1] Numbers in parentheses designate References at end of paper.

Fig. 2 Power spectral density plots of cam-and-follower operating at (a) 1000 rpm, (b) 1900 rpm, and (c) 2000 rpm indicating no frequency changes other than in amplitude

Table 1 Natural Frequencies of Components Making up the Cam-and-Follower Test Unit

	Calculated	Experimentally Determined	Observed in PSD Plots
Bearings	2190 Hz		1.9 KHz - 2.7 KHz
Cam		6660 Hz	6500 Hz
Follower		10 KHz - 13 KHz	7.9 KHz - 14 KHz

tensively. The equipment and computer programs are described.

PROCEDURE

The mechanism chosen for initial investigation was a spring-loaded radial cam and oscillating radial follower operating at 1200 rpm. A high frequency (125 Hz natural resonance) accelerometer was mounted on the frame supporting the cam-and-follower to monitor vibrations during operation (Fig. 1).

Because of the investigative nature of the study, we wanted the capability to analyze signals in as many different ways as possible. We felt that digital computer programming would best provide this needed flexibility.

The system used for data collection and analysis included an IBM 1130 digital computer, an EAI Model 580[2] general purpose analog computer, and a real time channel (RTC).

The analog computer was programmed to provide analog-to-digital conversion and signal preprocessing. The conversion provided a six-bit digital representation of an analog input voltage. The signal preprocessing functions included amplification, shifting of d-c level, filtering, amplitude limiting, and rectification.

The real time channel provided data and control interface between the 1130 CPU and the external test hardware. When digitally sampling an analog signal, a number of conditions and variables had to be specified and controlled. The data collection program was required to synchronize the sampling time with the test cycle (machine cycle), control the desired sampling rate, and store data where it could be retrieved and identified. This was accomplished through program control of the real time channel clock, read register, signal gates, cycle counter, and instruction and data address registers.

A number of analysis programs were used to perform the necessary data processing and to provide the appropriate output. Examples were ensemble averaging, fast Fourier transform routines, auto-correlation, and cross-correlation. A signal enhancing technique known as ensemble averaging provides a means for improving signal-to-noise ratio and for waveshape recovery. It operates on a number of signals, each synchronized to the function being observed but individually obscured by noise. This "noise" can be any undesired effect, random or periodic, which is not in synchronization with the period of interest. The in-phase periodic signals are reinforced as a number of signals are added point-by-point. The noise, however, which can be positive or negative, averages toward zero.

A fast Fourier transform (FFT) routine was formatted in a number of ways to allow frequency analysis of digital signals. The first was the discrete Power Spectral Density (PSD) plot. Power amplitude versus frequency was plotted to show the contribution of each frequency to the total power of the signal being analyzed. This could be accomplished for portions of a machine cycle, or an entire cycle. In addition to amplitude, the program also provides phase information for each frequency. Thus, if a number of similar consecutive cycles are analyzed, the result represents the power spectrum of an ensemble averaged signal (random noise removed) when phase information is considered. However, if phase information is neglected, the result is the average magnitude of each frequency including noise.

To investigate the frequency-time relationship for a machine cycle, a technique similar to the sound spectrograph or sonogram (used in speech processing and based on the fast Fourier transform) was employed (2). A method of overprinting on a line printer provides a variable intensity frequency-time-amplitude plot. The result is a display with the frequency and time axes on the ordinate and abscissa, respectively, and the intensity of printing proportional to power amplitude.

[2] Electronic Associates Inc., West Long Branch, N. J.

Fig. 3 Power spectral density plots of one revolution of the cam shaft operating alone (a) and the cam and follower operating together (c) compared to the average plots of many revolutions (b) of the cam shaft alone and (d) of the cam and follower together. The significant retention of signal on averaging seen in (d) was attributed to the action of the cam and follower causing vibrations to occur at the same point in time for each cycle.

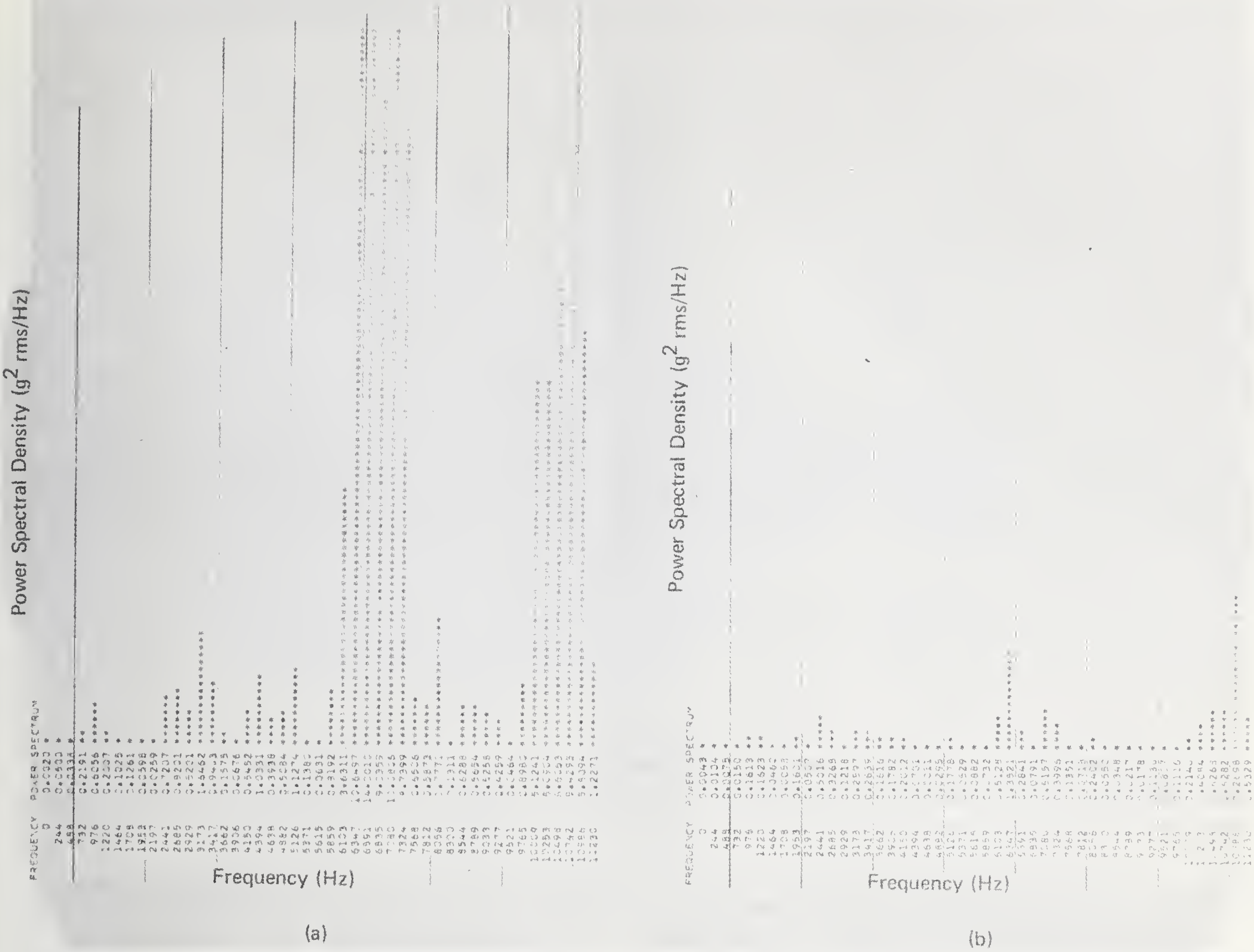


Fig. 4 **Power spectra indicating the higher vibration amplitude of a worn cam follower,** (a), as compared to a new cam follower, (b).

Signals stored in the digital computer were analyzed for similarity using correlation programs. The auto-correlation function (correlation of a signal with itself) was used to find any periodicity within a signal that may have been obscured by noise. A signal is shifted along the time axis and correlated with the unshifted version of the same signal. The correlation product reaches a peak when periodic components are in phase.

The cross-correlation function was used to detect similarity and timing differences of any to waveshapes by shifting one signal, point-by-point, with respect to the other and observing the correlation product. The magnitude of the correlation product and the amount of shift needed to reach a maximum indicates the degree of waveshape similarity between signals and the timing difference with respect to a sync point.

Although signal analyzers capable of these same tasks are commercially available, they generally lack the ability to perform them on single cycles of machine operation or portions of a cycle. This capability was required for investigation of functional cycles of mechanisms and machines.

Initial testing of the cam-and-follower mechanism was designed to provide information on the general nature of the units vibration characteristics. The tests included runs at different rpm's, runs of the cam shaft alone, and runs of both the cam-and-follower. The data from these tests were extensively analyzed using the analysis programs mentioned.

In later tests, typical failures and failure conditions were introduced to determine if

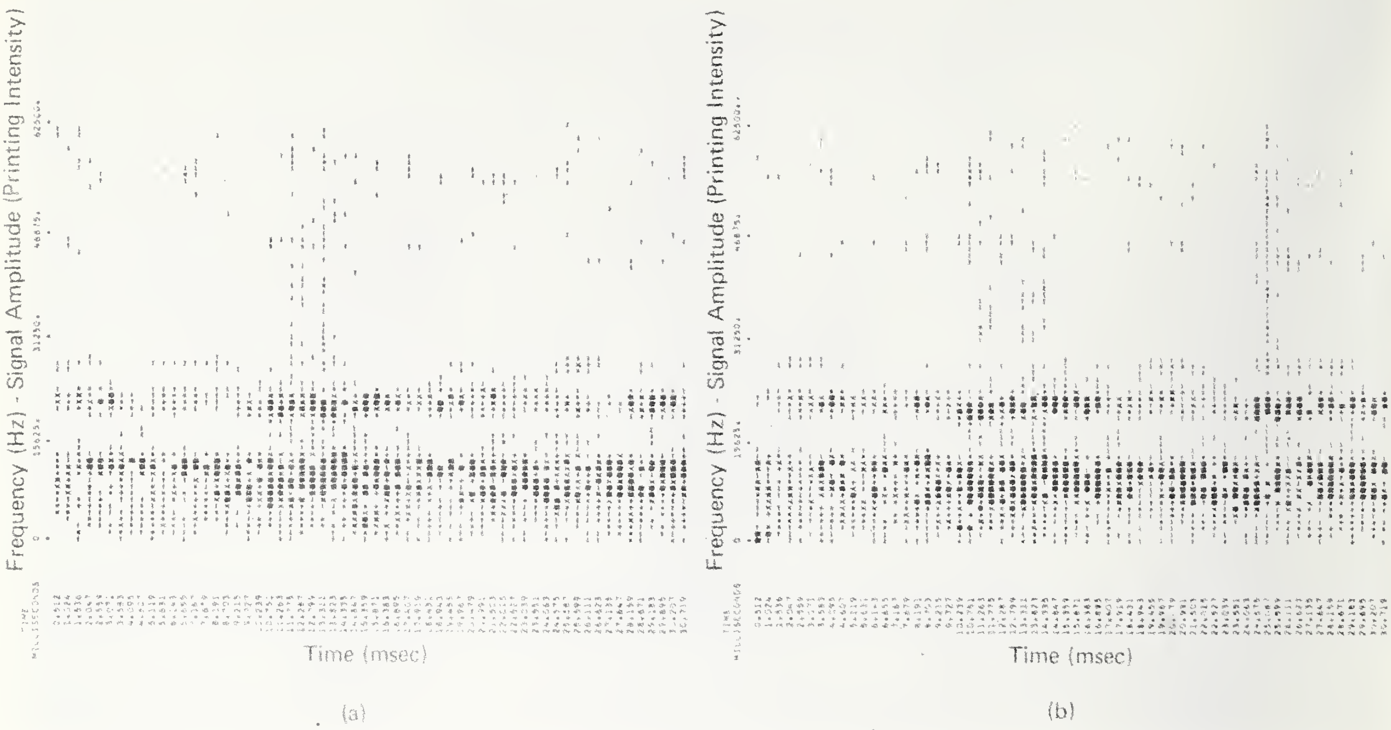


Fig. 5 Spectrographs (frequency - amplitude versus time) showing the insignificant difference in high temperature, (a) and normal temperature, (b) operation on cam-and-follower vibrations. A (1) represents a minimum signal amplitude and (a) represents a maximum.

detectable changes in vibrations would result. In the first of these tests, a field returned cam-and-follower set was compared with a new cam-and-follower under conditions of lubrication and no lubrication.

The second test was run to determine the effect of cam-and-follower run-in and temperature. A new cam-and-follower was run unlubricated for 10 min., with vibration signals collected during the last minute. After a cool down, another minute of signal was collected. This procedure was repeated a number of times on the same cam-and-follower. During the course of this test, fretting corrosion of the unlubricated cam surface developed. The resulting effect on the vibration signals was observed and analyzed.

The final test was run to determine the effect of cam wear resulting in a cam contour change, as opposed to wear causing rough spots. Approximately 0.002 in. of material was removed in two places on the cam surface; at a point of low acceleration and force, and at a point of maximum acceleration and force. The 0.002 in. just exceeded manufacturing tolerances.

RESULTS

The cam-and-follower mechanism was initially run at three different rpm's (1000, 1900, 2000) to determine if any changes would result in the Power Spectral Density plot. Fig. 2 shows the results of these runs. Only the amplitude of the frequency components changed and no new frequencies appeared. This indicated that the frequency content of the signals being detected were independent of operating speeds, i.e., rotational frequencies (such as the balls in bearings) were not contributing significantly to mechanism vibrations.

The test runs at varying rpm's, led to a further investigation of vibration sources. A first step was to determine the natural frequencies of the various components in the cam-and-follower mechanism. Because of their relatively simple configuration, the natural frequencies of bearing races could be calculated (3, 4). The cam-and-follower, however, required experimental methods. The mechanisms were caused to resonate by being struck. Resulting vibrations were de-

Fig. 6 Spectrographs (frequency - amplitude versus time) showing the higher signal strength of a new cam-and-follower, (a) prior to run-in, (b) after run-in. A (l) represents a minimum signal amplitude and (a) represents a maximum.

tected with a microphone and displayed on an oscilloscope. Photographs of the scope traces were studied to determine the frequency content of the signals. The results of the calculations and tests are shown in Table 1. Included in this table are frequency components observed in the Power Spectral Density plots of the cam-and-follower vibrations. A similarity between predicted and observed frequencies is apparent.

In the next series of tests, the cam shaft was run alone and the cam-and-follower operating together was run. The purpose was to determine the effect on vibrations of a periodic occurrence, such as the follower traveling over the cyclic contour of a cam. Fig. 3(a) shows a PSD plot of one revolution of the cam shaft. Fig. 3(b) shows the PSD of a number of cam shaft cycles in which phase information was retained (random signals are lost as in ensemble averaging). The significant loss of power indicated a highly random signal for a cam shaft operating alone. Fig. 3(c) and 3(d) show PSD plots similar to 3(a) and 3(b) for the cam-and-follower operating together. The high retention of power in 3(d) indicated that the amount of randomness from cycle to cycle was lessened. This was attributed to the cam contour causing the cam, follower, and bearings to be excited to their natural frequencies at the same point in time in each cycle. Hence, a greater portion of the signal strength was retained on averaging.

Figs. 4 to 9 and Table 2 show results of the three experiments in which failures were introduced. Fig. 4 and Table 2 illustrate results of an experiment to determine the effect of lubrication on a new and run-in (field-returned) cam-and-follower mechanism. The radial contacting surface of the field-returned follower was worn slightly flat. Therefore, as it traveled through the concave portions of the cam profile, higher vibration levels were recorded, as shown in the power spectra of Fig. 4 and certain significant frequencies of Table 2.

Table 2 shows the confidence levels of certain frequencies which indicated by their amplitude that new and unlubricated produced higher vibration levels than run-in and lubricated cams and followers. All frequencies from a number of PSD plots were statistically analyzed. A significant result was that the frequencies of greatest

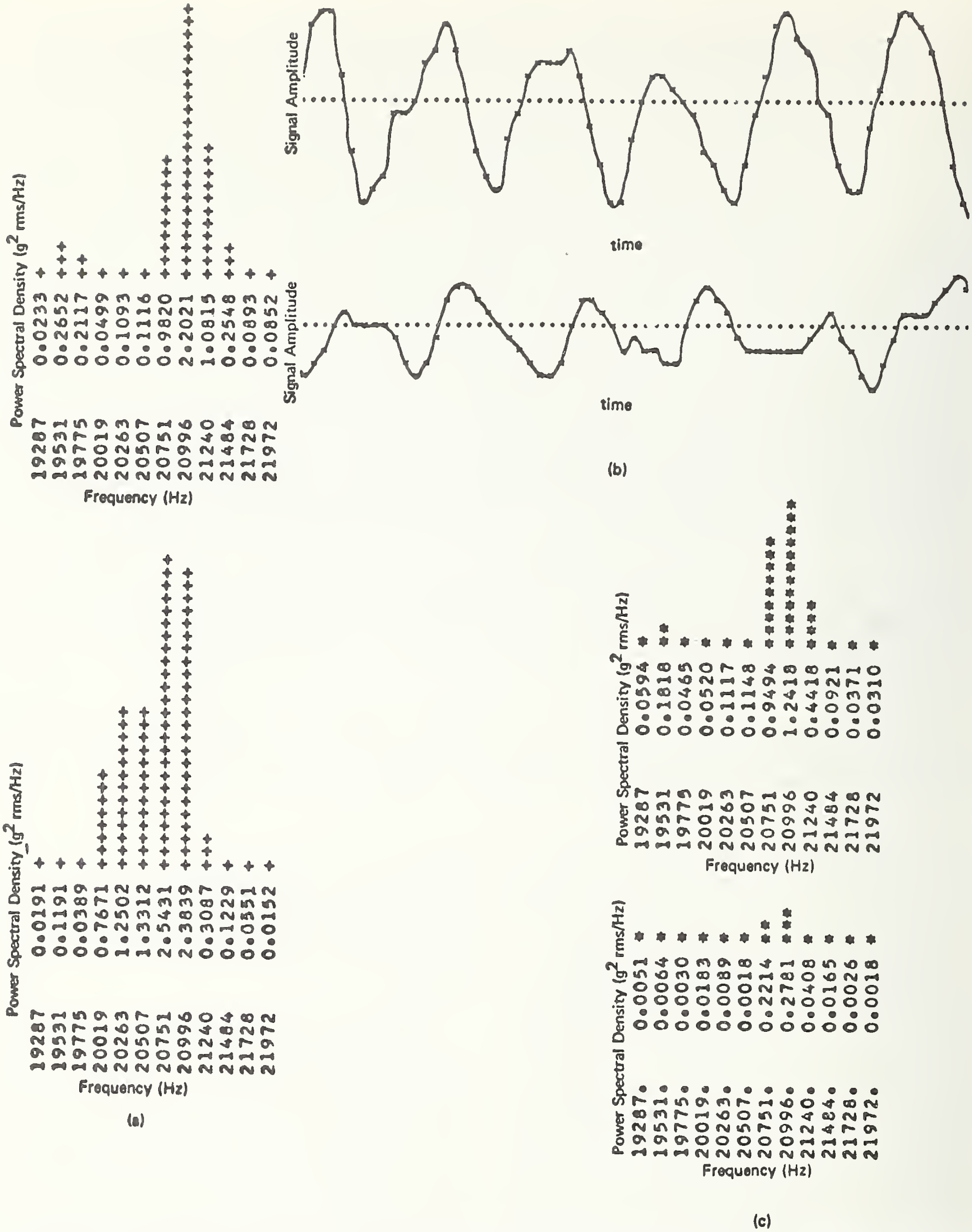


Fig. 7 Portions of average power spectra (phase information discarded) (a), ensemble average waveforms (b), and power spectra (c), of the cam-and-follower before (upper figures) and after fretting corrosion occurred, (lower figures) indicating signal strength lost on averaging due to randomness of vibration for a fretted condition.

confidence were those corresponding to the natural frequencies of the cam-and-follower.

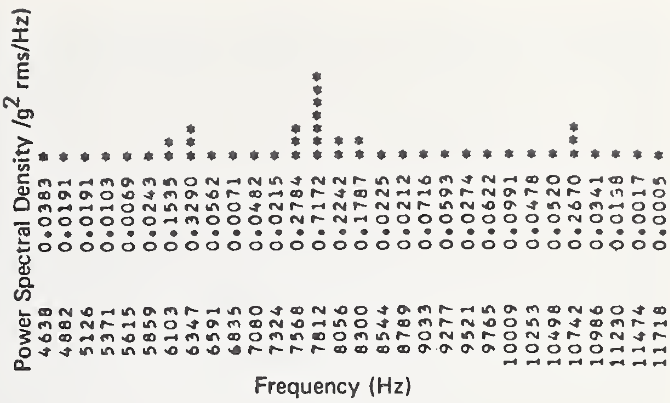
Figs. 5, 6, and 7 show results of the experiment to determine the effect of temperature and run-in. Fig. 5 shows that vibration signals of an unlubricated cam-and-follower were not significantly affected by high temperature conditions.

Fig. 6 compares vibration signals of a cam-and-follower during initial run and a later run, and shows a reduced level of vibration caused by run in.

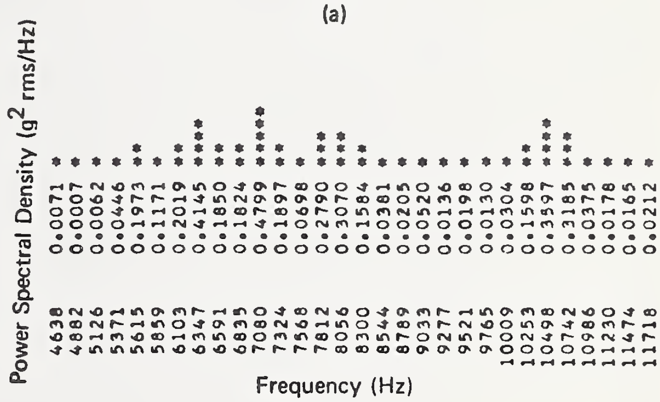
Fig. 7 shows the increased signal amplitude and greater signal randomness that resulted when the cam-and-follower were allowed to run unlubricated until a fretting corrosion condition occurred. In part (a) the average power spectrum (phase neglected) of ten raw cycles shows greater signal strength for the fretted condition. In parts (b) and (c) the ensemble averaged waveform and the power spectra of these waveforms indicate that a great deal of the signal strength for the fretted condition has been lost. This is due to the fact that the signal strength was random because of the fretting corrosion and, therefore, was lost during averaging.

Figs. 8 and 9 show results of the experiment designed to study the effect of a cam contour changed by wear. In Fig. 8, power spectra of wear and no wear conditions are quite similar.

Fig. 9 shows average waveforms and power spectra for wear at a point of higher cam-to-follower load than for the data of Fig. 8. An increase in signal strength is evident.



(a)



(b)

Fig. 8 Portions of power spectra of a cam within manufacturer's tolerances (a) and one worn 0.002 in. out of tolerance (b), at a point of low cam-to-follower force, indicating an insignificant difference between the two conditions.

CONCLUSIONS

The objective of this study was to investigate a means of machine failure detection which did not rely solely on the time consuming correlation of machine functional operation with signal waveform. The general vibration characteristics of elemental mechanisms (e.g., cam-and-follower) were investigated and the effects of certain operating conditions and failures on the vibrations were then determined.

Based on the results of an experimental study of a cam-and-follower mechanism, the following conclusions were drawn.

1 Most frequencies seen in a spectrum analysis of vibration data are natural frequencies independent of mechanism operating speeds.

2 Components, such as cams, followers, and bearings, when operating as a part of some mechanism, are excited to their natural frequency throughout a machine cycle.

3 A periodic mechanical action, such as a follower traveling through a cam contour change, causes the cam-and-follower to be excited at the same points in time for each cycle of operation.

4 Detection and identification of failed components can be accomplished by monitoring their natural frequencies for changes in amplitude.

A number of typical operating conditions and failures of a cam-and-follower mechanism were studied using various techniques of vibration signal analysis. The conditions studied were: lack of lubrication; worn follower; worn cam; irregular cam surface defects (e.g., fretting corrosion), and run-in.

The effects of these failures were predictable and appeared as increases in vibration levels (except run-in which reduced vibrations). However, the nature of the vibrations was not the same in all cases. A lack of lubrication caused increased signal strength throughout a cycle, while a worn cam and a worn follower caused increases only at the points of wear. Fretting corrosion also increased vibration levels throughout a cycle, although the amount of increase varied from one cycle to the next and also within a cycle. It was evident, therefore, that although the effect on vibration levels from various failures may be the same, their effect on the nature of the vibrations may be quite different. This then would allow identification of certain failures in some cases.

The significance of the work done in this study does not lie so much in the results pertaining to the specific cam-and-follower investigated but more importantly in the general type of

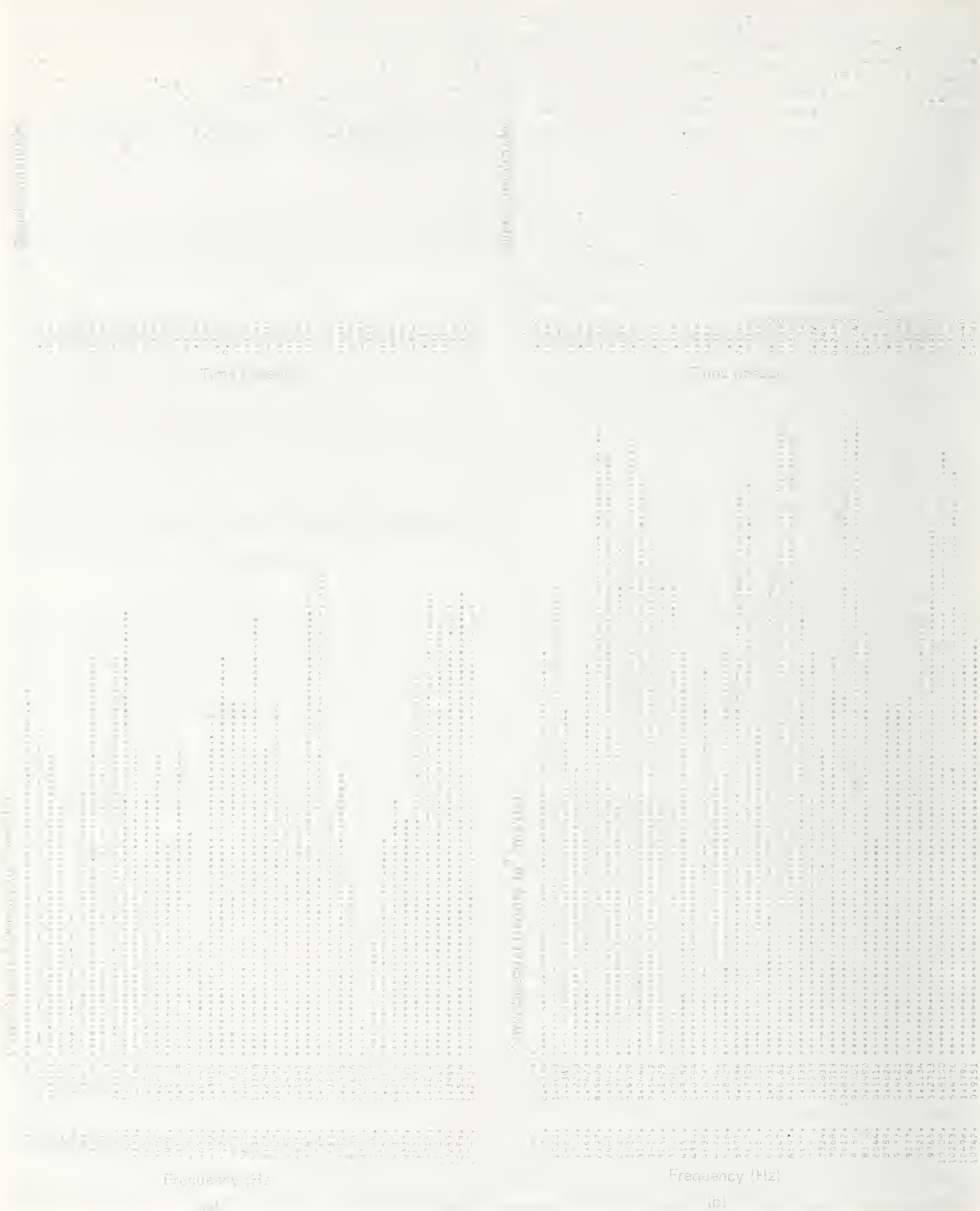


Fig. 9 Portions of average waveforms (upper traces) and power spectra (lower traces) of a cam within manufacturer's tolerances (a), and 0.002 in. out of tolerance (b), at a point of high cam-to-follower load, showing the increased signal strength due to the cam contour change.

Table 2 Confidence Levels of Significant Frequencies Showing that a Run-in Cam-and-Follower Ran Quieter than a New One, and that an Unlubricated Cam-and-Follower Produced Higher Vibration Signals than a Lubricated one.

	Follower at high dwell portion of cam profile			Follower at concave portion of cam profile		
Frequencies	Confidence levels of significance			Confidence levels of significance		
	Run-in cam vs new cam	Lubricated vs not lubricated	Combination of run-in and not lubricated	Run-in cam vs new cam	Lubricated vs not lubricated	Combination of run-in and not lubricated
1220					99	
2441	99	99	99			
2685					75	
2929				75	95	75
3173				75	75	
3417				75	75	
4150					70	
4638				75-90		
5126				75-90		
5615	99				75	
5859	99			75		
6103				75-90	70	
6347	70			70		
6591	90			75-90	70	70
6835	99			75-90	75	75-90
7080	99		99	75-90	75	75-90
7324	75-90				70	70
7568				70		
7812	97.5	90	95			
8789				70		70
9033				90	75-90	90
9277				99	99	99
10498	99				75	
10742	97.5				75	
10986	90	75-90	75	70	90	75
11230	99	99	99	70	75-90	75-90
11718	99	99	99			
11962	75	75	75	99*		
12207				99*	99	99
12451	70			75-90	75-90	75
12695	90	97.5	97.5	70		
12939				70		
13183				99*		
13427				75*		
13671				99*	99	99
14648				99*	99	99
14892				97.5*	95	95
16357				75-90	75-90	75-90
18554					99	
18798					99	
19775				99	99	99
20019					75	
20263					75-90	
21972				99	99	99
22216				99	99	99

* Denotes instances where the run-in cam-and-follower had higher signal strength because of follower wear.

knowledge gained (i.e., Conclusions 1 and 4). This type of information can be applied not only to other cam-and-follower types, but also to other mechanisms. This has been shown in the design of a probe for the detection of bearing failures (5). The natural frequencies of the bearing races are calculated, and then an accelerometer mounted in a spring loaded probe and a filter are used to monitor the levels of these frequencies. The system has proven capable of detecting extremely small race and/or ball damage that could not be found by monitoring the overall vibration level of rotational frequencies.

REFERENCES

1 Tatge, R. B., "Acoustic Techniques for Machinery Diagnostics," Paper presented at the 75th Meeting of the Acoustical Society of America, Ottawa, Ontario, Canada, May 21-24, 1968.

2 Rothauser, E., and Maiwald, D., "A Digitalized Sound Spectrograph Using FFT and Multiprint Techniques (Abstract)," Journal of Acoustic Society of America, Vol. 45, p. 308, 1969. Copies may be requested from IBM Thomas J. Watson Research Center, Post Office Box 218, Yorktown Heights, N. Y. 10598.

3 Harris, C., and Crede, C., "Shock and Vibration Handbook," Vol. 1, McGraw-Hill, New York, 1961, pp. 1-14, 1-15.

4 Love, A. E. H., "Treatise on the Mathematical Theory of Elasticity," Dover Publications, New York, 1944, pp. 451-453.

5 Martin, R. L., "Detection of Ball Bearing Malfunctions," Instrument and Control Systems, December, 1970.

6 Balderston, H. L., "The Detection of Incipient Failure in Bearings," Materials Evaluation, June, 1969, pp. 121-128.

7 Gibbons, J. B., "Mechanical Signature Analysis — Engine Diagnostics," Memorandum — MSA 65-2, Advanced Technology Laboratories, General Electric Company, July 12, 1965.

8 Weichbrodt, B., "Mechanical Signature Analysis, A New Tool for Product Assurance and Early Fault Detection," Report No. 68-C-197, General Electric Research and Development Center, Schenectady, N. Y., June 1968.

NOTE:

This paper was presented at the Vibrations Conference and the International Design Automation Conference, Toronto, Canada, September 8-10, 1971. It was published by the Design Engineering Division of the American Society of Mechanical Engineers (ASME) as Publication 71-Vibr-90.

DISCUSSION

Jack Frarey, Mechanical Technology, Inc.: As I understand it, most of the changes you saw were changes in amplitude and not changes in the resonant frequency content. On spectrograms that you were showing, what was the frequency region on the bottom? What was the highest frequency in which you found resonances?

James Wotipka: The spectrograms that I showed were not really too representative of the data we saw. We were seeing frequencies as high as 20-25 KHz for bearing outer races. These are the higher harmonics of these frequencies. For the particular cam we were studying, the resonant frequency that we had the most success with as a failure indicator was about 6 1/2 KHz and for the follower it was around 13-14 KHz. For the gears we were studying, we were in the range of around 20K and for the geneva incrementer a very good resonant frequency was 13-15K. So it's getting to the upper limit of the audible range, almost to the ultra-sonic range in certain instances. That's where these very small and sensitive accelerometers work as very good transducers.

Paul Craft, Hughes Aircraft: On the single cycle spectrum that you showed and it's apparent not averaging to zero over long periods of time, or at least apparently increasing the signal to noise, what type of frequency analysis were you doing? Single samples or real Fourier analysis is phase sensitive, so that mechanical synchronization is probably mandatory. Random phase fixed frequency is computationally, at least, cancelled out with the imaginary fast Fourier transform. What type of Fourier transform were you using?

James Wotipka: I wish Mr. Zelenski was here to answer that question (the other author). I do know that phase information was available from the fast Fourier transform that we did. The single cycle that you saw is the resulting spectrum from only one revolution of the cam. What we did, many times was to throw away the phase information and work only with amplitudes. Then we could add these to come up with somewhat of an average amplitude value. Now this would get away from the problem of randomness in the signal and give an indication of what you really had as an overall signal strength and still give you a frequency analysis. We also went the other way where we included the phase information. If there was randomness in the signal, if you did any ensemble averaging, for instance, this randomness would tend to filter out the noise and you would only see those parts that were synchronized to the event. So, signal strength was lost.

Paul Craft: I'm looking into a situation where I can not get an external mechanical synchronization, which means, essentially, with the measurement interval that I am taking, my phase information is going to be random. Have you done anything which indicated that that was even possible, that you could get the resonant frequency information without external mechanical synchronization?

James Wotipka: Just take the overall vibration signal. You don't really need a syncronizer to do that. First of all you can take a long record of a signal, it doesn't even have to be a repetitive or cyclic type situation, and do a fast Fourier transform of it. You do not have to have synchronization to get an indication of the signal strength at various frequencies. The comparison I made, before we did a final spectrum analysis when we ran the cam and follower together or when we ran the cam shaft alone for many cycles, was done after we had actually done signal averaging. We introduced the need for synchronization there because that's what we wanted to do. Those first 2 charts of the power spectrum were done on a single cycle, but they could have been done for many cycles and the results would have been essentially the same.

PANEL DISCUSSION

Panel Members: Charles Jackson, W. Hogg, James Wotipka, and David Brown

Panel Moderator: Jack Frarey, Mechanical Technology, Inc.

Paul Howard, SKF Industries, Inc.: What is the effect of wear on the mechanical emissions?

David Brown: The wear on a given device normally doesn't have a large effect upon the mechanical impedance because basically you're looking at the natural frequencies and the mode shapes of the structure. If wear has drastically changed the geometry of the system, it would have an effect on the natural frequency. In general, the stiffness of a given mode of vibration would have to be changed, which means that a cross-sectional area, or something similar to that, would have to be drastically changed by the wear problem itself. So wear normally would not have a big effect upon the mechanical impedance in a given system. Automobile manufacturers would like to use this for quality control, to inspect a frame to make sure all the welds have been made properly. In that case, there could be significant differences in the stiffness of various modes of vibration. But normally the wear would not have a big effect.

Paul Howard: I was thinking mainly about the hysteresis effect wear in bearings and gears would give.

David Brown: Normally in bearings, you do have some sort of damping term which depends on the type of contact in the system. I think that that would be a better indication of the system. The stiffness of a bearing may change due to wear and it is a very non-linear type of stiffness to start with. In that case, you may see a difference in the frequency response plot, but in a general mechanical system, I don't think that would necessarily be true.

Paul Howard: Is the mechanical impedance independent of the size of the forcing function?

David Brown: Yes, only if it's a linear system. If it's linear then the frequency response or mechanical impedance is a unique function of the system. If it's non-linear, then the mechanical impedance of the system depends upon the preload or the input conditions of the system itself. There are techniques for estimating the effect of the input on the estimated frequency response.

Morris Weber, Hughes Aircraft Co.: How do you estimate the structural damping in your analysis?

David Brown: There are a number of ways of getting structural damping. I wrote a little paper that indicated 20 different ways of doing it. You can do it in the time domain which means you can take a signal and basically do a log decrement type analysis. You can go in and excite a certain mode of vibration and look at the rate of decay of that mode of vibration. You can do it with a Z transform technique, the Z transform being a discrete Laplace transform. The Navy has developed a complex exponential technique which fits a curve having the form $e^{-\alpha t}$ to the data where α is a complex number. The real part of that is the damping coefficient. If you measure the frequency response you can go into the frequency domain, and you can measure the half power points, which is the frequency spread between the half power points. You can go into the display of the frequency response plot where you have the real plotted vs. the imaginary and measure the rate of phase change in that plot. That's a measure of damping.

Morris Weber: I know you can measure it. How do you predict it when you do it analytically?

David Brown: Analytically of course it's very difficult to predict damping in any given system. Most of the damping will come in through joints and things of that sort which are very unpredictable. In a casting, for example, we sometimes detect cracks measuring the internal damping in the system. In that case the damping is a little more predictable and you could probably get a good handle on what the damping coefficient would be initially. But in a general system the only way that I know to determine damping coefficients is to measure them. It's very difficult to estimate damping coefficients. They are so much a function of the geometry and boundary conditions of the system.

Charles Jackson: I thought you had a real good correlation of high-Q resonance response to fast change in phase. But I noted that some of the phases were 90°, 180°, 270°. Could you comment on the various phase shifts?

David Brown: Actually in that plot that we were looking at the phase angle really went between 0 and -180°. I just moved the plot so it would keep from jumping back and forth between -180 and +180°. The phase angle was going between 0 and -180°. Everytime you go through a resonance, it drops 180°. When you go through an anti-resonance it goes up 180°. Now if you get on a cross transfer function or a cross frequency response which is exciting here and picking up there, then everytime the phase angle goes through a resonance, it decreases 180°. But you don't have a resonance followed by an anti-resonance, so the phase angle can go down 180°, then go through another resonance and drop down to 360°, and just keep on going down depending upon your frequency response plot. That frequency response plot was obtained with a transient input into the system. We actually hit it with a hammer and measured the frequency response that way. To do that you Fourier transform the response of the system and divide it by the Fourier transform of the input. Out near the end of the Fourier transform record, you run into some transform areas that cause a little bit of a problem.

Jack Frarey: I take it from the last comment of your talk, that if one were performing spectral analysis of a machine, and if he ran an RPM sweep in the region of interest, and for that system found this amplitude to be relatively stable over that RPM range, that this would then in fact be a safe region to perform what we are now calling the traditional spectrum analysis of wave form analysis. The primary problem here comes in trying to look at a component over a speed range.

David Brown: In a very dynamic system, there are a lot of resonances. The response of the system is going to change drastically as a function of speed, not based upon the forcing function, but based just upon the characteristics of the system that you are looking at. Somehow, you have to compensate for that change. A relatively flat region would indicate a good speed range to obtain signature analysis from that device.

William Glew, Naval Eng. Test Establishment: What type of orientation do you use to get such an effect to transform function from merely hitting the gear wheel with a hammer?

David Brown: Basically, we have a load cell mounted in a hammer which measures the force. We measure the response of the system with an accelerometer or displacement pick-up or velocity pick-up. The frequency response of the system is defined as a Fourier transform of the output of the system divided by the Fourier transform of the input. You can very easily Fourier transform the input which happens to be nearly a unit impulse.

William Glew: How are you doing this across your whole frequency spectrum in about 1/2 second or the time of your impulse?

David Brown: A unit impulse input into a system has a very broad frequency content. For example, a hammer blow will have a frequency content depending upon what type of hammer you are using up to 10,000 cycles. We have different hammers and different heads. We have hammers that go up to 10,000 cycles and they are all calibrated.

William Glew: So this is a numerical analysis, not a shock spectrum analysis?

David Brown: It's numerical. It uses the Fourier transform. If you have noise in the system and you use a statistical technique of averaging the cross spectral density and the par spectral density and take the ratio of the cross spectral by the par spectral, you will obtain the frequency response. No matter what the input is into the system, you can get the frequency response providing you can measure the input and output of the system, and the system is linear. For example, in a machine tool, frequently they mount a transducer in the cutting tool. Then they monitor the transducer that measures the relative motion between the tool and the work piece. The cutting force is the input and the relative motion is the output. The frequency response obtained from this happens to be a very important plot for predicting machine tool chatter. On an automobile you can mount a motion pickup on one of the wheels, and you can mount another pickup on the steering wheel. You get the frequency response between those two points simply running the car over a rough road.

William Glew: Do you get better results from this numerical analysis than with a standard impedance plot using a false head and a velocity measurement point?

David Brown: The mechanical impedance technique is sort of standard. It's a very controlled test. You go in and put in very known force conditions, you control both the static force levels and the dynamic force levels, if you choose. You can very carefully control those levels. So I would say that the results from numerical analysis and the mechanical impedance technique are comparable, but of course mechanical impedance is still the standard.

William Glew: We have been trying at the Naval Engineering Test Establishment to improve machine performance by alternating the resonant conditions, and of course we're defining the resonant conditions by impedance plots. In practice we find that non-linearities are such that you certainly cannot apply the theory in the way that you've defined in the lecture. I would like your comments on this.

David Brown: You can handle non-linear systems if you do it properly. It requires some special techniques. For example, Structural Dynamics Research Corp. has taken complete automobiles and trucks for Ford and General Motors and broken them down in terms of their components, such as the suspension system, the frame, the cab, the engine, etc. They measured the dynamics of the components and then put them back together and got reasonably good correlation. An automobile happens to be reasonably non-linear; components such as tires, suspension system, and shock absorbers are very non-linear. You can handle the linear components very nicely, but then when you put it together with the non-linear components you have to put it together with a simulation system that can handle non-linearities.

Alan Duguid, Huntington Industries: Did you use preload bearings in your test system, and if you did, did you change the preload?

David Brown: In the little gear system we looked at, we didn't use preloaded bearings, we just used very simple ball bearings. A lot of the big machine tools we test have preloaded bearings in the spindles. These may be tapered roller bearings, although it depends upon the machine. We've tested a lot of machines with hydrodynamic or hydrostatic bearings which are very non-linear as far as speed is concerned, and to a certain extent, as far as preload is concerned.

Alan Duguid: Did you change the preload to see whether that affected it?

David Brown: Anytime we work on a system of that type, we always change the input conditions to check for non-linearities. As I mentioned before, the frequency response of a system is a unique function if the system is linear. If it's non-linear, then it's a function of the input. The input is varied to determine if there are any non-linearities. Measuring the frequency response under several different loading conditions will determine the non-linearities. The type of non-linearity can be determined from the way the system is loaded.

Alan Duguid: At the end of his talk Mr. Jackson showed a device that was used for checking the balancing of a rotating system. It was a disk and I think it had 2 or 4 carbon brushes.

Charles Jackson: I was getting ready to discuss alignment. We align with the non-contact pick-ups as well as optically. There were two eddy current probes that were being gap calibrated, one horizontal and one vertical.

Alan Duguid: In roll grinding equipment, several people have developed balancing equipment which is essentially dynamic, since if the grinding wheel is breaking down all the time, the balance of the system is changing. There are several different ways of doing it, but they all basically end up as planetary types of systems that reorient themselves in space to take care of the breakdown in the wheels. Have you ever attempted to do anything with any of that large equipment that you're trying to balance?

Charles Jackson: The one example that I gave we balanced a 5,200 lb. rotor at 11,000 RPM's which was operating above its second critical. Here you really get into a complex situation if you balance below the first critical and then you get a phase shift above this. These shifts don't always come as predicted. We had a real dilemma there. We were running about 2 1/2 mils peak to peak which was causing a little bit of shaking in our governor system. Our question is, what are we going to depend on? If we used a stroboscopic approach, could we depend on the phase lag being correct, having gone through 2 resonances? We chose to use the orbit display to allow us to position the correction weight. In other words, if we had a circular orbit with a marker straight down, say at 6 o'clock, that means that at the instant in rotation when the notch, the one event mark, passes the probe, the shaft is down below the first critical. This is actually how we placed our weights and we made a correction on the rotor with 10 grams at a 6 inch radius slightly overhung at the exhaust end bearing. This is on a coupling side which is an exponential type of response. To place it statically we simply find the marker at 6 o'clock, stop the machine, and roll it around until the marker is indicated.

Alan Duguid: What happens on rolls for rolling sheet? In rolling copper it was found that they could grind the roll, but they couldn't see anything in the roll. You could see a blush in the material which was due to the vibration caused by the out of balance, and by incorporating a balancing mechanism into the grinding system at speed, they were able to eliminate this. I would assume that you have transient loads at speed when you are operating that change the balance. Have you done anything about this?

Jerry Forest, Ontario Hydro: You're talking of an automatic balancing device, and there are such devices, but they work only below the first critical frequency. In fact, there are commercially available devices for balancing motorcycle tires. A silicone liquid is poured in the tires, and when the motorcycle is driven, the wheels balance automatically as long as you are below the first critical speed. I would like to ask Mr. Brown, if it's possible to make an inverse of this whole approach at the design stage so that an equipment manufacturer can predict even in a complex structure, what his system dynamics will be. If this can be done, he may be able to change his design so that his operating conditions will be in such a location that the system dynamics will not cause any problems. My experience has been that equipment suppliers perform usual natural frequency calculations for rotors and other such things, but they really don't relate to the practical "as installed" machine. There are definite problems later on that equipment manufacturers usually aren't able to cope with.

David Brown: I think the technology is essentially available. For example, that big electrical motor that I mentioned was a very complicated system consisting of 5 components, a base, foundation, rotor, bearing, and stator, basically tied together and they did a very good job of predicting that. The base and motor were finite-element modelled. That way they could incorporate design changes in those 2 components and then determine what the effect on the system would be. The idea was to keep resonances away from 3600 RPM's, which was the operating speed of the motor. The only problem is that right now this technique is still very expensive to use as a design tool. If you set the motor on any foundation, you can predict how that foundation will influence the overall dynamics of the system. This technique has been pretty much developed by Structural Dynamics. It's called the building block approach. This technique can be used, but right now it's expensive and only people like the manufacturers of very big equipment or equipment that is mass produced, can afford to apply this technique. Presently it would take maybe 2 or 3 months to do a complete design analysis of a big motor and may cost 30-40 thousand dollars.

Morris Weber: How many degrees of freedom are there?

David Brown: The degrees of freedom in that system are tremendous. In the 50 by 50 matrix describing the system, each term was a complete frequency response which may have 10 modes of vibration in it. One of the reasons for using this particular approach is that if you take a rather complicated system and describe it in terms of 10 modes of vibration, you talk in terms of modal properties. That tremendously reduces the effective degrees of freedom that must be carried through the computation in the system simulation. For example, in finite elements, you would have to carry a degree of freedom for each finite element in that particular model. In this case, all you do is consider the connecting point between two pieces of equipment. It tremendously reduces the effective degrees of freedom that you have to carry through your model.

Charles Jackson: We tested our last 14 cases at 1,000 RPM increments and we do take amplitude phase readings from the test stand; that's while it's still at the vendor's location. We had 3 units that missed the calculated first and second criticals by 21%. I don't really much care what the calculated value is. My main concern is what the actual value is. Some of the actual values do change under load. We're running under helium and we can't fully power our load. I think the move is to do more testing and get this problem corrected before it leaves the vendor's shop because it's difficult to correct it once you get in the field.

John Sudey, NASA: Did you ever look into what the coherence function is between the input and the output that possibly could give you indications of non-linearity of the system?

David Brown: We use coherence function quite a bit. I wrote a paper for the Internoise Conference in Washington that used coherence functions to locate sources of noise. It turns out that coherence functions are very difficult to use for non-linearities, merely because the coherence function is very much a function of any noise in the system, and the effect of non-linearities on the coherence function is very poorly understood. We are going to do some work for GE Evandale studying coherence functions because they would like to use it to identify sources of noise in the jet engine. The noise generated in the jet engine has to traverse through a shear layer, which happens to be a non-linear flow field, so acoustically it turns out to be a very non-linear element. They want to know the effects of non-linearities upon the estimate of coherence functions.

John Sudey: In the motors, did you examine the current which could possibly be an indication of a torque vs. the response?

David Brown: No, we were looking at the dynamics of the motor, not particularly the operating dynamics. We were interested in modes of vibration, that would be excited by any sort of magneto constrictive effect or the rotation of the motor itself. We were looking for resonances that would be excited by those forcing functions.

John Sudey: How do you know where to hit with a hammer?

David Brown: Normally you want the frequency response between two points in the system, or you want to get a mode shape which would be a series of frequency response plots. So what you do, is define the points of interest in your system and that's where you hit it. If you want a frequency between this corner and the table and this point over here because this is where a bearing is located or a force comes into the system, and a frequency response over here where your pick up is, you hit it either here or here, it doesn't make any difference. You use Maxwell's reciprocals.

John Sudey: What do you do with the system you can't get inside of?

David Brown: Then, of course you're stuck.

SESSION III

DIAGNOSTIC

SYSTEMS

APPLICATIONS

Chairman: Keith R. Hamilton
Wright Patterson Air Force Base

THE FIRST AUTOMATIC FAULT ISOLATION TESTING SYSTEM FOR RECIPROCATING ENGINES DESIGNED FOR PRODUCTION USE

by

George R. Staton

Fire Direction & Diagnostic Systems Division

Frankford Arsenal
Philadelphia, PA

The system described in this paper is known as DepotMAIDS---Depot Multipurpose Automatic Inspection & Diagnostic System. It was designed to supply the depot, for the first time, with a method for evaluating incoming assets to determine whether or not incoming engines are candidates for rebuild; and if found repairable, to indicate what repairs are required to place them in a fully operational condition.

Previous to the advent of this system, all incoming engine assets were torn down and completely rebuilt. In addition to this diagnostic function, the DepotMAIDS also serves as a quality assurance tool to insure that the repairs or rebuild have been properly accomplished, provides for automatic run-in after rebuild or major repair, and a complete product assurance test.

The system as presently configured consists of two test cells supervised by a general purpose digital computer. Complete instrumentation to acquire those measurements necessary, as well as control subsystems and switching, are provided.

The system block diagram is shown in Figure 1. This system is currently being extended to two additional engines cells and a cell for the testing and diagnosis of transmissions. When fully extended, the system will be capable of testing and diagnosing engines in 1790 cubic inch class, as well as two cycle diesel engines of the 6 and 8 cylinder variety. It will also handle all cross-drive transmissions in the Army inventory. Thus, the capabilities will extend to all combat vehicle engines and transmissions.

Figure 2 diagramatically shows the capability of the system as it is presently configured, and as it is being extended. Figure 3 summarizes the software functions designed into the system. A real time monitor controls the operation of the entire system, providing capability for self-checking, testing of units under test, reporting, and includes supervisory test plan monitor for each of four cells.

Each test cell operator thinks that he has full and exclusive use of the computer at all times. This capability is provided by the test plan monitor. Test cells can run fully asynchronously under this software system.

Figure 4 shows the instrumentation subsystem as it is configured for the 6V53 2-cycle diesel engine. The test cell subsystems are those permanently installed transducers and interlocks which are housed in the test cell. In addition to this instrumentation, temporary instrumentation for testing purposes is housed in kits which are applied to the engine prior to entry into the test cell. Also in the test cell are the transducer enclosures for the temperature transducers. Housed in the test cell operator station are the electronics, consisting of signal conditioning and control systems for the dynamometer and power supplies, which feed the individual test cell instrumentation subsystem. Back at the Master Controller, additional electronics are provided to interface the test cell with the Master Controller.

Figure 5 shows a listing of the 6V53 engine measurements used for runin and diagnoses. Two basic types of measurements are taken in the testing process: those which are characterized as steady-state; that is, taken at a fixed power point, and averaged over a number of engine cycles to obtain the mean value of the reading; and time varying measurements, such things as vibrations, injection processes, valve closing events, and so forth, which are taken instantanously, and analysized for proper functioning of the system under test.

Readings taken are then compared with prestored limits and grouped in what is known as a "truth word" for comparison with a prestored "truth table" to determine what specific malfunction, if any, is present. Results of this analysis are then printed out in plain English for the operator.

A typical truth table is shown in Figure 6, and the Printout as Figure 7.

The system has been operational since March of 1971, and has been highly successful during that period. Over 300 engines have been tested by the system during this period, either for quality assurance or for diagnostics. Significant savings have been achieved in both areas. For those engines tested diagnostically, the costs have averaged $2,000 per engine, as compared to an average of $7,500, if the engine is completely rebuilt. In addition to these savings, considerable savings have accrued from the fact that the engines rebuilt are carefully inspected so that no potential or latent defect escapes undetected. Therefore, their performance in the field is significantly enhanced.

This system has been operated on a two-shift, 5-day a week basis, and has experienced only three days down time in over two years of operation.

This digital computer controlled automatic test and diagnostic system, which is installed at Letterkenny Army Depot at Chambersburg, Pennsylvania, is contributing significantly to the modernization of our depot maintenance program. This system is the first of its kind in the world, and has been cited in the literature, both here and abroad, as a prime example of the future concepts in automotive maintenance.

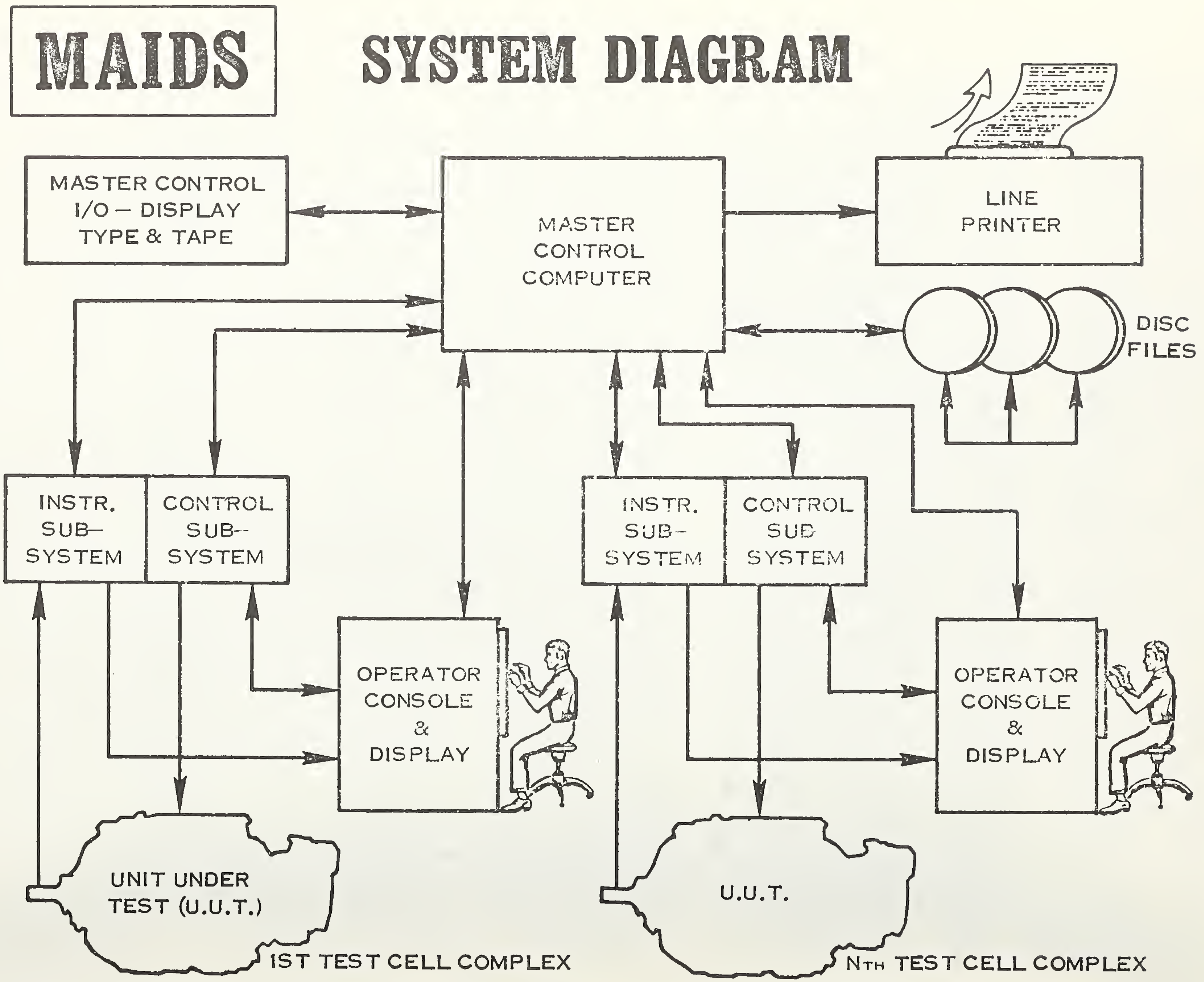


Figure 1. System Block Diagram

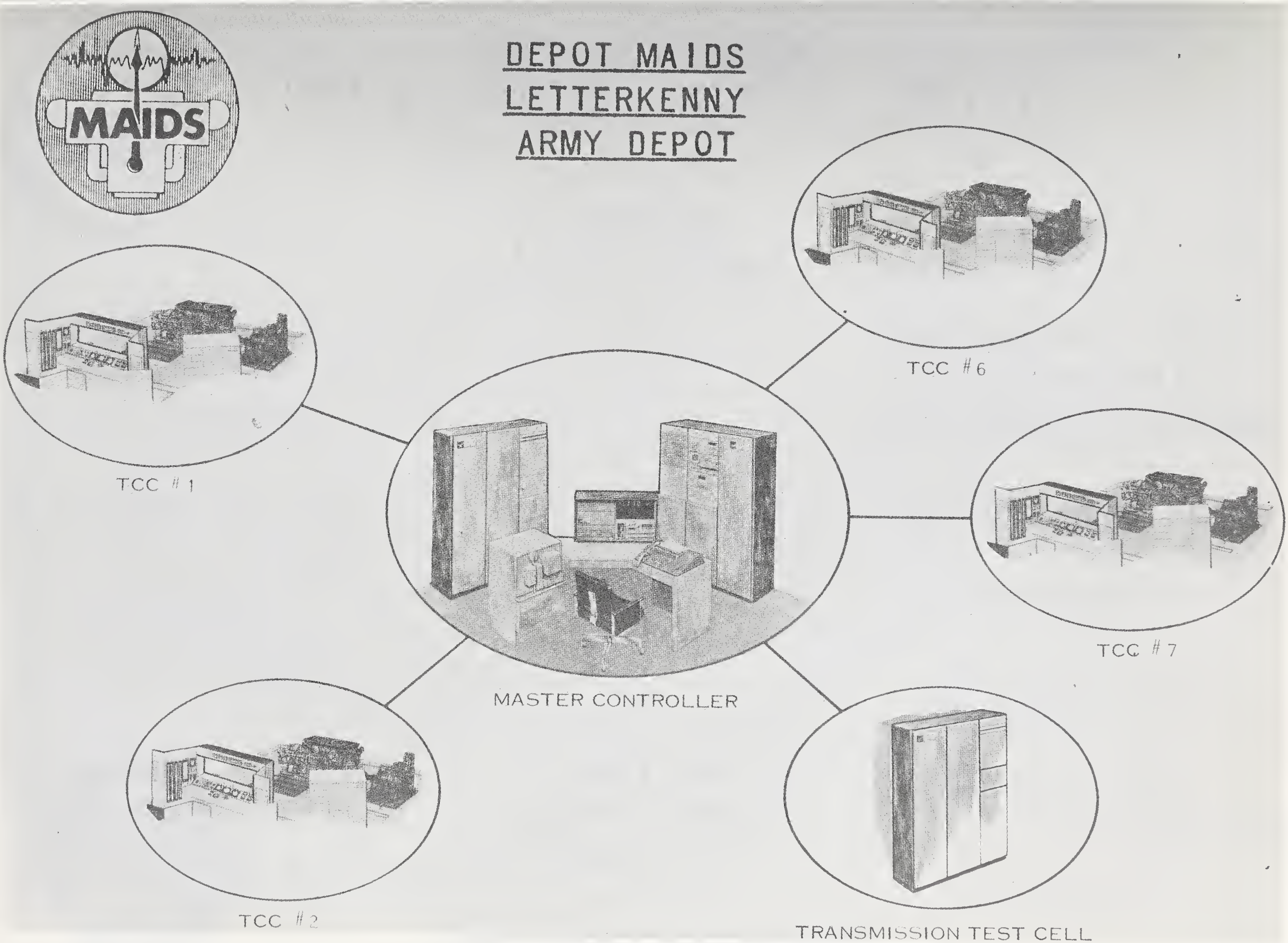


Figure 2. Capability of the System

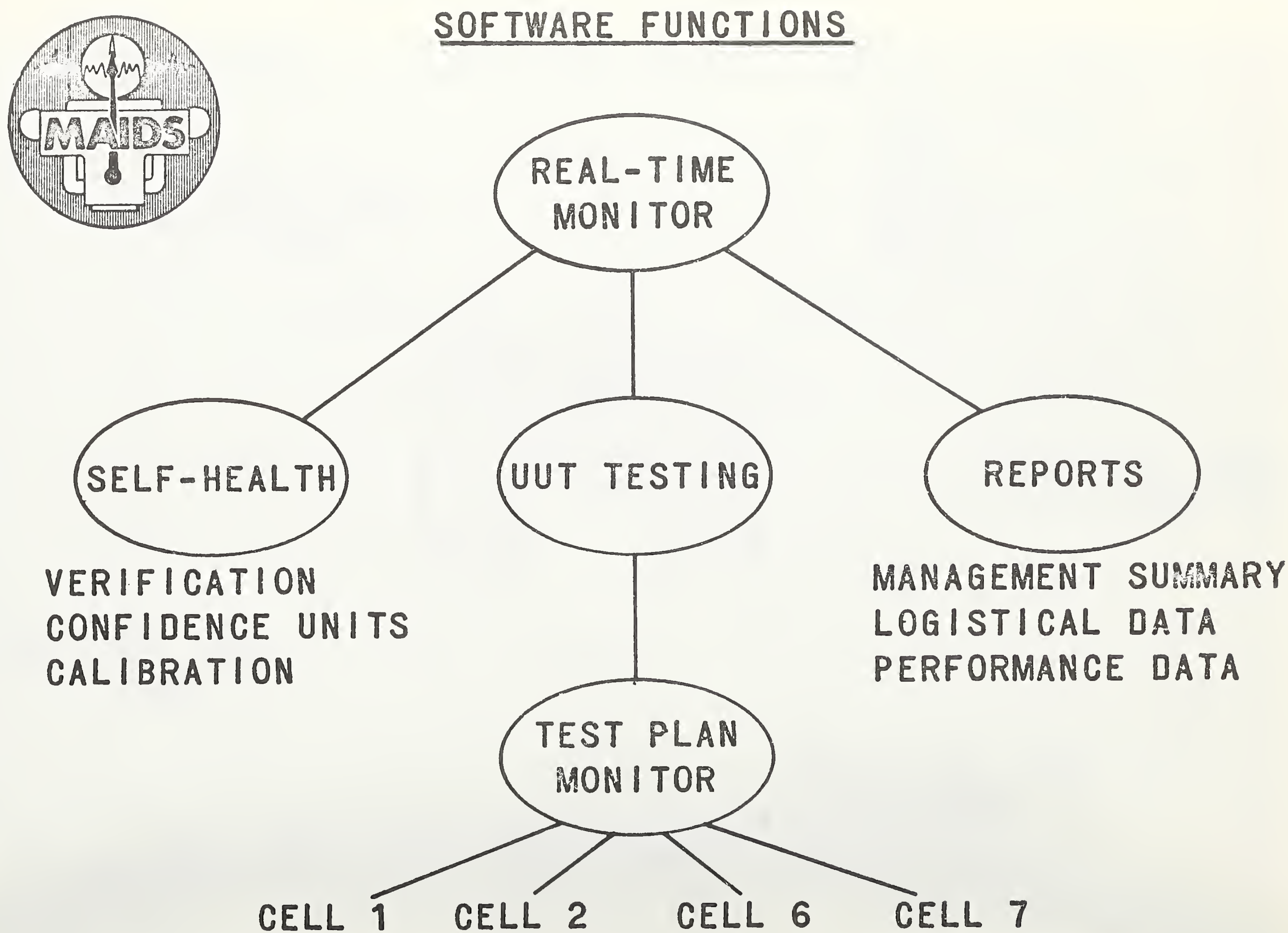


Figure 3. Software Function

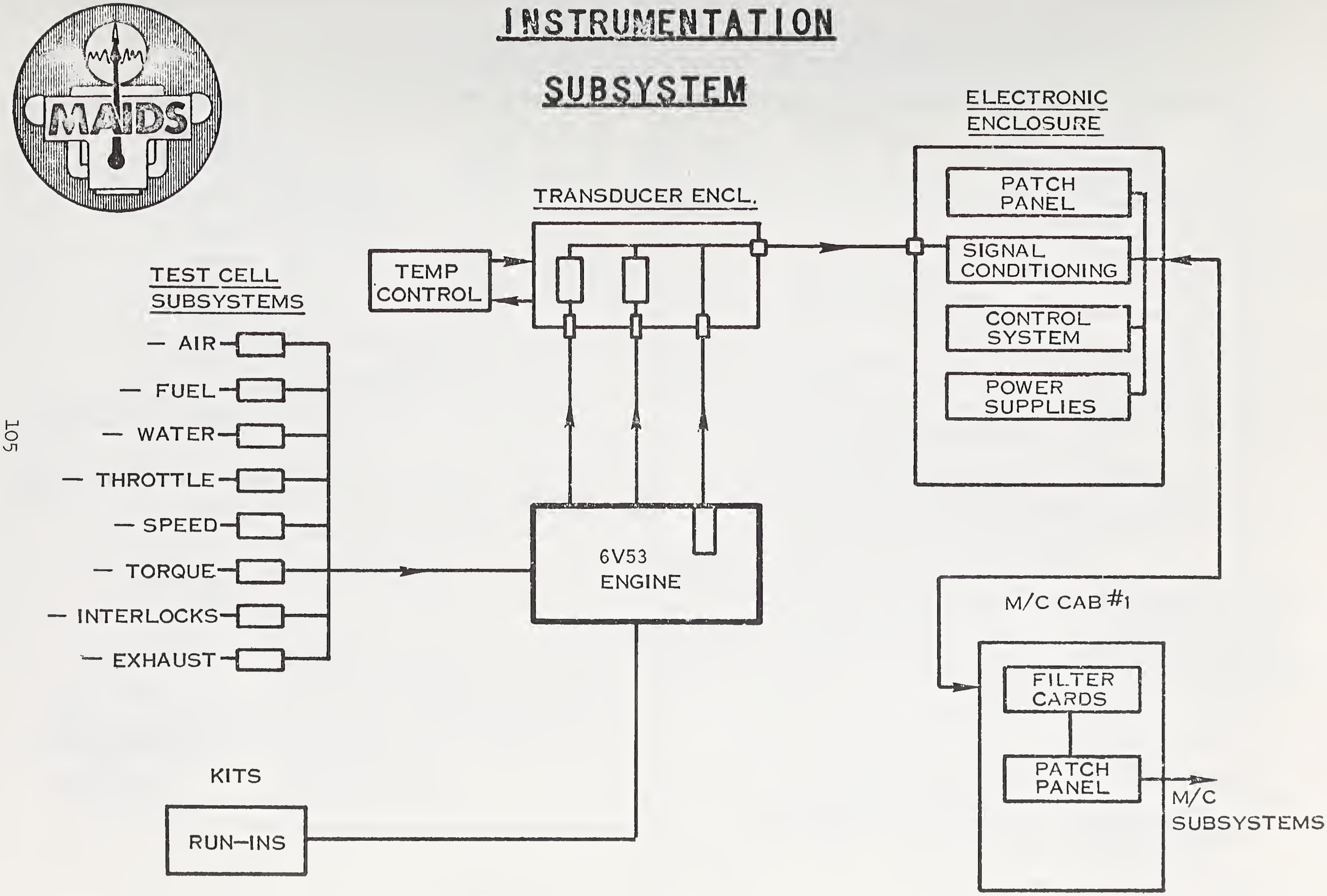


Figure 4. Instrumentation Subsystem

TYPICAL DIESEL ENGINE MEASUREMENTS

MEASUREMENT	SENSOR LOCATION	ENGINE SYSTEM ANALYSIS
TEMPERATURE (25 POINTS)	(12) EXHAUST PORTS (PIPE PLUG)	POWER & INJECTORS
	(1) OIL BEFORE ENGINE OIL GALLERY (OUT OF COOLERS INTO ENGINE)	LUBRICATION AND SAFETY MONITOR
	(2) OIL COOLING LINE (OUT OF ENGINE INTO OIL COOLERS)	LUBRICATION
	(1) FUEL SUPPLY LINE (SERVICE SUPPORT SYSTEM)	CORRECTION FACTOR
	(1) FUEL RETURN LINE	CORRECTION FACTOR
	(2) OIL RETURN LINE (TURBO SUPERCHARGER)	LUBRICATION & TURBO–SUPERCHARGER
	(2) EXHAUST OUTLET (TURBO SUPERCHARGER DIAGNOSIS)	TURBO–SUPERCHARGER
	(2) INTAKE MANIFOLD (TURBO SUPERCHARGER DIAGNOSIS)	TURBO–SUPERCHARGER
	(2) INTAKE AIR FLOW (TURBO SUPERCHARGER INLET)	CORRECTION FACTOR
	(1) AMBIENT	CORRECTION FACTOR
PRESSURE (9 POINTS)	(2) INTAKE MANIFOLD (AFTER TURBO)	TURBO–SUPERCHARGER AND INTAKE SYSTEM
	(1) ENGINE OIL GALLERYS (PLUG LEFT SIDE OF CRANKCASE REAR)	LUBRICATION AND SAFETY MONITOR
	(1) ENGINE OIL (OUT OF ENGINE INTO COOLERS)	LUBRICATION
	(2) AMBIENT PRESSURE (AT INTAKE TO TURBO SUPERCHARGER)	CORRECTION FACTOR
	(2) INJECTION PUMP	INJECTION PUMP, INJECTOR S, TIMING
	(1) FUEL PUMP (AT INJECTION PUMP INLET)	FUEL
VIBRATION (18 POINTS)	(12) ROCKER BOX (VALVE) COVER (ACCELEROMETER)	VALVE, INTAKE & EXHAUST, ADJUSTMENT & TIMING
	(1) ENGINE BLOCK VIBRATION (BLOCK ACCELEROMETER)	INTERNAL MALFUNCTIONS (BEARINGS, ETC.)ISOLATOR
FLOWS (7 POINTS)	(1) FUEL SUPPLY LINE (DIESEL SUPPLY LINE IN SERVICE SUPPORT SYSTEM)	INTAKE & FUEL
	(1) FUEL RETURN LINE (DIESEL RETURN LINE IN SERVICE SUPPORT SYSTEM)	INTAKE & FUEL
	(1) BLOW BY (BREATHING SYSTEM)	ENGINE WEAR
	(2) ENGINE OIL (OUT OF ENGINE INTO COOLERS)	LUBRICATION
	(2) AIR–INTAKE (SUPERCHARGER)	INTAKE
CURRENT (1 POINT)	STARTER (SERVICE SUPPORT SYSTEM)	STARTER & ENGINE COMPRESSION
POSITION (1 POINT)	CRANKSHAFT	TIME BASE
SPEED (3 POINTS)	(2) ENGINE COOLING FAN (CRITICAL MALFUNCTION)	COOLING SYSTEM
	(1) ENGINE UNDER TEST	BASIC CONTROL
TORQUE (1 POINT)	(1) DYNO (TEST CELL 2)	BASIC CONTROL
THROTTLE POSITION (1 POINT)	(1) ENGINE UNDER TEST	BASIC CONTROL
VOLTAGE (1 POINT)	MAGNETO PRIMARY	IGNITION
	STARTER	IGNITION

Figure 5. 6V53 Engine Measurements

GENERAL MALFUNCTION AREA	MALFUNCTION / MEASUREMENTS	ENGINE OIL PRESSURE	ENGINE OIL SUMP TEMPERATURE	TOTAL OIL CONSUMPTION	CYLINDER TEMPERATURE	INTAKE MANIFOLD PRESSURE LEFT BANK	INTAKE MANIFOLD PRESSURE RIGHT BANK	COOLING FAN AIR VELOCITY ACCESSORY	COOLING FAN AIR VELOCITY FLYWHEEL	BLOWBY	INDIVIDUAL CYLINDER POWER DROP	CORRECTED B HP	AIR-FUEL RATIO	VOLUMETRIC EFFICIENCY	BSFC	BSOC
LOWER CRANKCASE	MAIN, AND / OR CONNECTING ROD BEARINGS WORN	Lo		Hi												Hi
	BENT CONNECTING ROD				Hi						Lo	Lo			Hi	
UPPER CYLINDER	PISTONS, RINGS AND CYLINDER WALLS WORN			Hi		Hi	Hi			Hi	Lo	Lo			Hi	Hi
AIR COOLING SYSTEM	BROKEN OR STRIPPED COOLING FAN DRIVE		Hi		Hi			O	O							
	SLIPPING COOLING FAN CLUTCH		Hi		Hi			Lo	Lo							
LUBRICATION SYSTEM	DEFECTIVE OIL PUMP AND / OR PRESSURE REGULATOR	Lo / Hi	Hi													
INDUCTION SYSTEM	AIR INTAKE AND/OR CARBURETOR RESTRICTION				Lo	Hi (AND OR)	Hi (AND OR)									
	INTAKE MANIFOLD LEAKS AIR				Hi / Lo	Lo (AND OR)	Lo (AND OR)									
POWER SYSTEM	DEFECTIVE FUEL PUMP		Hi		Hi ALL							Lo	Hi		Hi	
	CARBURETORS OUT OF ADJUSTMENT		Hi		Hi ALL							Lo	Hi / Lo		Hi	
	THROTTLE RODS OUT OF ADJUSTMENT					Hi (DIFFERENCE)	Hi (DIFFERENCE)									

Figure 6. Truth Table

TYPICAL LOGGING TYPEWRITER DATA
DIESEL DIAGNOSTICS

ENGINE MODEL NO. 1790-2A   TRANSMISSION MODEL NO.   MAGNETO TYPE NA   DATE OF TEST 1-21-71
ENGINE SERIAL NO. 123123   TRANSMISSION SERIAL NO.   XC

DESCRIPTION AND CODE	FED STOCK NO	ORD NO - PARTS MANUAL	QUANTITY	SECT.
STARTER CURRENT ANALYSIS			REPAIR REFERENCE - TM 9-2815-200-35	
TRUTH TABLE NAME PT0012 1B CHECKSUM 1024 DATE 6100				
LIMIT TABLE NAME PL0025 CHECKSUM -28454 DATE 8070				
(ABSENCE OF MALFUNCTION LIST INDICATES NO PROBLEMS)				
INJECTION SYSTEM ANALYSIS			REPAIR REFERENCE - TM 9-2815-200-35	
TRUTH TABLE NAME PT0010 1A CHECKSUM 1032 DATE 6100				
LIMIT TABLE NAME PL0022 1A CHECKSUM 13562 DATE 11050				
REPLACE FUEL INJECTOR 5R	2910- 64-6269		1	58
REPLACE FUEL INJECTOR 3R	2910- 64-6269		1	58
REPLACE FUEL INJECTOR 6R	2910- 64-6269		1	58
REPLACE FUEL INJECTOR 2R	2910- 64-6269		1	58
REPLACE FUEL INJECTOR 4R	2910- 64-6269		1	58
REPLACE FUEL INJECTOR 2L	2910- 64-6269		1	58
REPLACE FUEL INJECTOR 4L	2910- 64-6269		1	58
REPLACE FUEL INJECTOR 1L	2910- 64-6269		1	58
REPLACE FUEL INJECTOR 5L	2910- 64-6269		1	58
REPLACE FUEL INJECTOR 3L	2910- 64-6269		1	58
REPLACE FUEL INJECTOR 6L	2910- 64-6269		1	58
UNCATEGORIZED INDICATORS				
,2- 16				
INTAKE VALVE ANALYSIS			REPAIR REFERENCE - TM [illegible]	
TRUTH TABLE NAME PT0005 1B CHECKSUM 28672 DATE 6110				
LIMIT TABLE NAME PL0012 4A CHECKSUM 9022 DATE 1131				

Figure 7. Printout

CONDITION MONITORING FOR AIRCRAFT GAS TURBINE ENGINES, PAST, PRESENT, AND FUTURE

by J. F. Kuhlberg and R. K. Sibley
Pratt & Whitney Aircraft

INTRODUCTION

The cost and complexity of advanced turbine powerplants coupled with requirements for lower operational and maintenance costs are the basis for growing interest in propulsion condition monitoring systems. Additionally, electronics is playing an increasingly important role in turbine engine control and power management. The propulsion system variable geometry inlet and engine controls, the condition monitoring system, and the propulsion management system can all be designed around electronic computers for advanced engine designs. It would seem logical therefore to think in terms of integrating these systems for cost, weight, and space savings through the use of common electronics. This paper explores possible methods for integrating condition monitoring systems with modern propulsion system control and management requirements and identifies advantages and disadvantages of various integration schemes.

Before addressing the integration problem, let's briefly review what conditioning monitoring is, how it's been used in the past, how it is used today and how it may be used in the future.

What Is It

The monitoring of critical aircraft gas turbine engine operating parameters to assess the engine health is achieved through a process called condition monitoring. Condition monitoring is a diagnostic process which results from the application of fundamental engineering laws and specific empirical relationships to determine the condition of the engine at any point in time, and provide necessary corrective action. See the diagram on Figure 1. The system provides the gas turbine operator with an understanding of the engine's condition in terms of mechanical and structural health, aero and thermodynamic performance, oil and fuel system condition and control system performance. The application of engine condition monitoring can afford the operator cost effective support in the maintenance and operation of aircraft gas turbine engines. It permits the timely scheduling of overhaul or repair based on actual engine condition, rather than on the traditional time basis alone, thus extending engine "up time" by avoiding unnecessary removals. Additionally, operational cost savings can be achieved by the diagnostic system through the avoidance of flight delays, inflight shutdowns, mission aborts, and other costly occurrences.

Past

The concept of diagnostics has, in one way or other, been applied to aircraft engines since the beginning of aviation. Figure 2 is a picture of one of the earliest specialized diagnostic devices, known as the "Magic Wand". In past years, diagnostics were employed to determine whether anything had happened, hence mandatory maintenance was needed before the engine could continue in operation. These monitoring practices included walk around inspections of the powerplant, periodic inspections of lubrication and fuel system screens and plugs, and removal of the engine after a critical operating limit (vibration or turbine temperature) had been exceeded. Many of these techniques proved very useful and are still employed.

Present

Presently, additional diagnostic techniques are being used on aircraft gas turbines to determine if an engine event is imminent, is happening or has occurred such that engine maintenance is required. Included are such techniques as monitoring engine aero and thermodynamic performance data either on the ground or in the air to determine whether the engine components are performing nominally, monitoring mechanical systems by measuring vibration, oil pressure, oil quality and quantity, and a variety of non-destructive tests such as borescope or X-ray inspections. In addition some diagnostic equipment which correlates the life of certain components has been introduced, particularly with regard to gas turbine hot section parts.

Figure 3 is a photograph of a so-called "hot section analyzer" which monitors time-temperature exposure of critical hot section (burner and turbine) components. When the Hot Section Factor, a time temperature integration, reaches a predetermined level the hot section should be inspected. Additional information regarding exposure to various turbine temperature levels are also displayed.

Spectrographic oil analysis is another technique which has been adopted by many operators. A sample of engine oil is analyzed at regular intervals to determine the presence of metallic particles suspended in the oil. A change in engine condition is indicated by a change in the count of metallic particles in the oil.

There are other diagnostic techniques which have been used in the past and present, and are too numerous to review in detail in this paper. Many of them have been successful in some degree.

Future

The future of condition monitoring lies in the combining of many present day and new diagnostic techniques in an automated system. With this variety of information available to a common system, substantially greater information on propulsion system present and predicted condition will be quickly and accurately determined.

In the past, the automated engine condition monitoring system has been difficult to economically justify as a system by itself. On future aircraft, electronics will be playing an increasingly greater role in the airbreathing propulsion system. Engine control, inlet control, propulsion management and condition monitoring are all requirements of these advanced propulsion systems. Each of these control and management requirements is envisioned as employing basic designs centered on dedicated digital computers. In the interest of reduced cost, weight and size, combining of these various propulsion related computer functions must be considered. The integration of various condition monitoring diagnostic routines with the propulsion control and/or management requirements through the use of common sensors, signal conditioners computational elements and display devices may provide automated condition monitoring as a relatively inexpensive by-product.

Two potential integration concepts that will be discussed in detail are:

- Propulsion management and condition monitoring
- Propulsion control and condition monitoring

Before getting into these systems, let's define propulsion management and control. Propulsion Management is a system which strives to optimize mission performance by scheduling engine power setting for take-off, climb, cruise, etc. and matching power between engines on a multiengine aircraft.

Propulsion control is a system which modulates the fuel to the engine while coordinating the propulsion system variable geometry such as spike, by pass door bleeds, stators and nozzles. The control is also designed to protect the propulsion system against destructive limit exceedance such as over speed and over temperature.

Condition Monitoring and Power Management Integration

One approach to the integration of diagnostics is an engine condition monitoring system coupled with a propulsion management system. This consists of both airborne and ground support units. Schematically, the airborne system consists of the components shown in Figure 4. Many of the requirements for engine condition monitoring are compatible with those of a propulsion management system. For instance, the engine instrumentation requirements for a propulsion management system consists principally of engine aerodynamic parameters and comprises, broadly 70% of that required for engine monitoring. Airframe parametric instrumentation (i.e., alt., IAS, etc.) required for propulsion management more than satisfies the monitoring requirements. Another compatibility point in propulsion management is in the signal conditioning requirements, whereby engine and airframe signal sources are processed in order to provide proper input into the data processor. The added engine monitoring data requirements will mean that a signal conditioning system will have to provide some additional electronic capability but the external design considerations, weight, volume, and cost should not be greatly affected.

The computer function in a propulsion management system is to process engine and airframe data, and conduct programmed calculations to provide information that will result in optimizing the mission being conducted. This will include information regarding optimum airframe configuration and engine power scheduling for the mission. This same computer could be used to process engine data to monitor condition. The major requirement is to provide proper software in the computer so that the engine health can be assessed. This means computer memory capacity in the propulsion management computer would be increased but system weight, size and cost characteristics should not be greatly affected.

The function of the display in the propulsion management system is to provide the pilot with specific action required to achieve mission optimization either through airframe or engine operating procedures. This display can also be used in engine condition monitoring by providing a diagnosis of any engine condition that affects the mission and the necessary corrective action.

Completing the airborne monitoring requirements are a tape recording system and maintenance advisory device. The maintenance advisory device provides a gross diagnosis, while the tape recorder acquires data for detailed diagnosis. Some recent work has been conducted, under a Navy contract, in evaluating the feasibility of some of the airborne system just described. A picture of the airborne signal condition/computer and the maintenance advisory panel are shown on Figure 5. Approximately 50 parameters are acquired in the signal conditioner. Computations are then conducted on a real time computer which provide a gross analysis to the advisory panel which has 43 different maintenance flags concerning the engine, fuel and control system.

A ground processor should be used to augment the in-flight diagnostic capability achieved from the monitoring portion of an integrated system. The general function of the ground processor is shown on Figure 6. The ground processing system must do detailed computations, requiring a large memory. While immediate maintenance action requirements and a go/no-go status are provided through the airborne system, a ground system should be available with a minimum time lag to provide an up to date permanent record of engine events and much more detail regarding engine condition and any required maintenance action. In addition, the trending data acquired at pre-selected flight conditions should be evaluated to see when engine shifts from nominal are projected. The ground processor system can be dedicated to just the engine data reduction function or, as an alternative, time sharing of a base computer could result in a cost effective method of processing the engine data. A picture of the ground processing system currently in use in the Navy program is shown in Figure 7. This is a 16K memory computer. It provides a permanent record of the engine condition items indicated on the advisory panel and can augment the indications of the panel with about 20 more diagnostics.

Condition Monitoring, Engine Control Integration

Figure 8 illustrates a possible functional integration of the propulsion control and condition monitoring system. In this example, the condition monitor can diagnose the inlet, engine, and control using the same parameters. Control inputs and outputs are supplied to the condition monitor. Other non-control related parameters such as vibration, oil condition, etc. are also supplied to the monitor to fill out its repetoire. Discrepancies between control inputs and outputs as determined by the condition monitor indicate control malfunction and discrepancies between the control outputs and inputs indicate possible engine problems.

The apparent attractiveness of the integrated electronic systems must be carefully traded against important considerations such as reliability, maintainability, and basic requirements of the systems. Idealistically, the reliability of the condition monitor must be better than that of the propulsion system. The use of common sensors and computer carries with it the possibility that a control failure also becomes a condition monitoring failure. Thus at the crucial point when a system failure occurs, the condition monitor also fails. Use of redundant sensors and computers can alleviate this problem while lending a fail-operational capability to the control system.

Data display constitutes another area where the pros and cons of integration must be studied and traded. A data display under the control of the condition monitor will provide the flight crew with the most complete information regarding the propulsion system. This type of data is probably desirable in that it provides comprehensive data for mission completion and safe return decisions. On the other hand, a condition monitoring data display is unsatisfactory in the event of a condition monitoring computer system failure. In this case the pilot has no reference from which to operate his engines. This suggests that a completely separate additional instrumentation system may be required to supply the flight crew with the essential information needed to operate the engines safely. Also the human engineering aspects of any data display need to be thoroughly evaluated before settling on a final system.

Other considerations must be addressed when studying possible integration of system control and monitoring elements. Paramount among these is computer cycle time. Presently, a limiting element in propulsion control by digital computer is the time needed to run through a single complete computation of the control laws. The addition of condition monitoring equations and logic adds to this calculation time. In some cases the time addition can be reduced by performing only a part of the condition monitoring laws on each control calculation pass. This is acceptable because, whereas the control calculations need to be updated on the order of 50 times per second, condition monitoring data need only be updated once per second and slower in some cases.

Computer accessibility must also be considered in determining the advisability of system intergration. The condition monitor system must be readily accessible for routine maintenance after each flight. The data tape recorder must be changed and flag panel checked and reset for the next flight after servicing. The engine control computer, on the other hand may best be located on or near the engine to minimize communication problems between the engine and control.

The number of engine condition monitoring computers required for a multiengine aircraft also must be considered. While it may be desirable to maintain individual engine controls, a single condition monitoring system may suffice for the aircraft.

Conclusions

Integration of propulsion system controls, airframe controls, avionics and condition monitoring is possible; however, careful trade studies must be conducted to insure that the reliability, maintainability, vulnerability, and other aspects of the various systems is maintained. In addition the trade studies must insure that integrated computation and data management is in fact the lightest, smallest, and lowest cost approach.

In all probability there will be some integration of the various electronic systems on board future aircraft. Initial integration will be system related, such as propulsion controls, inlet and engine, followed by the integration of peripheral systems such as propulsion management and condition monitoring. Condition monitoring may be split between the propulsion control and central power management systems. Central checks and engine limit exceedance may be monitored by the control while general engine condition is monitored by the propulsion management system with data provided by direct sensing and from the control.

In summary, we have observed the following:

1. Diagnostic procedures have proven valuable in the condition monitoring of operational gas turbines.

2. Advancements in electronics have now made possible the combination of power management, engine control and condition monitoring into an integrated system.

3. Engine monitoring is a cost effective by-product of the control and power management system requirements being determined for future aircraft.

4. Electronic systems made up of sensors, avionics, and computers must demonstrate satisfactory characteristics in the areas of reliability, accuracy, and maintainability if the full potential of integrated electronic engine control/monitoring/management is to be achieved.

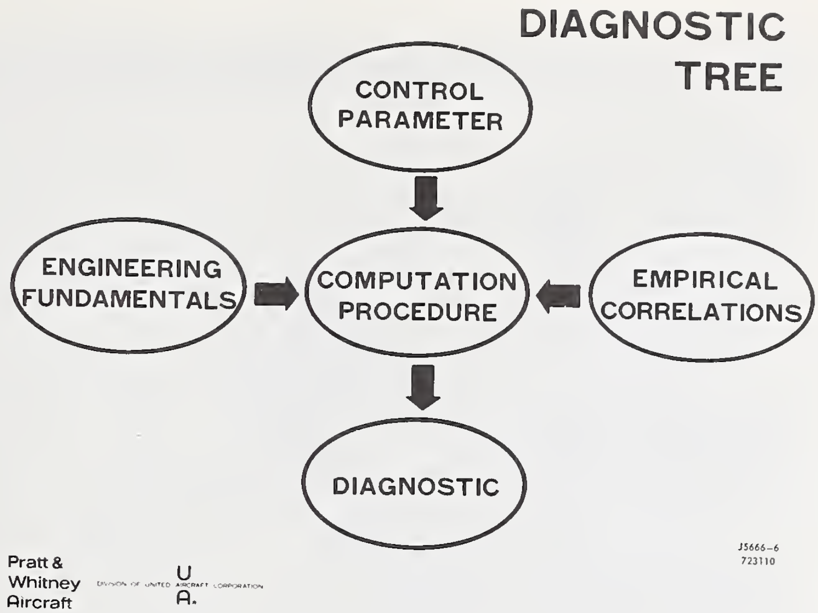
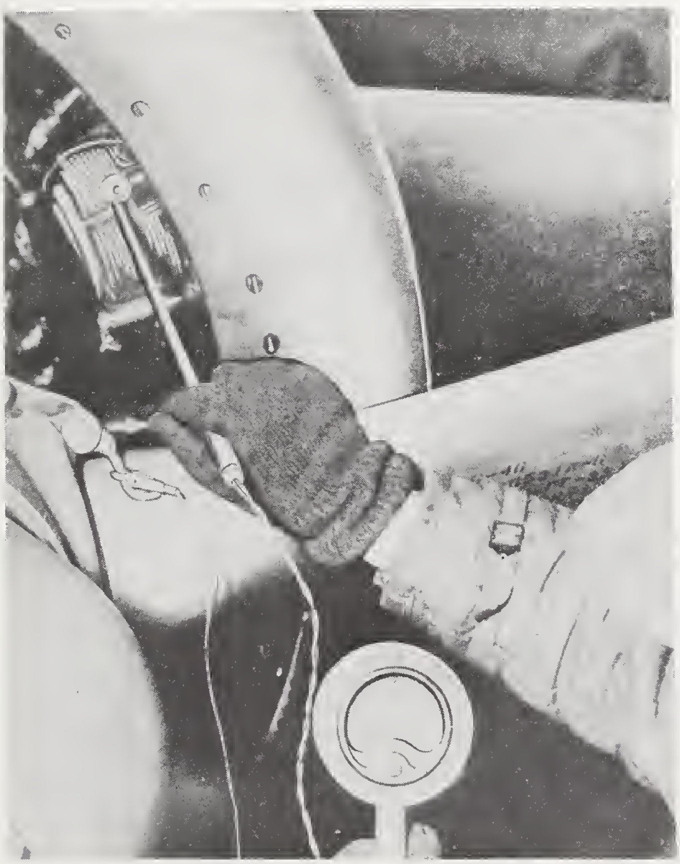


Figure 1



"Magic Wand"

Figure 2

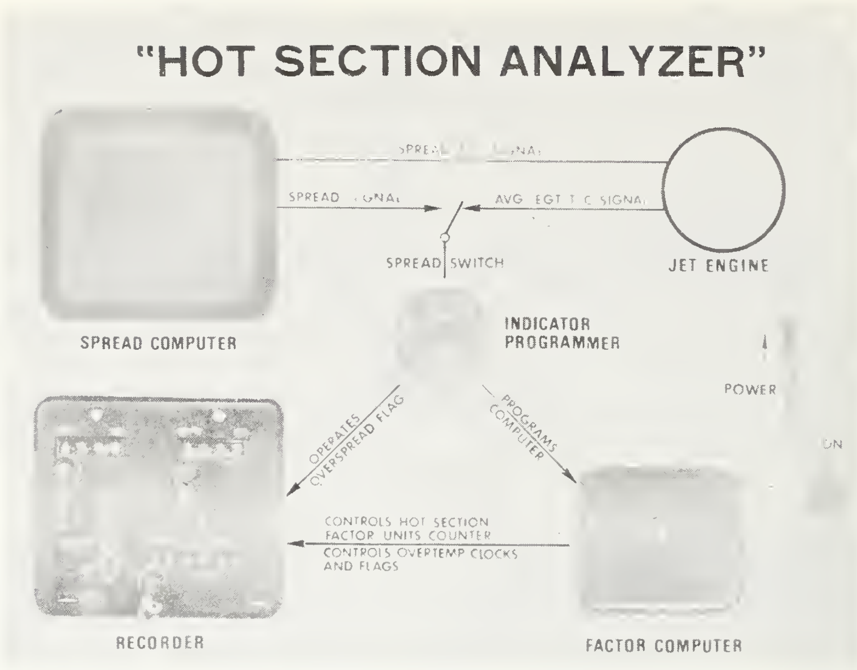


Figure 3

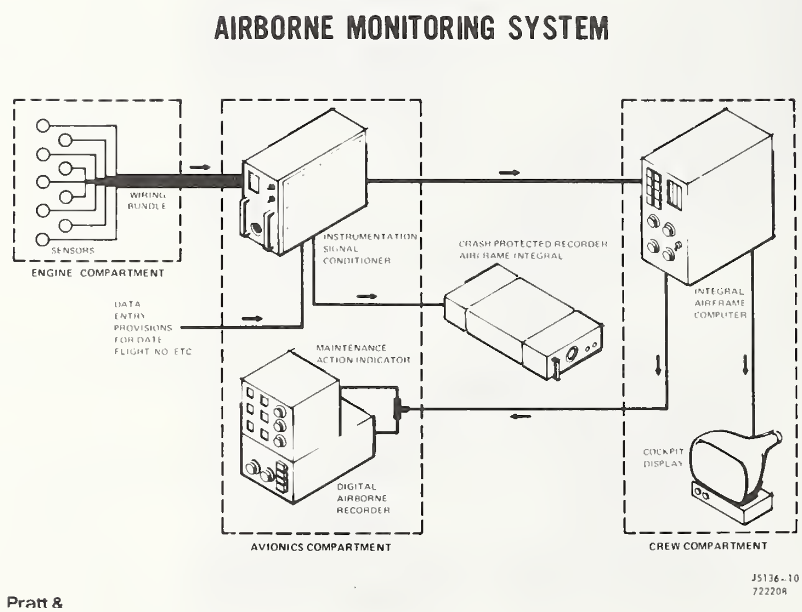


Figure 4

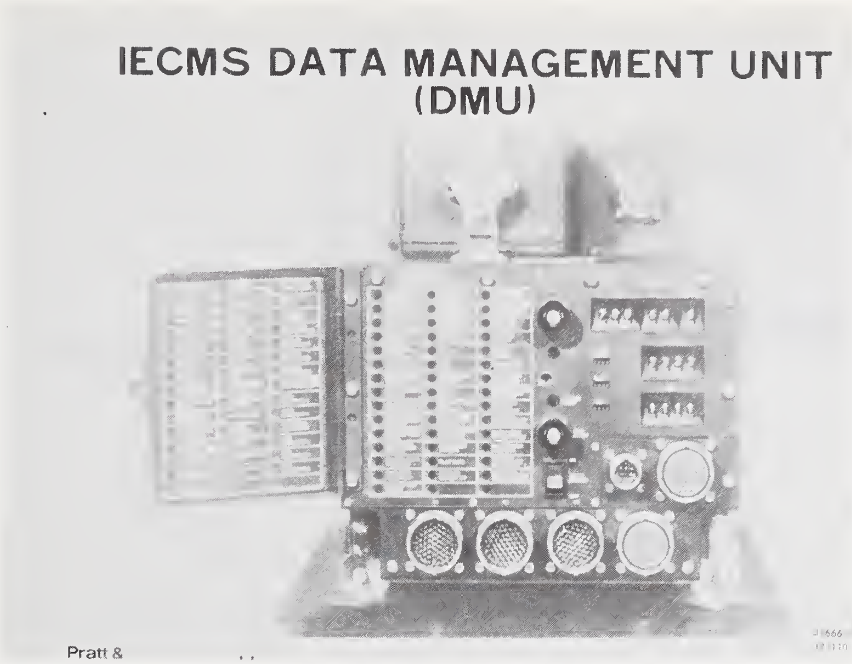


Figure 5

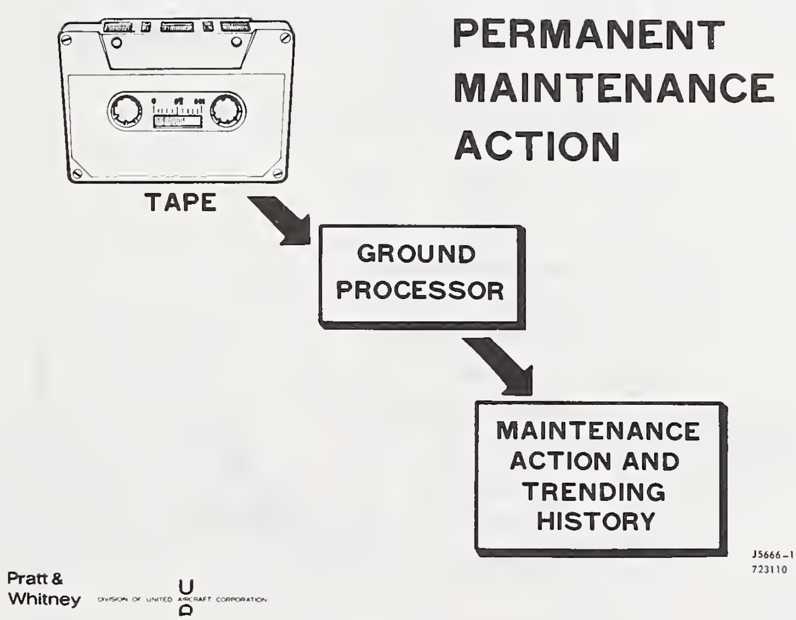


Figure 6

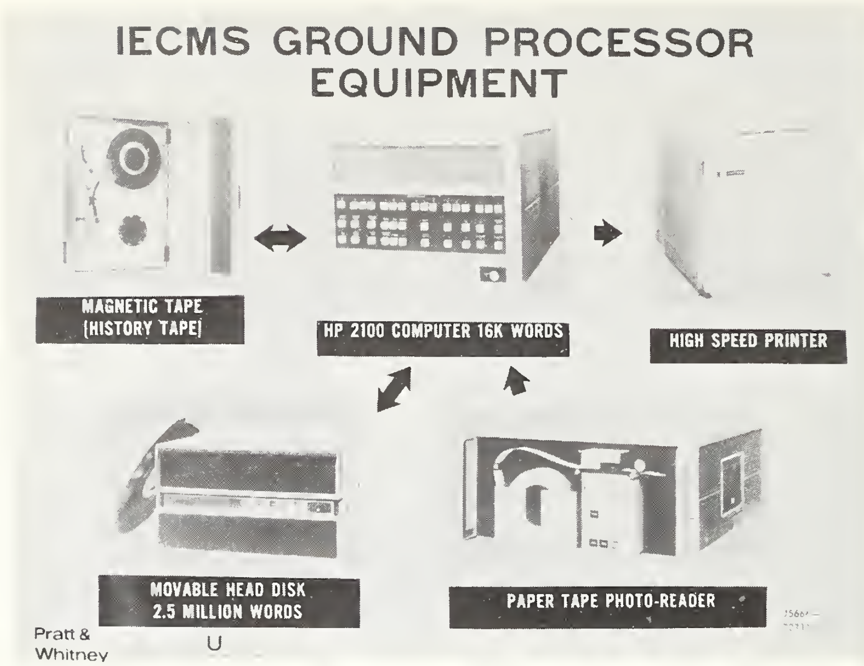


Figure 7

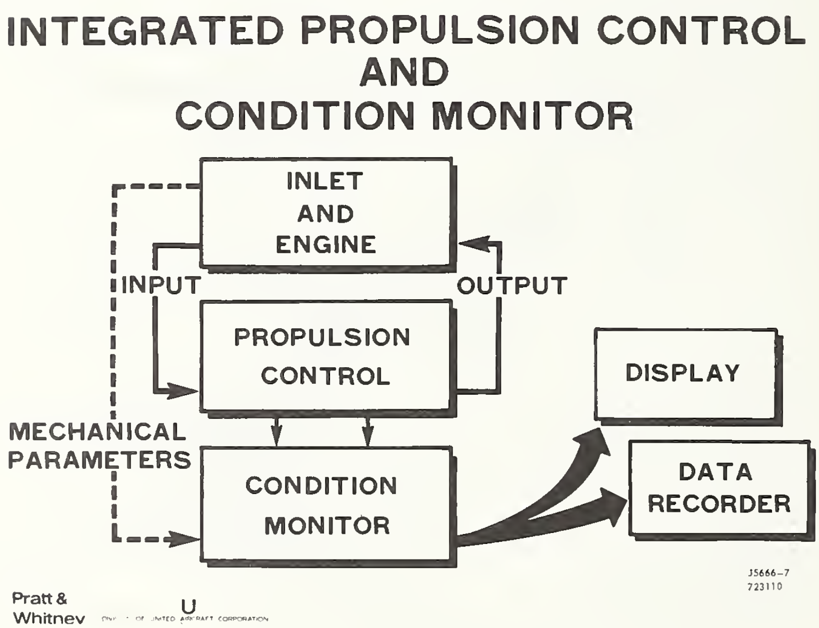


Figure 8

AN AUTOMATIC ENGINE CONDITION DIAGNOSTIC SYSTEM FOR GAS TURBINE ENGINES

by

Murray Hoffman, Program Manager

Hamilton Standard
Division of United Aircraft
Windsor Locks, Connecticut

The operator of gas turbine and/or reciprocating engine - powered equipment naturally requires that his equipment has the longest possible operating life and does not break down unexpectedly. To assure that this type of operation is realized, he applies maintenance procedures to minimize the rate of equipment deterioration and monitors its operation closely to detect the need for adjustment, additional maintenance or overhaul. Without this monitoring, engines which are deteriorating are very likely to run until secondary damage occurs necessitating a major overhaul. However, these procedures for maintenance and monitoring are difficult to apply effectively and efficiently and consequently early engine warning signs are inadvertently undetected.

Hamilton Standard's extensive experience with these systems has led to the development of an automatic engine condition diagnostic system for gas turbine engines called TRENDS^{TM}

TRENDS, an acronym for:

TRend analysis and

ENgine

Diagnostic

System

is an automatic real time engine condition diagnostic system that serves as a maintenance tool for gas turbine engines. TRENDS provides an engine condition monitoring system that ensures the earliest possible detection of an impending loss of performance or malfunction, performs a diagnosis, provides a prognosis and indicates the corrective action required. It remains in operation continuously, 24 hours per day, seven days a week even though the engines may not be operating. It is a completely independent system in no way interferring with the operation of the gas turbine generator system. Its successful operation, or failure, cannot directly affect the engine. TRENDS is a maintenance tool which provides significant timely information which may or may not be immediately acted upon. It does not replace the annunciator alarm system.

The primary tasks of TRENDS are to give adequate warning of the approaching need for maintenance action, be it routine maintenance or maintenance necessitated by deterioration in plant condition. The warnings are issued as English language instructions printed out on a "tear-off" strip printer. No data reduction or evaluation is ever required by the operator. The warnings are repeated at intervals until the problem has been rectified or, in the case of routine maintenance, the operator acknowledges, by means of push buttons, that he has implemented his instructions.

To enable TRENDS to generate these warnings it must monitor engine system condition. This is done by means of a number of sensors, many already installed on the engine as part of the basic control and alarm systems. Temperatures, pressures, vibration levels, oil quantity, speeds and other parameters are continuously monitored in real time. Then, by the utilization of a number of different techniques such as gas path analysis and limit comparisons against truth tables, the condition of the engine is constantly evaluated.by the TRENDS computer.

In addition to its primary tasks, TRENDS provides the operator with a direct printout of the data being monitored. This STATUS printout contains the last steady state set of data showing each measurement in engineering units and corrected for ambient conditions. Also included is a printout of all unexecuted maintenance instructions. It is available on demand by merely pressing a button marked "STATUS PRINT OUT" which is conveniently mounted on the front of the TRENDS cabinet. At periodic intervals, set at the option of the user, TRENDS will print out for record or logging purposes. The printout will also include every maintenance instruction flagged and completed since the last record report.

The monitoring and evaluation techniques used are those which have been developed over the past years by the Flight Operations Section of Pratt and Whitney Aircraft and the Engine Performance Group of Turbo Power & Marine Systems, Inc. These techniques were semi-automated by Hamilton Standard in their AIDS (Aircraft Integrated Data Systems) programs and TRENDS takes it the next logical step to a fully automatic real time TRENDS detection and diagnostic system.

Hamilton Standard decided to demonstrate the utility of these techniques by a practical demonstration which would also provide first hand experience of in-the-field operation. A demonstration was therefore arranged jointly with Turbo Power and Marine Systems, Inc. and Northeast Utilities at the Hartford Electric Light Co.'s (HELCO) South Meadow facility. The field site is shown in Figure 1.0-1. A TRENDS unit is installed on both gas turbine engines of a Twin Pac Generator System. Figure 1.0-2 shows the TRENDS unit installed in Control House 12.

Now, after more than 3000 hours engine time on a pilot plant installation at Northeast Utilities' HELCO station in Hartford, this "early-warning" diagnostic system can reduce down-time, through controllable improvements in operating efficiency, and in-service detection and correction of incipient failures. As a result, action can be taken before gas turbine performance or component life is compromised.

Over the last 18 months, during which HS has been fine tuning the equipment at the HELCO installation, diagnostic monitoring has succeeded in detecting a number of problems before they became evident to station operators - and has determined the maintenance actions required to correct them.

<u>FOULED COMPRESSOR.</u> Variations in the turbine outlet temperatures indicated the compressor was fouling at an unacceptable rate and should be cleaned as soon as possible. The compressor was subsequently cleaned without first making an inspection,

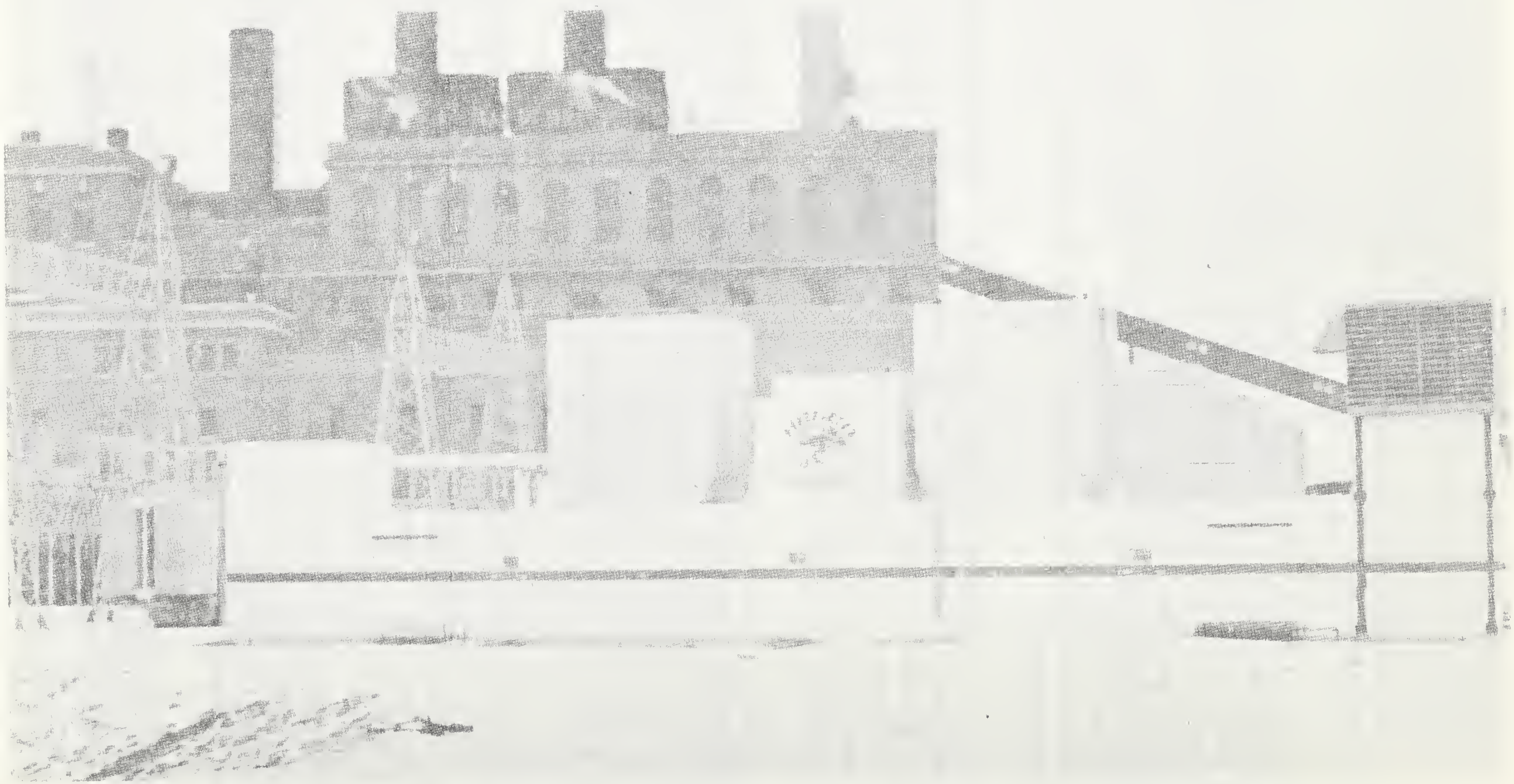


FIGURE 1.0-1 SITE OF TRENDS DEMONSTRATION

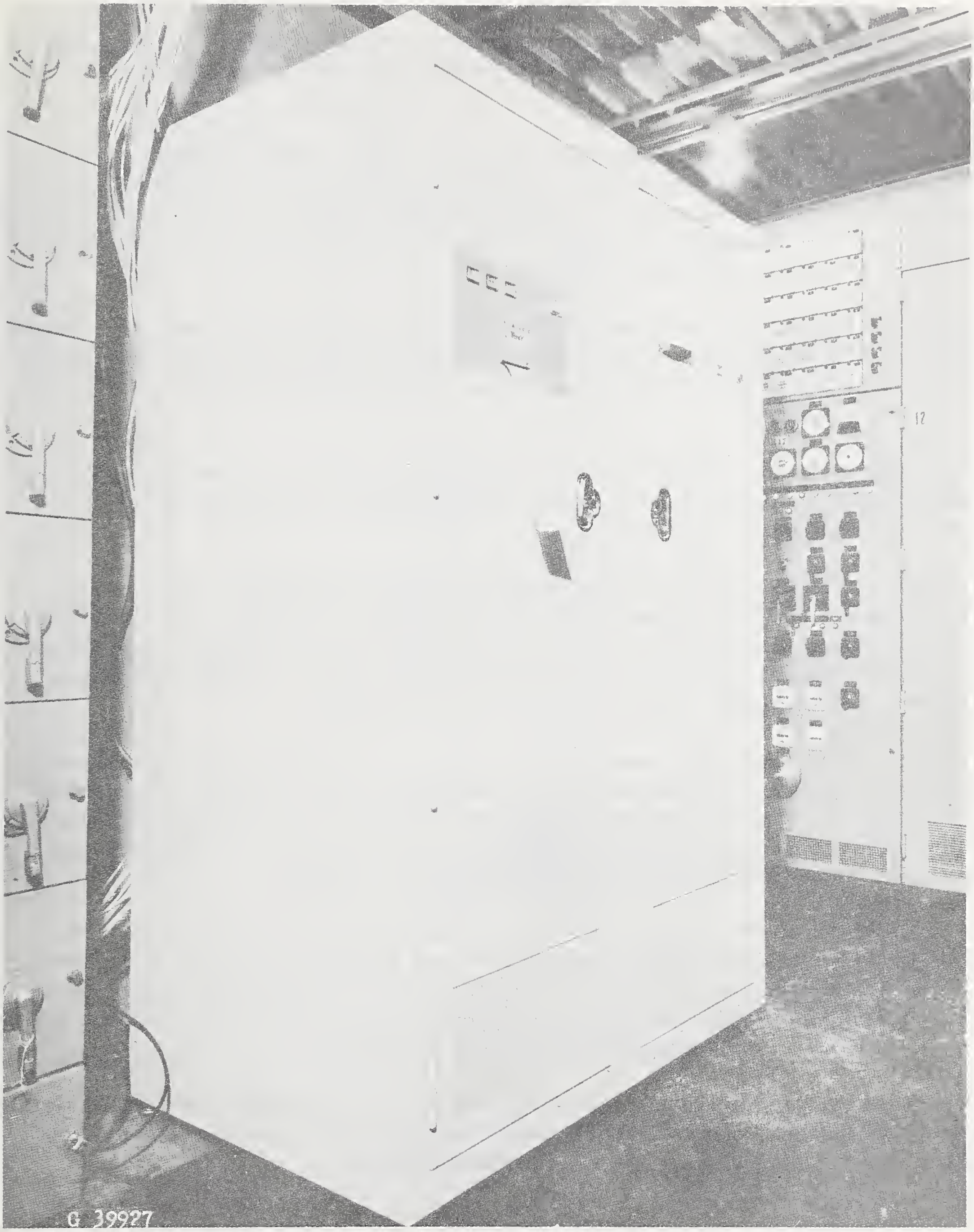


FIGURE 1.0-2 TRENDS DEMONSTRATION UNIT (INSTALLED IN CONTROL HOUSE AT DEMONSTRATION SITE)

and new data showed the turbine was restored practically to its baseline performance.

About 2 weeks later, data indicated the compressor was again fouled. This time an inspection was accomplished (to verify the condition) and confirmed the warning. The compressor was again cleaned.

BOWED NOZZLE VANES. Monitored data showed the compressor needed cleaning and that there were signs of "bowed first stage nozzle vanes." A hot section inspection confirmed the fouled compressor but visually, at least, the nozzle vanes looked acceptable. Just to make sure, however, the crew measured the vane angle and found they were bowed, but still within acceptable limits.

PRESSURE BUILD-UP. Data showed a "rapid increase in breather pressure in the gas generator oil system" which called for an inspection. Station operators shut down the turbine, found the anti-spark filter was clogged, cleaned and replaced it. Pressure was restored to normal.

LOW OIL LEVEL. At another time, the data showed a dangerously low level of oil in the reservoir. Station operators shut down the turbine, checked the oil level, and found the system needed replenishing.

TURBINE BLADE PROBLEM. Data showed a strong trend towards fouling of all turbine stages which was followed by a small step change in gas generator vibration and an increase in scatter of data. A hot section inspection confirmed a blade fatigue crack which required removal and repair of the turbine wheel. A significant failure might have evolved if this situation had been left undetected.

Normally, with the possible exception of the damaged turbine blade, those problems would have eventually been detected by a maintenance crew or resulted in an automatic engine shutdown. The early warning, however, prevented any rapid degradation in performance and made it possible to schedule minor maintenance actions at system convenience before situations could develop into major problems.

The TRENDS diagnostic system is modular in construction and flexible to allow for growth in the number and type of measurements and the number and type of engines. A single central computer provides the core of the system. Basically, there is a Central Unit (CU) for each total system, a Data Collection Unit (DCU) for each grouping of one to two engines, and a set of sensors on each engine.

The sensors are used to convert pressure, temperature, oil quantity, and speed signals to electrical signals. Other electrical measurement signals are taken directly from existing instruments. Figure 1.0-3 is the block diagram.

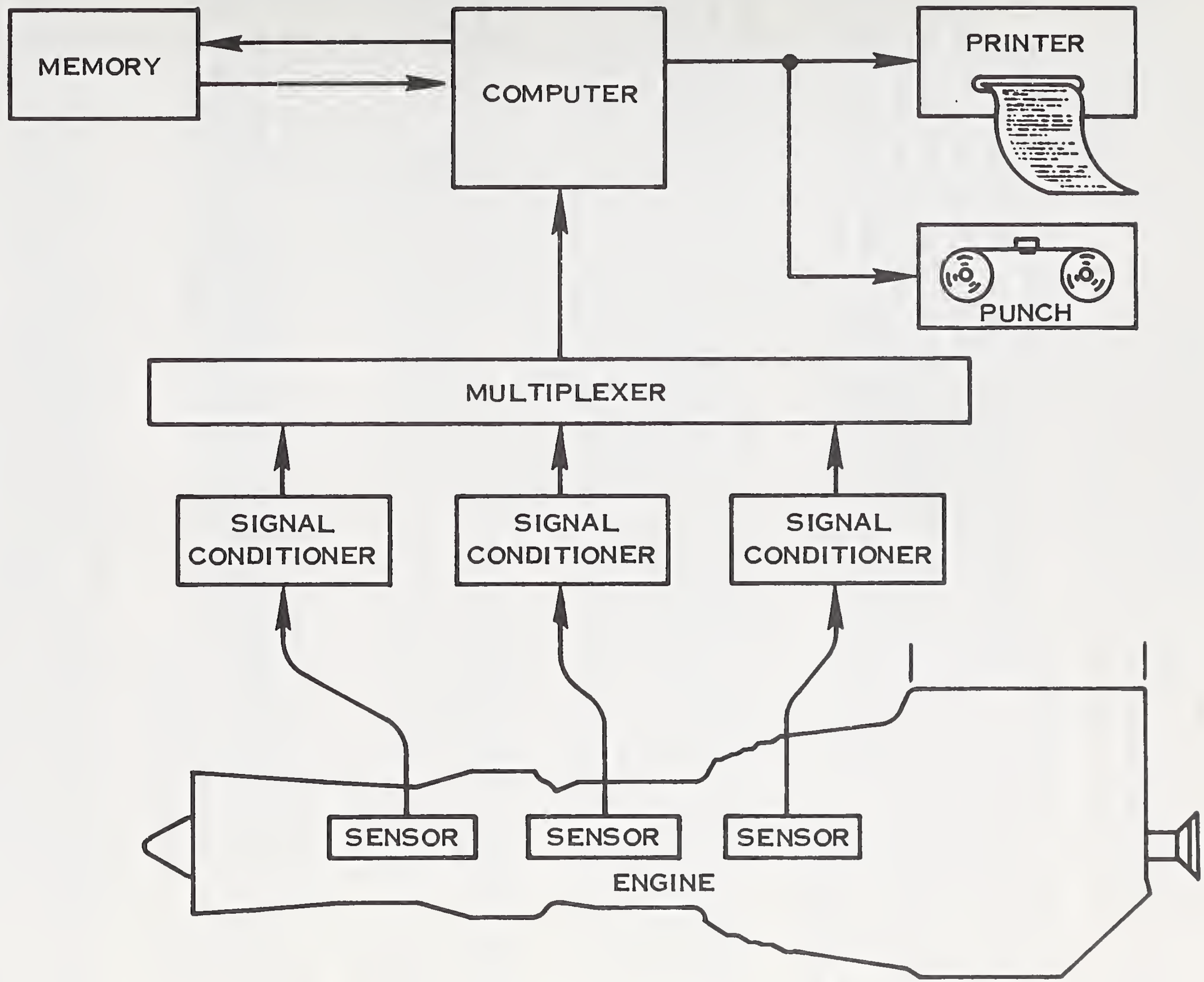


FIGURE 1.0-3 TRENDS SIMPLIFIED BLOCK DIAGRAM

The DCU collects the electrical signals, conditions them and converts them into a form suitable for transmitting in sequence along common cables to the CU. A DCU will handle up to 2 engine installations. Each engine can have up to 56 measurements. Additional engines necessitate an additional DCU or DCU's. Each DCU feeds directly into the CU, and the CU can handle up to 24 DCU's (48 engines).

The minimum size TRENDS system could consist of one DCU, one engine set of sensors, and one CU. The capability of the computer is such that a single CU can handle up to 48 engines in a single installation over direct wire; or they can be connected to the CU without geographic or wire length limitations through the integral modems permitting direct communication over telephone lines or telemetry links. The output of the DCU is a digital code, programmably compatible with all present digital transmission systems, which, theoretically, can be sent around the world.

Sensors mounted on the gas turbine engine monitor a variety of operating functions including compressor inlet pressure and temperature (PT2 and TT2); compressor discharge pressure (PS4); gas generator output pressure (PT7) and turbine outlet temperature (TT7); low and high pressure compressor rotor speeds (N1 and N2); compressor and turbine vibration levels; lube oil system pressures, temperatures, and levels; electric generator bearing temperatures; Figure 1.0-4 and Figure 1.0-5.

Periodically, about once every two seconds, the DCU samples all the sensor inputs from each engine (takes about 1/2 second per engine), converts the analog signals into digital form, and transmits them to the CU where they are sampled 16 times and then averaged to reduce the effect of any spurious inputs to arrive at a "true" reading. These 16-value averages are temporarily stored for data processing and emergency retrieval.

This constantly updated "retrieval storage" provides a history of engine performance immediately prior to any engine failure or abnormal deviation in performance. The computer memory retains a 5- to 6-minute record by continually recording every set of averaged parameters, which it does by deleting the oldest or first set of readings each time a new set is added.

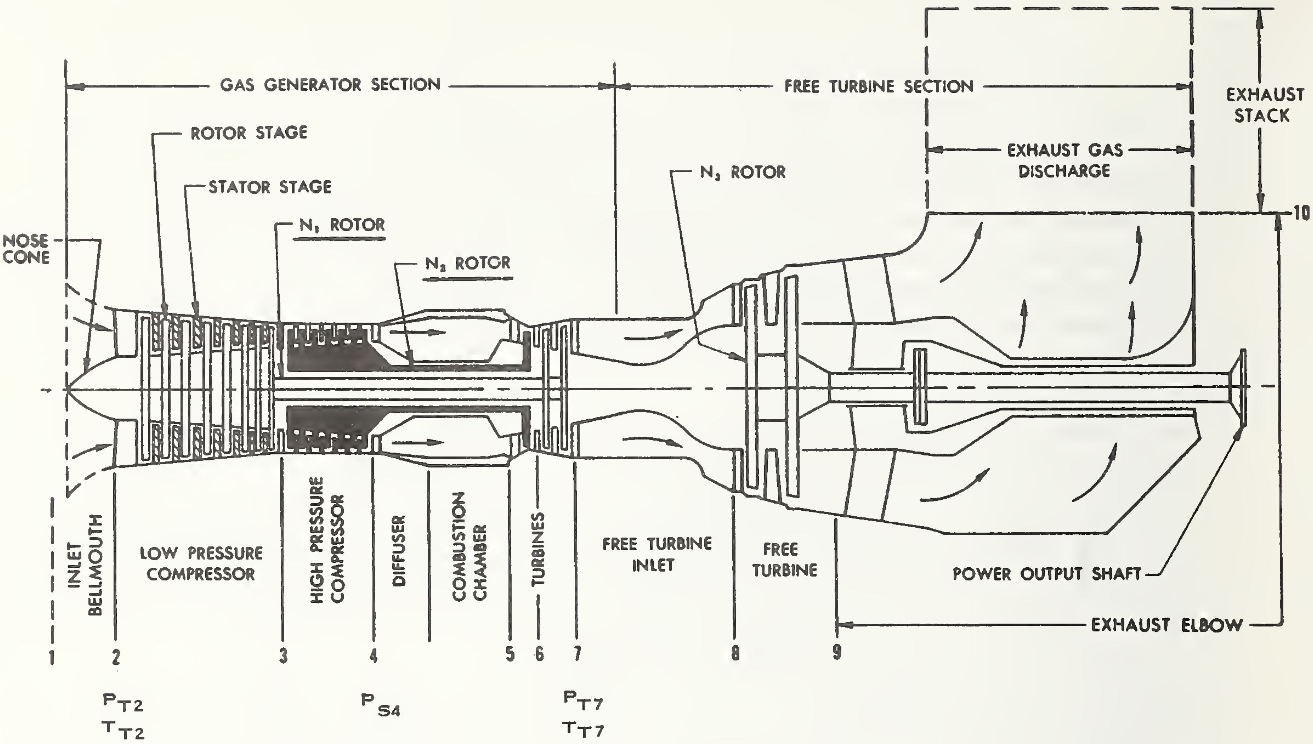
The permanent memory storage contains the baseline values of the various gas turbine parameters being monitored. Baseline information includes fixed data such as operating limits as well as variable data which varies between engines and with engine time. This data is obtained from initial performance readings taken when an engine is first installed new or following overhaul.

MEASUREMENT LIST

Time
Megawatts
Total engine operating hours.
Number of engine starts.
L. P. compressor speed
H. P. compressor speed
Compressor discharge static pressure.
Gas generator exhaust temperature
Engine inlet total pressure.
Gas generator discharge total pressure
Steady state operation.
Engine inlet temperature.
Ambient pressure.
Oil temperature to gas generator.
Oil temperature to free turbine.
Oil breather pressure gas generator.
Oil breather pressure free turbine.
Oil filter pressure drop gas generator.
Oil filter pressure drop power turbine.
Oil level gas generator.
Oil level free turbine.
Oil consumption gas generator.
Oil consumption free turbine.
Vibration Station #1
Vibration Station #2
Vibration Station #3
Anti-Icing
Generator bearing temperature coupling and exciter ends.
Generator cooling air temperatures in and out.
Exciter air temperature out.

FIGURE 1.0-4

MEASUREMENT LIST



SUFFIXES 1 THRU 9 SHOW INSTRUMENTATION STAGE

N	=	ROTATIONAL SPEED
P_T	=	TOTAL PRESSURE I.E. STATIC PRESSURE PLUS DYNAMIC HEAD
P_S	=	STATIC PRESSURE
T_T	=	TOTAL TEMPERATURE

FIGURE 1.0–5 SIMPLIFIED MECHANICAL SCHEMATIC

Each engine is assigned a small electrically alterable memory module, provided as part of the TRENDS hardware, in which is stored all the information relevant to that serial number engine such as baseline details, engine hours, regression analysis information, serial number, etc. This memory is plugged into TRENDS' DCU so that it can be interrogated by the computer to transfer its information to the main core memory.

Each engine is also fitted with a coded socket which is connected to the DCU so that the computer can ascertain the serial number of the engine and hence interrogate the correct electrically alterable memory module. Thus, when an engine is changed, TRENDS automatically detects the new coded socket and searches for the new engine memory module.

With a new engine, or one that's just been overhauled, it's necessary to "calibrate" the engine to generate the baseline information entered into the memory module. This involves running the engine at steady state for at least 1/2 hour at various calibration power settings so that the computer can monitor the running at each condition and enter the calibration readings as baseline inputs into the memory module.

The baseline values are used as reference points for comparing "as is" versus "new" component performance when evaluating on-line service. The computer subtracts the baseline readings from the sensor readings to generate "delta" values which represent the current amount of change from the installed or baseline value. Depending upon the parameter, the delta values are given as plus or minus percentage figures or in engineering units.

Trends in performance are determined by a method of linear regression analysis which calculates the slope (m) and intercept (b) of the straight line which defines the measured deviations from baseline over a period of time. Two trend lines are used to show the rate at which changes are taking place. There's a long term trend based on measured deviations recorded during the last 150 hours of engine operation, which gives an overall view of what's happening, and a short term 20-hour determination which magnifies the effects of more immediate changes. Figure 1.0-6.

Thus, with a single point defined by m, b, each trend line, it's possible to store data that would otherwise occupy literally thousands of bits of memory storage for each parameter monitored. This is the reason TRENDS is able to match the data processing capabilities of much larger and more expensive computers -- and also makes it possible for TRENDS to keep tabs on as many as 48 different turbine installations.

Knowing the slopes and intercepts of the trend lines, it's possible to predict when deviations from the baseline will become so large as to warrant maintenance action. Whenever the short term "m" deviates from the long term value by more than set

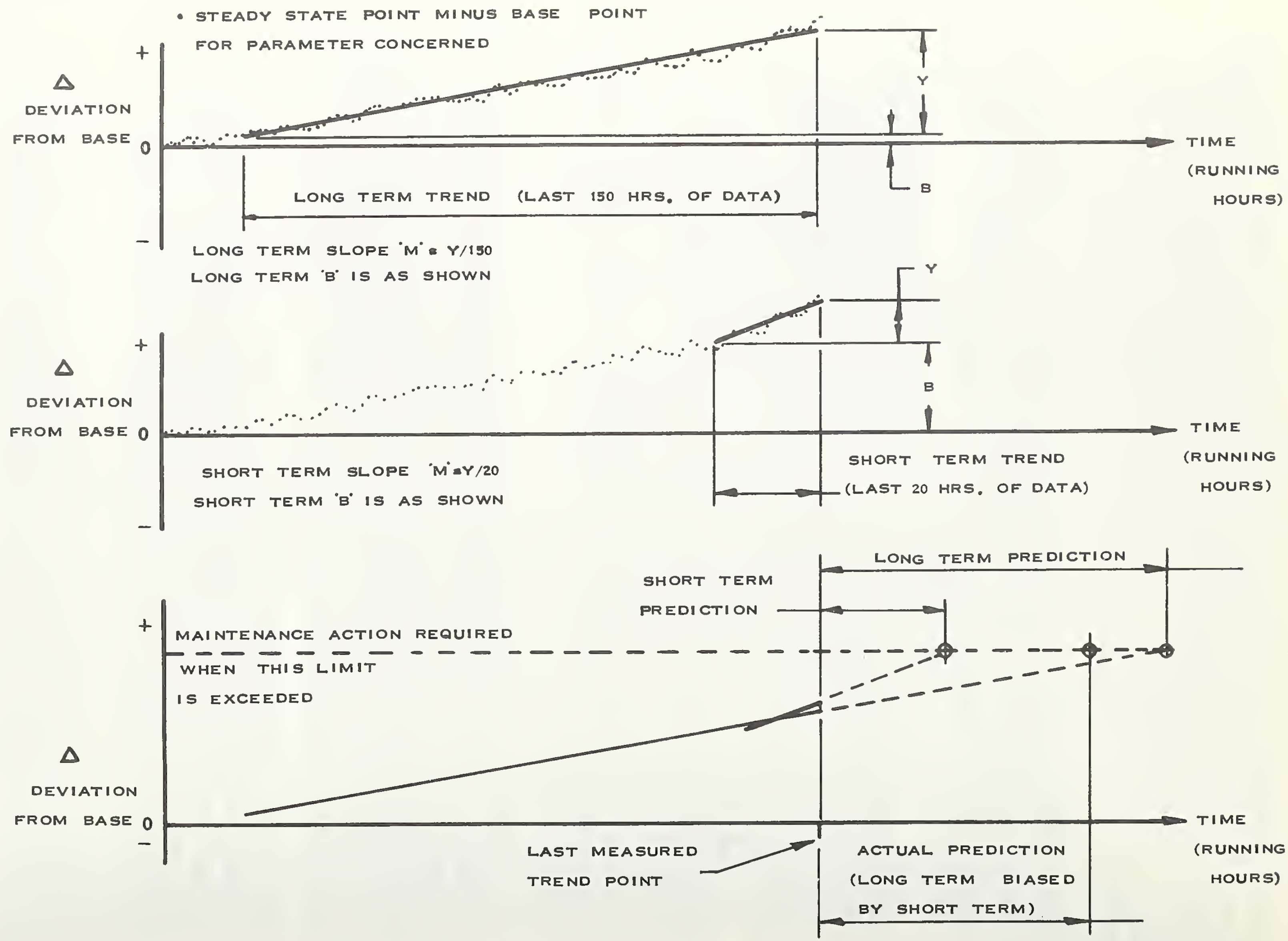


FIGURE 1.0–6 TRENDS LINEAR REGRESSION TECHNIQUE

limits, it implies a change in the rate of deterioration which is taken into account when predicting the time available for maintenance action.

Two other characteristics which the computer looks for are sudden changes in the short term "b" relative to the long term value, and a sudden or progressive increase in the randomness of the data collected. Either of these changes, suggests a fault other than wear or deterioration. In such cases TRENDS gives an alarm, prints out instructions for remedial action, and records the time of the message. Diagnostic "truth" tables are used to program the computer so that it can analyze the pattern of deviations to arrive at the most likely cause or causes. Figure 1.0-7.

Basically, TRENDS generates four kinds of outputs:

<u>FAULT MESSAGE.</u> Whenever the continuously performed trend analysis shows the need for a corrective maintenance action within 75 hours or less, the computer prints out a brief "fault message" which identifies the problem and gives instructions for its correction. This can happen at any time of day so the system is designed to flash a red light on the recording console to attract attention and ask for an acknowledgement by the station operator.

Until action is taken, TRENDS will repeat the message every 8 hours (without the flashing red light) unless performance begins to deteriorate at a faster rate calling for more immediate action. It steps up the message intervals to every 4 hours when only 24 hours are left, and then every 2 hours from 10 hours before the predicted event until 10 hours after it has passed. Figure 1.0-8.

<u>STATUS REPORT.</u> A daily status report is printed out for each engine installation automatically, at a set time of day, which shows the value of each measurement being made, the number of operating hours to date, and the "deltas" or deviations between baseline and current performance values. It also lists any outstanding maintenance actions waiting to be performed as well as all maintenance actions completed since the last daily printout was issued. Figure 1.0-9.

This same status report will also be generated upon request at any time. All you have to do is press the "status" button on the recording console for an immediate printout of the latest data being recorded and processed.

Many gas turbine station superintendents, particularly those with large fleet responsibilities or with remote station installations find the daily status report a very attractive operational feature.

<u>WARNING ALARM.</u> Whenever an operating limit is exceeded, for any of the parameters being monitored, the system prints out the last 5 minutes of recorded data for all the parameters to provide a history of what led up to the problem immediately before it was detected. In addition, the system also prints out a relevant fault message and instructions for remedial action.

	T_{77} INCREASED FROM LAST CLEAN	N_1 INCREASING	N_2 INCREASING	N_1 REDUCING	N_2 REDUCING	PS4 REDUCING	PS4 NO CHANGE	VGG HIGH	GG DIFF P HIGH	FT DIFF P HIGH	T_{OIL} GG HIGH	T_{OIL} FT HIGH	VFT HIGH	T_{OIL} GG NORMAL	T_{OIL} FT NORMAL	MOUNTS LUBED	GG CONS. HIGH	FT CONS. HIGH	GG LEVEL LOW	FT LEVEL LOW	P_B GG HIGH	P_B FT HIGH	T_{77S} HIGH	NO ANTICE COMMAND	VALVE OPEN	VALVE NOT OPEN	VALVE CLOSED	VALVE NOT CLOSED	ANTICE NEEDED	$T_{AMB} > 57$ F	NACK	$T_{AMB} \leq 57$ F	TGB HIGH COUPLING END	TGB HIGH EXITER END	DTG HIGH	DTE HIGH
CLEAN COMPRESSOR	●																																			
CLEAN COMPRESSOR		●																																		
CLEAN COMPRESSOR			●																																	
BOWED FIRST STAGE	●			●	●	●																														
FOULED TURBINE OR WORN SEALS	●			●	●		●																													
INSPECT HIGH TURBINE BLADES	●			●	●			●																												
INSPECT LOW TURBINE BLADES	●		●	●				●																												
CLEAN OIL FILTER GG									●																											
CLEAN OIL FILTER FT										●																										
INSPECT GG BEARING								●			●																									
INSPECT FT BEARING												●	●																							
LUBRICATE MOUNTS								●						●																						
LUBRICATE MOUNTS													●		●																					
INSPECT COMPRESSOR BLADES								●						●		●																				
CHECK GG OIL LEAK																	●																			
CHECK FT OIL LEAK																		●																		
FILL OIL GG																			●																	
FILL OIL FT																				●																
CHECK OIL SEALS GG																					●															
CHECK OIL SEALS FT																						●														
INSPECT BURNER																							●													
ANTICE NO POWER																								●												
ANTICE STUCK OPEN																								●	●			●								
ANTICE STUCK CLOSED																										●	●		●							
ANTICE STUCK MIDWAY																										●		●								
ANTICE ERROR																													●	●						
CHECK ANTICE AUTO																															●	●				
INSPECT GEN. BEARING																																	●			
INSPECT GEN. BEARING																																		●		
INSPECT GEN. COOLING																																			●	
INSPECT EXITER COOLING																																				●

NOTES—
1. A LIMIT IS SET FOR EACH CHANGE GIVEN IN THE VERTICAL COLUMNS.
2. ALL ITEMS CHECKED HORIZONTALLY MUST BE PREDICTED TO EXCEED THEIR SET LIMIT WITHIN 75 HOURS BEFORE A MESSAGE IS FIRST GENERATED.

LEGENDS—
- T_{77} AVERAGE EXHAUST GAS TEMPERATURE
- T_{77S} SPREAD OF EXHAUST GAS TEMPERATURE SENSED
- N_1 LOW COMPRESSOR ROTOR SPEED
- N_2 HIGH COMPRESSOR ROTOR SPEED
- PS4 PRESSURE AT STATION 4
- VGG VIBRATION GAS GENERATOR
- VFT VIBRATION FREE TURBINE
- GG GAS GENERATOR
- FT FREE TURBINE
- T_{OIL} LUBE OIL TEMPERATURE
- T_{AMB} AMBIENT TEMPERATURE
- P_B OIL BREATHER PRESSURE
- NACK NO ACKNOWLEDGEMENT OF THIS MESSAGE
- TGB TEMPERATURE GENERATOR BEARING
- DTG RISE IN AIR TEMPERATURE THROUGH GENERATOR
- DTE RISE IN AIR TEMPERATURE THROUGH EXCITER

FIGURE 1.0–7 TYPICAL DIAGNOSTIC TABLE

```
A GG  CLEAN          -74
    COMPRESSOR
ACK CODE    1 3
TIME 306  16   12   56

A GG  INSPECT       -49
  BOWED FIRST STAGE
ACK CODE    3
TIME 306  16   12   56

A GG  IN PECT       -71
    FOULED  TURBINE
    OR   ORN  SEALS
ACK CODE    1 2
TIME 306  16   12   55

A TIT               -30
ACK CODE    2
TIME 306  16   12   55

A SFC               -41
ACK CODE    1
TIME 306  16   12   55
```

FIGURE 1.0–8 TYPICAL FAULT MESSAGES

TIME	306 16	12 56
MW	49.95	50.67
TEH	20.	20.
NS	41.	5.
ART	0.3	4.0
EPR	2.309	2.435
N1	5792.	0.
N1C	5752.	0.
N2	8257.	0.
N2C	8199.	0.
PB	173.08	122.45
PBC	12.029	8.515
TT7	966.4	480.8
T7C	1405.8	927.1
T7S	1.1	2.0
PT2	29.288	29.277
PT7	38.21	41.86
SS	10.	7.
TAM	66.67	66.67
PAM	29.433	29.433
OTG	0.0	83.3
OTT	0.0	0.0
OPG	9.52	37.73
OPT	11.01	42.13
DPG	8.63	23.78
DPT	7.53	24.41
OLG	16.09	7.66
OLT	16.05	3.73
VG	0.000	1.447
VT1	0.000	1.412
VT2	0.000	1.406
DMWC	0.00	0.00
DN1C	- 0.7	- 3.3
DN2C	- 1.7	-49.1
DPBC	- 3.1	4.4
DT7C	3.6	-34.3
DT7S	-111.8	-85.0
DOTG	13.5	-56.2
DOTT	-140.0	22.9
DOPG	-14.91	8.53
DOPT	-18.41	5.27
DDPG	- 1.38	16.88
DDPT	- 1.16	17.10
DOLG	6.34	- 1.75
DOLT	3.05	- 9.68
DVG	0.499	- 0.941
DVT1	0.673	- 0.781
DVT2	0.972	- 0.484
DSFC	2.46	-37.09
DTIT	2.31	-51.84
DCEL	- 1.65	141.15
DFT	- 1.68	-87.24
DGFS	2.18	-26.34
DAFL	- 0.03	35.53

NOTE: TWO COLUMNS INDICATED DATA FROM EACH OF TWO ENGINES OF A TWIN PAC INSTALLATION.

FIGURE 1.0-9 TRENDS STATUS REPORT

While the engine continues to operate above limits, the acknowledge button glows red. At any time the operator can press the "status" button on the recording console for a printout of the last set of 16-second average points.

MAINTENANCE MESSAGE. As required, the system issues routine preventive maintenance instructions for the station operators and maintains a status report on what's been completed and what is outstanding at any given time. Schedule frequency and message content are tailored to the requirements of the site, and may be based on engine running hours or elapsed time. Figure 1.0-10.

Each maintenance message is identified by an "acknowledgement code" which provides a link between the station operator and the computer. Whenever the system issues a maintenance message, it records the time, the action to be taken, and when it's to be completed. Whenever the maintenance is completed, the station operator enters the acknowledgement code into the computer as a signal that it has been performed and should be removed from the next day's status report.

In looking at what TRENDS can do, the questions loom large in the minds of users: How effective is it in covering the major problems most commonly experienced during turbine operation? And, how accurate is it in predicting and diagnosing these problems?

TRENDS can successfully anticipate at least 75% of the gas turbine problems now being experienced in industry while they are still in the formative or developing stage--early enough to be able to take remedial mesaures and avoid an unscheduled shutdown. Figure 1.0-11 is a tabulation of the estimated accuracy of fault prediction.

```
180 DAY PREVENTIVE
    MAINTENANCE
ACK CODE    1 2
TIME 227  13  28  28

  CHECK FIRE EXT
ACK CODE    2
TIME 227  13  28  27

  CLEAN BREATHER
ACK CODE    1
TIME 227  13  28  27

  CHECK EXHAUST
ACK CODE    1 2 4 5
TIME 227  13  28  27

  CHECK INLET
ACK CODE    2 4 5
TIME 227  13  28  26

  AIR STARTER
  MAINTENANCE
ACK CODE    1 4 5
TIME 227  13  28  26

  LUBRICATE FUEL
  SYSTEM  MOTOR
ACK CODE    4 5
TIME 227  13  28  26



 90 DAY PREVENTIVE
    MAINTENANCE
ACK CODE    1 2 3 5
TIME 227  13  28  25

  CHECK ALTERNATOR
  OVERSPEED
ACK CODE    2 3 5
TIME 227  13  28  25

CHECK FUEL  FILTER
ACK CODE    1 3 5
TIME 227  13  28  24



 60 DAY PREVENTIVE
    MAINTENANCE
INSPECT COMPRESSOR
ACK CODE    3 5
TIME 227  13  28  23
```

FIGURE 1.0–10 ROUTINE MAINTENANCE MESSAGES

ESTIMATED ACCURACY OF FAULT PREDICTION

The reliability of TRENDS to spot trouble before it shows up as an unacceptable deterioration in performance or component failure can range from as low as 25 to 30%-where, by the time the symptoms appear, it's too late to take corrective action--up to 100% for classically perdictable failure patterns.

Item	Technique	Accuracy
Blocked/Streaky Nozzles or Damaged Cumbustor	gas path analysis	100%
Fouled Compressor Blading	temperature and speed	100%
Bearing Seal Deterioration	pressure and temp	80%
Bowed First Stage Nozzles	pressure and temp	80%
Fouled Turbine Blading	speed and pressure	70%
Damaged Compressor Blade	vibration and temp	30%
Damaged Turbine Blade	vibration and temp	30%
Bearing Problems	vibration and temp	25%
Anti-Icing Valve	switch position	100%
Oil Filter Blockage	pressure	100%
Excessive Oil Consumption	reservoir level	100%

FIGURE 1.0-11

ESTIMATED ACCURACY OF FAULT PREDICTION

The TRENDS diagnostic system provides the following action messages as rapidly as performance changes are detected and evaluated.

MESSAGE	REASONS FOR TAKING RECOMMENDED ACTION
Clean Compressor	Increased EGT lowers the life of the hot section. Increased specific fuel consumption increases operating costs.
Inspect-Fouled Turbine or Worn Seals	Increased EGT lowers the life of the hot section Increased specific fuel consumption increases operating costs.
Inspect-Bowed First Stage	TRENDS detects this problem, which is the start of hot end deterioration, at an early stage and thereby enables the cost of repair to be minimized. The problem is self aggravating in that it causes increased EGT's which in turn increases the rate of hot end deterioration.
Shutdown No Anti-Ice Power	Continued operation of the engine could result in compressor damage caused by ice collecting on the intake, breaking away and entering the engine.
Inspect-Anti-Ice Valve Stuck Open	Increased specific fuel consumption increases operating costs. Unable to obtain maximum power on warm days.
Shutdown Anti-Ice Valve Stuck Midway	Anti-icing system only partially effective. Compressor damage could occur in icing conditions. In non-icing conditions the increased specific fuel consumption would cause higher operating costs. On hot days maximum power would not be available.
Override Anti-Ice Signal Error	Increased specific fuel consumption increases operating costs. Unable to obtain maximum power on warm days.
Check Auto Anti-Ice	The engine is operating in icing conditions with the anti-icing system manually switched off. This could result in damage to the compressor by ingested ice.

MESSAGE	REASONS FOR TAKING RECOMMENDED ACTION
Clean Oil Filter	The forewarning given by TRENDS enables the maintenance action to be scheduled. Clogged filters result in lowered oil pump inlet pressures and have an adverse effect on pump life.
Inspect for Bearing Failure	Bearing failures are progressive leading to major overhaul. Early repairs result in major cost savings.
Inspect Compressor Blades	TRENDS detects possible compressor blade problems at an early stage enabling the cost of the repair to be minimized.
Inspect Turbine Blades	Early detection and repair of turbine problems enables cost of repair to be minimized. Damage may extend to free turbine if allowed to progress.
Lube Mounts	Excessive vibration reduces life of engine and components. Vibration caused by badly lubed mounts could mask vibration caused by more serious potential problems. Early warning enables maintenance action to be scheduled.
Check Oil Leak	Loss of oil increases operating costs. Leakage may be to a hazardous area or create a hazard. Avoid unscheduled shutdown due to low oil level.
Fill Oil	Adding oil when the engine is off line avoids an unscheduled shutdown due to low level alarm.
Inspect Oil Seals	Bearing lubrication may become inadequate. Oil may be lost to exhaust causing infringement of exhaust emission regulations.
Inspect Burner Bad Tt7 Distribution	Uneven Tt7 distribution may be the result of a burner can problem. Damage to the nozzle guide vanes can result. A second probable cause is clogged or streaky burner nozzles. This can damage and distort burner cans and nozzle guide vanes. May also result in failures to start.

MESSAGE	REASONS FOR TAKING RECOMMENDED ACTION
Generator Coupler (or Exciter) Bearing Overhot Inspect Generator Over Hot-Inspect Exciter Overhot-Inspection	Avoid possible damage to generator by appropriate maintenance action taken before problem develops.

The TRENDS™ Diagnostic System is designed to provide users of gas turbine engines with an automatic data analyzing tool which will:

* IMPROVE ENGINE MAINTENANCE PRACTICES, which will tend to:
* Reduce maintenance costs,
* Reduce operating costs,
* Extend engine life, and
* Reduce forced outages

The TRENDS diagnostic system is "on-line" continuously, providing maintenance action messages as rapidly as conditions change and are evaluated. The high accuracy and short cycle sampling times detect changes in performance at their very onset.

This early maintenance action, taken when required, should result in less costly repairs and as procedures improve, resulting in better maintained engines, maintenance action requirements will be reduced. In addition, the probability of damage due to secondary failures is greatly minimized. This increased visibility into the engine condition and the improved maintenance actions permit confident "on-condition" maintenance procedures which can maximize time between inspections. It permits scheduling maintenance actions at the convenience of the user and reduces the probability of unscheduled outages.

The automatic and remote capability of the TRENDS system also provides detailed reports without the requirement of an on-site operator.

As recommended by the manufacturer, routine maintenance tasks are necessay to keep the engine operating at optimum performance and to prevent premature wear-out. It is important that they be performed on schedule. The TRENDS diagnostic system provides a print-out when the actions are due and provides a record of when they were executed. Keeping the engines operating at or near peak efficiency for most of their life avoids the over-stress conditions which sometimes lead to excessive and expensive repairs, reduced life, and unscheduled outages.

The real worth of TRENDS is the knowledge and confidence the station operator has in knowing that his equipment is operating properly and is available for immediate power demands. In addition, the savings associated with scheduled repair and maintenance and reduction of expensive failures make TRENDS a cost effective investment.

A SYSTEMS ENGINEERING APPROACH TO

EFFECTIVE ENGINE CONDITION MONITORING

David W. Leiby *

Aircraft Engine Group,

General Electric, Evendale, Ohio

ABSTRACT

The development and application of a variety of engine condition monitoring techniques and equipment is currently being accelerated as a result of increasing demands for improved aircraft flight readiness availability, improved mission success probability, and reduced maintenance costs. Although individual monitoring methods provide valuable information, optimum effectiveness can be achieved only through their application and utilization in an engineered system of complementary techniques. To achieve effective condition monitoring, therefore, an integrated system of airborne and ground monitoring, diagnostic, and inspection techniques and equipment applied to aircraft turbine engines for purposes of problem detection, isolation, and trend monitoring is required.

Currently, a wide variety of monitoring techniques and equipment is being applied to aircraft turbine engines. These range in maturity from well-established methods (such as borescope inspection, Spectrometric Oil Analysis Program, and radiography), through items still in early application evaluation phases (such as vibration analyzers, in-line oil monitors, and parameter trend monitoring), to advanced development techniques (such as turbine blade pyrometers and associated maintenance recorders). The purpose of this paper is to present the system engineering aspects of condition monitoring from an aircraft engine manufacturer's point of view.

Presented as Paper at the Symposium on Instrumentation for Airbreathing Propulsion, Monterey, California, September 19-20, 1972

* Systems Engineer, Conditioning Monitoring Systems

INTRODUCTION

The larger sizes and higher costs of today's aircraft turbine engines, the growing economic impact of flight and mission delays and cancellations, and increased costs of maintenance material and labor, have made sophisticated condition monitoring economically attractive. More and more attention is now being focused on the application of condition monitoring techniques and equipment.

A variety of different techniques and equipment has been developed. Instrumentation manufacturers have developed and improved their equipment to provide desirable monitoring capabilities in an effort to expand their markets; certain aircraft users reduced their operating and maintenance costs through the application of cost effective monitoring techniques and equipment; and engine manufacturers have developed and applied new and improved monitoring methods to enhance their capability to provide engines with lower costs to the user in current and future aircraft applications.

Condition monitoring must be more than a random collection of measurement, inspection, and diagnostic techniques and equipment applied to aircraft turbine engines. It must be an integrated system of complementary technology, which is intimately interfaced with the engine design, coordinated with the engine application and operation, and incorporated into the engine maintenance plan.

The purpose of this paper is to present a systems engineering approach to condition monitoring from an engine manufacturers point of view. The philosophy, history, and technical requirements are reviewed briefly for background information. The main body of the paper is devoted to a discussion of selected condition monitoring techniques and equipment, including the monitoring rationale, system description, and examples of its uses and applications.

BACKGROUND

DEFINITION AND PHILOSOPHY - For the purpose of this paper, condition monitoring is defined as that integrated system of techniques and methodology applied to aircraft turbine engines to achieve problem detection, fault isolation and trend monitoring. Through integration of the system with the engine design, its installation and application, and its maintenance plan, effective condition monitoring can be achieved.

The tools of condition monitoring vary widely in technology and in application, encompassing such disciplines as pneumatics, hydraulics, electronics, optics, mechanics, infra red radiation, thermodynamics, and gamma ray radiography. Some monitoring techniques are applied in flight, either full-time or at selected points in the flight profile. Others use in-flight recorded data with ground processing. Still others such as radiography and borescope inspection, are used only during ground inspections.

MONITORING EFFECTIVENESS - Effectiveness and accuracy of the overall condition monitoring system is enhanced by using the indication of one technique to complement and verify that of another. In line, full-time monitors may be used as early warning devices, with other techniques applied for confirmation, isolation, and detailed analysis.

Proper application, integration, and utilization of a system can result in significant cost effectiveness in aircraft operations. Economic benefits from increased operational reliability, and reduced maintenance costs can result. The early detection of engine distress, followed by adequate corrective action, reduces the incidence of unscheduled maintenance, flight delays or aborts, and inflight shutdowns. Maintenance costs can be reduced through the reduction of trouble shooting time, and by permitting more effective scheduling of maintenance personnel, equipment, and facilities.

HISTORICAL STATUS - Much of the basic technology currently being applied in condition monitoring has been in use for some time. The monitoring capabilities provided by cockpit instrumentation, ground testers, and non-destructive inspection are the basis for the development of more sophisticated monitoring techniques. The historical background of individual techniques will be discussed later.

Advances in electronics, such as miniaturization, high temperature capability, and higher reliability, have aided in the general advance of condition monitoring capability. Advances in condition monitoring have also been stimulated by increased knowledge of the mechanisms of failure, failure progression, and diagnostic logic. It is now possible to formulate diagnostic logic to provide more reliable indications of engine condition. Automated routines are now being developed which provide go/no-go types of indication, minimizing the role of the human operator in the diagnostic operation.

DESIGN INTEGRATION REQUIREMENTS

In the establishment of design interfaces between the aircraft gas turbine engine and the condition monitoring system, it is necessary to consider the technical requirements which influence system design.

ACCESSIBILITY - Access to the engine must be provided to place sensors at positions where desired parameters may be obtained. For those techniques where actual physical contact is required, such as magnetic plugs, borescope inspection, and radiography, access to the engine for the implementation of these techniques must be possible with a minimum of preliminary disassembly and removal.

ACCURACY - The accuracy of engine condition assessment depends directly upon the accuracy of the input data to the condition monitoring system. The sensor and its signal conditioner must be capable of accurately converting the parameter signal into an electrical input signal to the monitoring system.

ECONOMICS - The application of a specific condition monitoring technique, and the depth and detail in which it is applied, must be economically feasible. Suffice it to say, engine modifications and implementation of a specific condition monitoring technique cannot exceed the economic gains to be realized through the application of that technique. There is a danger that costs per engine would be prohibitive if condition monitoring techniques and equipment of insufficient maturity are utilized.

EFFECTIVENESS - In the case of techniques requiring sensed parameter data, the proper parameters must be available, and they must be sensed at their optimum location. For such techniques as borescope inspection and radiography, the access and extent of coverage must be sufficient to provide effective application of the technique.

MAINTAINABILITY - The sensors, instrumentation, and special engine features required to accommodate condition monitoring must be applied to the engine with ease of maintainability and calibration as design criteria. In addition, the maintenance features of the basic engine must not be compromised by the presence of condition monitoring.

RELIABILITY - Additions and modifications to the basic engine to accommodate the condition monitoring system must be selected, designed, and applied with high reliability as a goal. The integration of several complementary techniques to comprise a system will make it possible to segregate false indications of distress caused by failure within the monitoring system. Coincidentally, incorporation of the system into the basic engine must not degrade the reliability of the basic engine.

MONITORING TECHNIQUES

Fully effective condition monitoring capability on a particular aircraft gas turbine engine can be attained only through the coordinated integration of several monitoring techniques. This cannot be emphasized too strongly. Without such an approach, condition monitoring techniques utilized individually fall short of their potential impact upon engine operation and maintenance. Several of the more effective monitoring techniques are discussed in the following sub-sections, including monitoring rationale, historical background, system and equipment description, and current application status.

PARAMETER MONITORING - Trending of the performance of aircraft turbine engines through the periodic monitoring of selected engine parameters is one of the primary techniques used in condition monitoring systems.

Monitoring Rationale - The mechanical condition of an aircraft gas turbine engine can be deduced from engine thermodynamic cycle data. Selected pressure, temperature, flow, speed, and position measurements and their manipulation through proper logic routines can provide performance indices, efficiencies, and margins which can be trended to monitor rates of change of engine performance characteristics. Such changes are indicative of mechanical deterioration of the engine.

<u>Historical Background</u> - A form of trend condition monitoring has been the observation and comparison of sequential thermodynamic performance data from the pilot's or flight engineer's log. The concept has now been expanded by the application of more sophisticated data gathering and processing techniques. Using airborne data systems, parametric data are recorded at selected times and/or at selected points in the flight envelope, and are processed through trending logic on the ground. Activities are currently underway in several areas in which an airborne computer is applied to the performance monitoring system to provide real time trending of the engine during flight.

<u>Technique Description</u> - Trend monitoring of the relative operational health of an engine by way of measurements on major sub-assemblies and accessory systems (modules) requires the availability and processing of a number of engine parameters. Fortuitously, the majority of the parameters required for condition monitoring are also used in control functions or are displayed on cockpit instrumentation.

In performance monitoring of the thermodynamic air cycle of a turbofan engine for trending of overall engine health, and the isolation of degradation to the major engine modules, the following parameters may be used.

PLA	Power Lever Angle
VSVP	Variable Stator Vane Position
P2	Fan Inlet Pressure
T2	Fan Inlet Temperature
EGT	Exhaust Gas Temperature
NF	Fan Speed
NG	Core Speed
P54	LPT Inlet Pressure
WFM	Fuel Flow
P25	Duct Total Pressure
PS3	Combustor Static Pressure
T3	Compressor Discharge Temperature

The thermodynamic performance of the engine and its modules is typically indicated by corrected values of certain measured parameters and calculated performance indices or factors. Included among the output data are:

EGTK	Corrected Exhaust Gas Temperature
PCNFK	Percent Corrected Fan Speed
EPR	Engine Pressure Ratio
WFMK	Corrected Fuel Flow
ETAT	High Pressure Turbine Efficiency
ETAC	Compressor Efficiency
PCNK	Percent Corrected Core Speed
FANQ	Low Pressure Turbine Energy Function
T2CK	Corrected Compressor Inlet Temperature
P4Q54	High Pressure Turbine Pressure Ratio
W4CK	Corrected HPT Inlet Flow
FAR4C	Fuel/Air Ratio at HPT Inlet
DPL	Power Lever Angle Schedule Deviation
DSVP	Stator Vane Position Schedule Deviation

These output data are expressed either in engineering units or in dimensionless, normalized units based on a selected control point.

Application Status - Currently, these systems are undergoing application and routine operational evaluation at General Electric to provide monitoring of development and flight test engines. Long term trend monitoring is obtained off-line from recorded data through the Characteristic Analysis Trending Technique (CATT) program. The real time limit monitoring of the performance engines is being accomplished through the application of the Trend Analysis Digital System (TADS)

Characteristic Analysis Trending Technique (CATT)

This technique was developed to provide printed and plotted trending information to assist the engine user in monitoring relative engine thermodynamic health by major subassemblies. The CATT program provides trend plots of selected engine parameters by processing data from the most recent engine parameters, and incorporating this point into a plot of historical data extracted from the data file of that particular engine. Trending of a given parameter is accomplished by plotting the algebraic difference of the present value of the parameter and the same parameter on the original point associated with the current configuration of the engine.

The CATT program is applied to trending overall engine relative health based on EGT and component efficiencies at the take off power level. Analysis of flight test data from the DC10 has shown that the most effective monitoring data was obtained during the take-off roll, just prior to lift-off at 135 knots indicated airspeed, with a data point averaged from 4 samples per second per parameter for a 2 second duration. The selection of this point in the flight envelope and the averaging of 8 data points produced trending results with a minimum of data spread. A typical CATT trend plot of EGT is shown in Figure 1.

Trend Analysis Digital System (TADS) - General Electric's Aircraft Engine Group at Evendale, Ohio, has designed and built, and is currently utilizing in engine development testing, an on-line data acquisition system providing continuous monitoring and display of critical engine operating parameters. The Trend Analysis Digital System (TADS) consists of a cathode ray tube display (CRT), an ASR33 Teletype terminal, and a two-bay cabinet which houses an analog multiplexer, an analog-to-digital converter, digital magnetic tape unit, and a GE/PAC 30-02 Data Processor. Figure 2 shows a typical TADS equipment arrangement. The TADS presents on the CRT "real time" calculated engine parameters corrected to standard conditions to be used for engine safety monitoring while the engine is under test.

The TADS continuously scans forty-eight (48) engine parameters at four (4) samples per second per channel. The current complement of parameters includes four (4) positions, eighteen (18) pressures, eight (8) temperatures, ten (10) vibration signals, three (3) frequencies, and five (5) miscellaneous parameters. Eight samples are averaged over a two-second period, and passed through an error reject routine. The data processor performs, in real time, the conversion to engineering units, the calculation of critical engine parameters or performance indices, checking for out-of-limit values, and computes "delta" values for out-of-limit checks. The CRT display, which is updated every three seconds, presents five absolute values of critical engine parameters, six "delta" values of the parameters, and thirteen manual data items. The TADS will generate, upon request, a ten-point trend plot on the teletype, a printed hard copy of selected parameters, a punched paper tape of the averaged data, and a magnetic tape record of the raw engine data.

VIBRATION ANALYSIS - Since many turbine engine problems are characterized by an increase in vibration levels, considerable attention has been given to the detection and interpretation of vibration signals.

Monitoring Rationale - A gas turbine engine generates considerable acoustical and vibration energy over a broad frequency spectrum. The acoustical and vibration output is the composite of the driving forces of once-per-revolution of the main engine shafts, the blade passing frequencies, bearing rolling element passing frequencies, gear mesh frequencies, pump element frequencies, mechanical resonances, aerodynamic resonances and gas flow turbulance. By analyses of this complex waveform signal through the segregation of significant signal components from the background "noise", it is possible to monitor the changing mechanical condition of the engine in a particular area. A number of different techniques are applied to achieve this signal analysis.

Historical Background - Engine vibration has long been recognized as an indication of engine distress, and vibration sensors and display systems have been applied widely. Generally, these systems sense velocity or acceleration and display the vibration in mils of displacement in a frequency band which includes the once-per-revolution signal of the engine shafts over their operating range. However, by the time an engine has exceeded the vibration limits sensed by such a system, the engine distress may have progressed to such an extent that secondary engine damage has occurred and there is increased risk of unscheduled engine shutdown. Therefore, various attempts have been made to obtain incipient failure information at a much earlier point in the failure progression to provide an early warning of impending problems, and to allow timely corrective action.

Early vibration analysis techniques included methods using acoustical data obtained from microphones placed near the engine and vibration data from accelerometers mounted on the engine. These analysis techniques concentrated upon the problem of enhancing the desired signal, separating it from undesirable background signals, and interpretating the significance of the signal. Among the techniques applied were a variety of sophisticated pattern-recognition-procedures, including digital and analog methods. Narrow band tracking filters have been applied to improve signal-to-noise ratios of selected frequency signals.

The complexity and cost of analyses equipment, the high skill level of data acquisition and reduction personnel required, the marginal signal-to-noise-level operation, and requirements for tuning the system to a particular engine posed serious limitations to their universal application. As a result, many vibration monitoring methods which could demonstrate a certain level of effectiveness on prototype laboratory demonstrations, run into various problems in attempting to fulfill the requirements in the market place of practical applications.

Technique Description, Implementation, and Application - The detection of aircraft engine distress through the application of vibration analysis techniques requires both the availability of data-rich signals and equipment capable of processing and analyzing them. Since the engine operates in a strong acoustical and vibration noise environment, it has been found that best results are obtained through the use of data from accelerometers close-coupled to the areas of interest.

The application of vibration monitors using waveform pattern recognition techniques requires broad-band vibration input data. Since frequency components up to 15,000 hertz are significant in the vibration signal, sensors and sensor locations suitable for once-per-revolution AVM systems may no longer be sufficient.

Internal Accelerometers - Accelerometer/vibration system development programs are currently being accomplished, with the cooperation of hardware vendors, to improve hardware reliability, reduce cost, and improve quality of data obtained from vibration equipment and monitoring sensors. Changes in accelerometer design, connector design, cable bending capability, routing and clamping procedures and elimination of connectors have been introduced to improve accelerometer systems. The current internal accelerometer is an integral, stainless steel-sheathed, hard-lead unit, secured with thru-bolts, preferably directly to engine hardware. A connector is provided as an integral part of the hard lead immediately outside the engine casing.

FOD Detection - Automatic detection of Foreign Object Damage (FOD) and Domestic Object Damage (DOD) is oriented toward the detection of impacts which fall between those causing no damage and those causing such severe damage that long lasting disturbance of other parameters, stalls, or shutdowns occur.

The impact of an object on the rotating parts of an engine releases measureable energy in the form of large, fast-rising transients in the output of an accelerometer, hard mounted internally, near the rotor thrust bearings. These transients are detected and compared to a preset limit by the FOD detector. Figure 3 shows the test cell model of the FOD detector.

Bearing Monitor - Several types of bearing defects such as race spalling, rolling element spalling, end wear and foreign material damage, provide suitable signals for automatic detection.

The latter is accomplished by the recognition of invariant signal patterns produced by internal, bearing-housing-mounted accelerometers. The measurement used to identify the patterns is called "Impact Index", and can be continuously measured with the bearing monitor. Figure 4 shows the test cell model and the airborne version of the monitor. Normal bearings of all engines monitored to date have produced impact indices of 2.0 to 3.0. When a defect develops, the index is driven past 3.0, and increases as the severity increases.

Mass Unbalance Monitoring - Engine rotor mass unbalance can be caused by many different malfunctions arising in an engine. To detect this unbalance condition, amplitudes of vibration in the rotor rotational frequency range are measured and compared to either fixed limits or on a trend basis. Monitoring systems for this "Once-per-rev" signal in the past have suffered from lack of sensitivity, extreme noise and general lack of useability to detect engine problems.

In the Mass Unbalance Monitor (MUM) system currently under evaluation, the combination of the MUM unit with internally mounted accelerometers is expected to give greatly increased capability for unbalance monitoring. The unit is essentially a high-response, digitally-controlled, narrow-band, synchronous, digital filter. The response is sufficient to track the engine through throttle bursts and chops. With the greatly increased sensitivity of the internal sensors, a very powerful means of detecting unbalance problems is available.

Gearbox Monitoring - Defects arising in gearboxes, such as gearshaft end bearing wear, spalls, high clearance, eccentric gears and bent gearshafts, may be detected by measuring a fixed pattern characteristic of individual gear meshes. The measured characteristic is called the "Modulation Index". This index increases as the defect appears, and continues to increase proportionally to the defect severity.

OIL SYSTEM MONITORING - Lubricating oil, in addition to its functions as a lubricant and coolant, acts as a carrier of information as it collects and circulates wear particles, dirt, and oil breakdown products. When properly monitored, valuable data can be obtained on the condition of oil-wetted parts. Numerous methods have been developed and applied to sense and interpret oil condition and its associated engine components, including magnetic chip detectors, acid number, conductivity, viscosity, spectrometric analysis, indicating filter screens, and light transmission/scatter techniques.

Monitoring Rationale - Generally, an engine's oil consumption rate and its oil system capacity are such that lubricating oil dwell time in the engine, on the average, is only a few hundred hours. This eliminates oil breakdown and loss of lubricity as oil monitoring factors. The main concern, therefore, is the accumulation of failure material from the oil-wetted parts.

There are two functions required of the oil monitoring system. First, it must detect the presence of wear material in the lubricating oil, and must detect this material at such a threshold that the proper maintenance action can be scheduled before there is serious risk of unscheduled shutdown or possible secondary damage occurs. The second function required of the monitoring system is the capability for indicating the area of the engine in which the distress is located. This feature is necessary to allow timely maintenance action to take place without undue trouble shooting to locate the problem area. Therefore, effective oil system monitoring requires both problem detection and fault isolation capability.

The value of the engine condition information which can be obtained from the oil system depends upon both the effectiveness of the data and the ease and convenience by which the data can be acquired. In the integrated system approach to oil system monitoring, two or more methods are applied in a complementary manner. A continuous, in-line oil monitor can be used to provide an alarm that metallic and/or non-metallic particle concentration has increased beyond established limits. With this early warning, magnetic chip detector and filter screen checks can be scheduled, and an oil sample can be obtained for spectrometric oil analysis.

Historical Background - A variety of techniques and equipment have been applied to the monitoring of oil systems during recent years. Many of these applications have been addressed to problems peculiar to specific engine models. One of the techniques which has been applied extensively is the magnetic chip detector. Magnetic chip detectors are comprised of permanent magnets mounted in the oil stream, with provisions for easy removal for inspection. The progression of engine distress can be assessed by trend monitoring the magnetic material buildup on the plugs. This technique does have some limitations in that it cannot detect non-ferrous material, and requires access to the engine for physical removal and inspection of the plugs.

Another oil monitoring method which has been successfully applied to a number of different engines is the Spectrometric Oil Analysis Program (SOAP). In this technique, samples of oil are obtained periodically from the engine and the constituant elements in the oil are spectrographically determined. This method is very sensitive, and can detect the presence of excessive wear long before the distress has progressed to a condition of significant risk. The system does have its limitations, however, in that fault isolation is limited, and the logistics of the sampling, shipment, and laboratory analysis may cause delays between the taking of the sample and availability of results.

Applications - One currently-applied, in-line oil monitor operates on the principle of the infra-red light scattering and light transmission characteristics of the oil. The unit is normally installed in the oil line between the oil tank and oil supply pump, with total oil flow passing through the sensor. Figure 5 shows the installation of such an oil monitor in the oil supply line of a turbofan engine.

The condition of the oil is trend monitored by periodic indications from the in-line oil analyzer. The indicated data must be interpreted with the proper recognition of the dilution effects of new oil additions. When the measured level of particulate matter in the oil exceeds the established alarm limit, additional oil system data should be obtained from the examination of the magnetic chip detector and oil filter, and the withdrawal and processing of an oil sample.

The growth of distress in the wear surfaces of the oil-wetted parts is accompanied by the build-up of particulate matter in the oil stream. Large chips and debris accumulate on the pump inlet screen; smaller materials are caught by the oil filter, and the small particles (smaller than the filter mesh, such as 40 microns) continue to circulate and build up in the oil system, resulting in an increased signal level from the in-line oil monitor.

A promising development in oil system monitoring is the indicating oil screen. This screen, which is constructed of a special composite of conducting and non-conducting mesh material, provides a method for remote electrical determination of the amount of build-up of electrically conductive debris collected on the oil screen. Figure 6 shows a prototype model of such a screen for use in the scavenge line input of a large turbofan engine. The application of such indicating screens should allow problem detection, as well as fault isolation.

However, the required size of particle for detection may reduce the capability of this technique to detect engine problems early in their growth progression.

<u>BORESCOPE INSPECTION</u> - A variety of optical devices are available for the visual inspection of the internal components of aircraft turbine engines. Either a glass optics system housed in a rigid cylindrical barrel, or a flexible fiber optics bundle is used to bring the optical image external to the engine. Illumination means include an external light source through a fiber bundle, internal incandescent lamps, and internal gas discharge arc lamps. An integrated system comprised of a borescope and illumination means, a low light level television camera, and a video tape recorder has recently been assembled and used successfully. The value of this system will be determined in time.

<u>Monitoring Rationale and Interface Requirements</u> - Many of the symptoms of deterioration of engine internal components can be readily assessed by visual inspection. The presence and magnitude of nicks, tears, rips, rubs, erosions, burns, cracks, buckles, distortions, coating removal, deposits, etc. in the compressor, combustion, and turbine areas of the engine provide very significant and accurate bases for maintenance decisions. The basic requirements of a borescope inspection system is to provide an acceptable external image of the internal engine parts of the engine. The quality of the image is determined by its magnification, resolution, and brightness; it defines the scope of

the manual observation and photographic documentation of the borescope inspection.

The photographic capability of a borescope inspection system is an important aspect of its application. The old adage that "a picture is worth a thousand words" is certainly true in this case. The analysis of successive borescope photographs of the same area of the engine is an effective method for trend monitoring distress progression.

The effectiveness of borescope inspection of an aircraft engine depends upon the capability of the basic borescope system to extract a useful image from the engine, and the access provisions for inserting the borescope into the areas to be inspected. These two criteria are related, since the engine access ports set the maximum diameter of the borescope, thus establishing its resolution, illumination, light gathering power and viewing distance. In a typical large turbofan engine, the access ports are sized to accept a 0.400 inch diameter borescope. This is the maximum diameter that can be reasonably accommodated between stator vanes at the high pressure stages of the compressor, through the cooling holes of the outer combustion liner, and between the airfoils of the first stage high pressure turbine nozzle diaphragm.

To meet the requirements for borescope inspection, four sets of access ports are incorporated into the turbofan engine. Access provisions are incorporated in the compressor case for access between the stator vanes of each stage to allow the inspection of the trailing edges of the upstream rotor blades, and the leading edges of the downstream rotor blades as the core engine is slowly turned over. In the combustor area, six access ports are approximately equally spaced around the circumference of the engine. These ports provide access for inspection of fuel nozzles, combustor swirl cups, all areas (forward, middle, and aft) of the inner and outer combustor liner, and the leading edges of the first stage, high-pressure turbine nozzle. The two stages of high pressure turbine blades are inspected through access ports in each of the two turbine nozzle diaphragms. The low pressure turbine blades are inspected through access ports in the respective nozzle diaphragms of each stage.

Historical Background - Borescope inspection of aircraft gas turbine engines has continued to grow in effectiveness as improved equipment has become available, and inspection access has been designed into the newer engines. Early borescope applications used the adaptation of medical diagnostic optical instruments, and suffered accordingly from the dissimilar environments of the operating room and the aircraft flight line.

One of the continuing problems of the older borescopes was that of obtaining sufficient light inside the engine to provide an adequately bright image. Among the methods used were miniature "grain of wheat" incandescent bulbs operated over-voltage to get maximum light, with a resulting short bulb life, and the possibility of bulbs shattering inside the engine. Another method utilized a special gas discharge lamp at the tip of the borescope. This provided adequate illumination for visual and photographic inspection, but presented its own set of problems, such as a complex and expensive power supply, high voltages in the system, and a hot exposed bulb.

Continued progress has been achieved by working with borescope manufacturers to design special equipment tailored for aircraft engine application, with simplicity and ruggedness high on the list of design priorities.

Borescope Equipment and Applications - Periodic borescope inspection is a generally accepted standard operating procedure. Borescope inspection data provides early warning of distress in the engine and allows a scheduling of maintenance before the distress grows to the point where unscheduled shutdown occurs. The engine inspection requirements specify such borescope features as depth of penetration, viewing angle, field of view, illumination intensity, magnification, and object distance.

A current borescope system being applied to a large turbofan engine uses a rigid optics train for image transmission, and a glass fiber bundle for illuminating light transmission. The basic system consists of a number of different probes, each having a selected angle of view and magnification, a light source, and the interconnecting glass fiber light bundle. The power supply provides two light sources, one an incandescent projector bulb for continuous operation, and a 1000-watt gas discharge lamp for intermittant use in photography. The light bundle is protected by an armored sheath and an outer plastic coating.

An important aspect of borescope inspection is its capacity for pictorial trend monitoring, in which the progressive growth of distress is observed, and predictions are made as to when distress has progressed to its allowable limit. This requires storage, and comparison of sequential borescope inspection data. In one method of pictorial trend monitoring, a 35mm SLR camera is used to obtain black-and-white or color photographs through the borescope. For obtaining quick turn-around on the availability of borescope photographs, a Polaroid pack attachment is fitted to the back of the 35mm camera body. These two photographic attachments are shown with the borescope probe in Figure 7. Excellent borescope photographs have been obtained of many areas of different engines. Figure 8 and Figure 9 illustrate some of the typical areas of the engine which can be photographed successfully through the borescope.

Since the exposure time for a given photograph is determined by the combined effects of film type, borescope optical system losses, engine area reflectivity, object distance, and illumination intensity, borescope photography is often an art, rather than a science. However, recent equipment developments and the availability of such equipment have removed the guesswork, and made possible the obtaining of properly exposed borescope photographs under the diverse combinations of variables encountered in the application. The device which accomplishes this feat is the through-the-lens light meter with booster amplifier, which automatically adjusts the shutter speed for actual light reaching the film. With such a camera system, the inspector need only select the ASA setting for the film being used, and the camera will automatically expose the film correctly, covering a range of a few milliseconds to up to 70 seconds.

The application of a lightweight, low-light-level, television camera to the borescope has opened new avenues of borescope inspection capability. A prototype system used in the application evaluation of the concept is shown in Figure 10. The borescope is coupled to the vidicon TV camera through a two-stage image intensifier. The output of the system is displayed on a conventional video monitor and can also be recorded on video tape.

The borescope-video system provides a real-time image of the internal area of the engine which can be viewed simultaneously by a number of people. In addition, it brings the image to a more comfortable area for study, since many borescope inspections require awkward and tiring positions for manual, direct viewing. The use of a video recorder can provide a record of each blade of a turbine stage having up to 100 blades with an elapsed time of several minutes. Still photographs of each blade would require up to four hours. Another very important feature of the borescope-video-recorder system is its use as a training aid. Through the generation of typical sequences of engine images it is possible to train inspectors in the implementation of standardized inspection techniques.

RADIOGRAPHY - The application of radiographic inspection techniques to condition monitoring of the internal components of assembled, on-wing engines has increased considerably in recent years. In this technique, a gamma ray source is used to expose industrial x-ray film placed on the appropriate area of the engine. Three different techniques are applied in obtaining radiographic data. In the major number of radiographic inspections, a radial technique is applied in which the source is placed along the engine centerline, and the film is placed on the surface of the engine. A reverse radial technique is sometimes used in which the film is placed at the engine centerline, and the source is placed on the surface of the engine. In the third technique, chordal shots are obtained by placing the film on one side of the engine and the source on the other.

Monitoring Rationale and Interface Requirements - The engine areas amenable to radiographic inspection include compressor stator vanes, the combustor and fuel nozzles, the high pressure turbine nozzle diaphragms, the transition ducts, and the low pressure turbine nozzle diaphragms. Excellent correlation has been obtained between radiographic inspection data and teardown inspection.

Radiographic inspection is applied to aircraft engines during periodic inspections, or whenever there is reason to suspect engine distress amenable to detection by this technique. The results of radiographic inspection are used as the basis for maintenance action decisions. Limits are established for different types of distress, such as length, location and number of cracks in the combustor, and amount of bowing of first stage high pressure turbine nozzle diaphragm airfoil trailing edges. Trend monitoring of the results of sequential radiographs of a particular area of the engine allows determination of the deterioration rate, and prediction of the remaining operating time before the distress will reach the limit value.

Radiographic inspection of aircraft gas turbine engines requires design and configuration features compatable with radiographic inspection technology. First, convenient access to the centerline of the engine is required. Second, the surface of the engine cases and frames should be free of obstruction to allow optimum film placement. And third, the basic design of the engine must be amenable to radiographic inspection, with areas of interest capable of clear radiographic projection on film fan visualization.

Historical Background - X-ray inspection techniques have been used in the routine, non-destructive testing of airframe and aircraft engine components for many years. Portable x-ray equipment and gamma ray sources have been applied to the inspection of airframe structures and equipment to obtain information on the condition of these structures and components without removal and disassembly. However, in the case of aircraft turbine engines, the application of radiographic inspection techniques has been a more recent development.

Among the technical advances which have contributed to the development of radiographic inspection technology was the advent of the depleted uranium shielded source container which provided up to 100 curies of Iridium - 192 in an easily portable 40-pound container. The current generation of large turbofan engines with their easily accessible hollow shafts and "clean" engine surfaces have made the application of radiography practical. The commitment by General Electric of several man-years of effort exclusively to the development, refinement, and application evaluation of radiographic inspection has provided the basic knowledge for establishing a firm inspection technique.

From this development program, a system of Radiographic Inspection Procedures has been produced, which are documented in the format of a Non-Destructive Inspection Manual. Also, a kit of special support equipment has been designed for the application of this technique.

Applications - Radiography has become an accepted tool in the maintenance plans of many aircraft engine applications. It is being used routinely to monitor the condition of development and cyclic endurance engines in test cell operations, as well as of flight test engines. The same techniques and equipment are being applied on-wing by airline personnel to monitor the condition of selected engine areas on revenue aircraft during periodic inspections.

In a typical radiographic inspection of a large turbofan engine, the source tube is inserted along the centerline of the engine either at the fan spanner, or through the center vent tube at the rear. Figure 11 shows the source tube being installed in the engine. The film cassettes are held on the surface of the engine by an elastic rubber strap, as shown in Figure 12, and provide total circumferential coverage of the area of interest except where access is prevented by engine structures.

Figure 13 shows a typical radiograph of the first stage, high pressure turbine nozzle diaphragm area. With an ideally exposed film read on a high intensity viewer, considerable data on the condition of this area of the engine can be obtained. These include the presence and size of platform cracks, trailing edge distortion or bowing, burning or erosion of the platform, and airfoil segment weld integrity.

In the chordal radiographic inspection technique, the film cassette is placed against the engine surface at an angle, and the source is placed at the opposite side of the engine. The gamma rays pass through the area of interest and impinge on the cassette perpendicular to the plane of the cassette. Figure 14 shows a chordal radiograph of the interface penetration at the aft end of the combustor and the 1st stage high pressure turbine nozzle diaphragm.

OTHER MONITORING TECHNIQUES - Several monitoring techniques will now be discussed which are at various stages of development and which show potential for inclusion in the list of useful condition monitors.

Infrared Turbine Blade Pyrometer - One of these advanced techniques, which is rapidly maturing into operational status, is the infra-red turbine blade pyrometer. In the pyrometer system, an infrared detector is mounted on the turbine case, and oriented to allow the sensor to scan along the blade as it passes under the pyrometer. The transient response of the pyrometer system is sufficiently high to provide the temperature profile along each individual blade for a blade passing frequency of up to 20,000 per second. The simplest form of data interpretation is the gross average temperature of the turbine blades. Pyrometers are currently being used in development test engines to evaluate blade cooling design effectiveness. A pyrometer measurement system is being designed into the control system of an advanced engine.

Dynamic Pressure Monitoring

Dynamic pressure monitoring techniques are being developed to utilize the high frequency components of gas and liquid streams of the engine. Special sensors and diagnostic methods are being applied to the assessment of pump condition, valve operation, and gas path flow characteristics.

Maintenance Recorder

The maintenance recorder concept is being pursued in an effort to obtain a system of monitoring methods which will provide an indication or measure of the consumed life of the engine. By the integration of such concepts as operating severity modeling, cycle counting, and time-at-temperature, a measure of the influence of thermal cycling and low cycle fatigue upon the expected life of engine parts can be estimated. The goal of the maintenance recorder is to provide a maintenance factor, made up of the combined, weighted factors of engine duty which can be used as a guide in scheduling detailed inspections of the engine, or estimating remaining service life before risk of engine problems become significant.

CONCLUSIONS AND RECOMMENDATIONS

This paper has been restricted to a review of the basic condition monitoring techniques, their design interface with the engine, and their current application status. The achievement of an effective operational condition monitoring capability requires additional interface considerations, which are beyond the scope of this paper. These areas include coordination with the airframe design for the transmission of data from the engine, data processing, data recording and data display. Other important interfaces are the integration of condition monitoring into aircraft operations, and integration into the engine maintenance plan.

A broad overview examination of condition monitoring shows that it has come a long way in approaching its true potential, with demonstrated condition monitoring techniques directly applicable to gas turbine engines. However, there still is much to be done. The major thrust of basic technology development has been completed. The time has now arrived for the hardnosed evaluation of condition monitoring in the real world of aircraft operation. The techniques, equipment, and systems which survive will be those which demonstrate their worth as cost-effective additions to aircraft operating and maintenance plans.

From a review of the status of the individual condition monitoring techniques and overall system concepts, the following recommendations are made:

- Maximum effectiveness of a condition monitoring system demands its integration with the basic engine design and installation.

- Condition monitoring must become a viable element of the engine maintenance plan.

- A condition monitoring system cannot remain a static concept, but must exhibit flexibility for change and growth to meet the challenges of engine design improvements, technique changes, and changes in engine distress patterns.

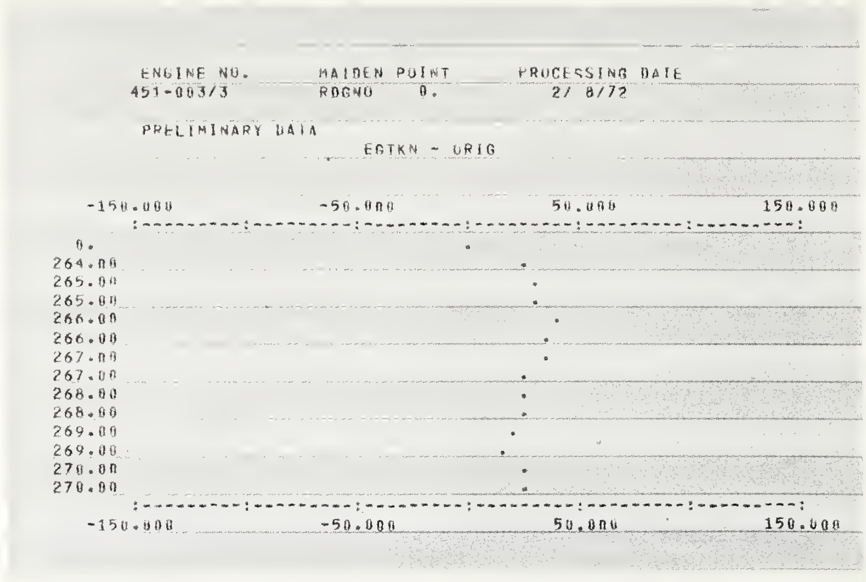


Figure 1. CATT Output Trend Plot of Normalized Exhaust Gas Temperature.

Figure 2. Trend Analysis Digital System (TADS) Equipment.

Figure 3. Test Cell Model of FOD Detector.

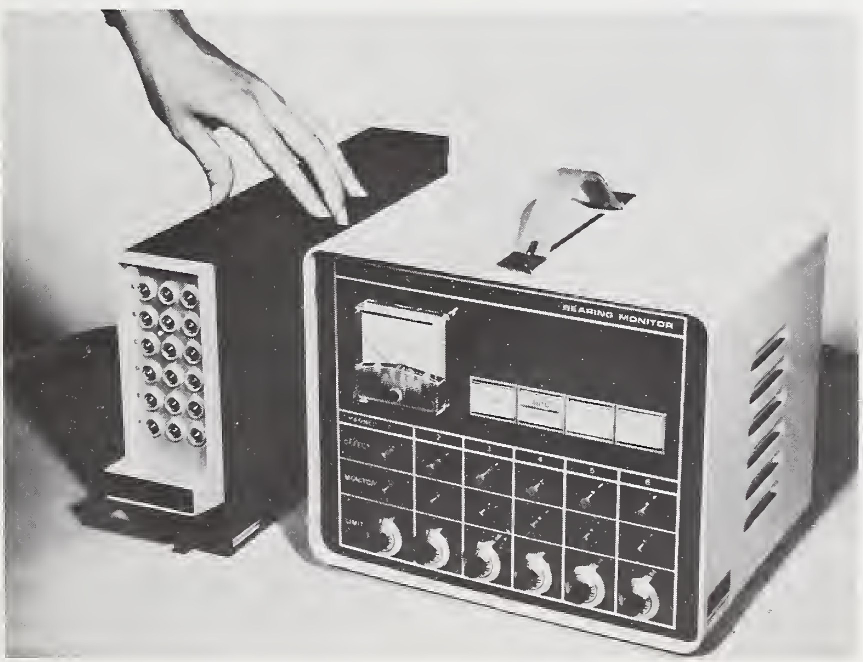


Figure 4. Airborne and Test Cell Models of 6 - Channel Bearing Monitor.

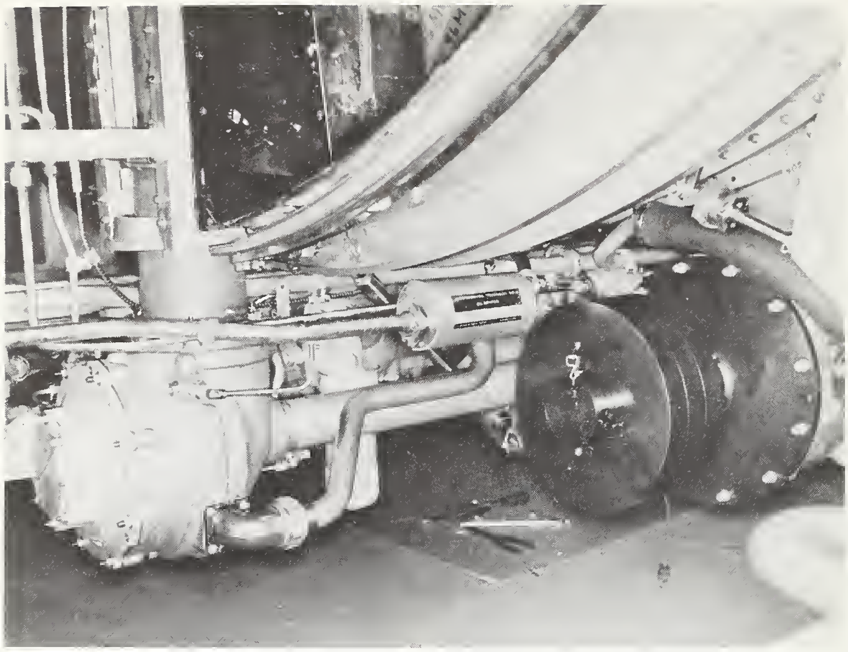


Figure 5. In-Line Light Scatter/Attenuation Monitor.

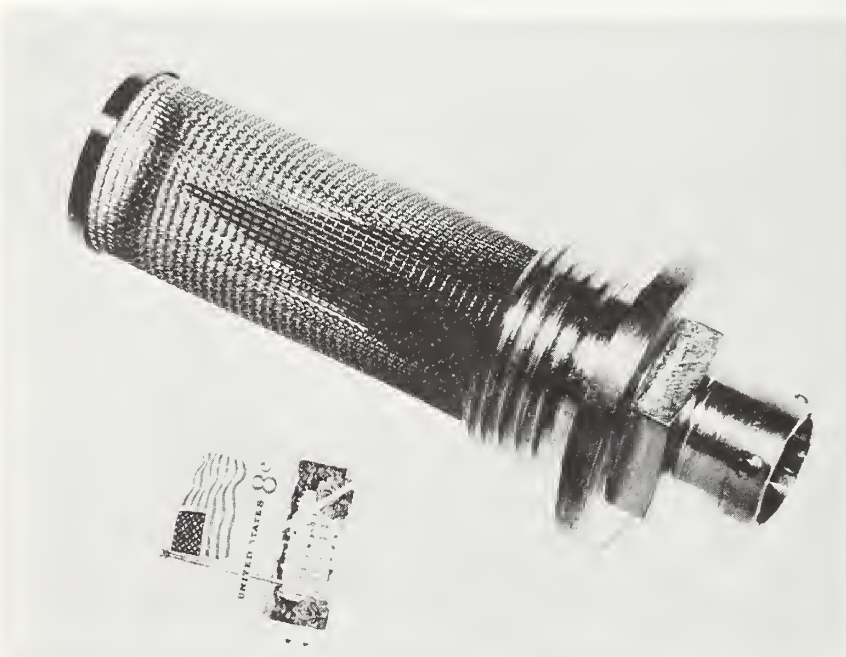


Figure 6. Conducting Debris Monitoring Oil Screen.

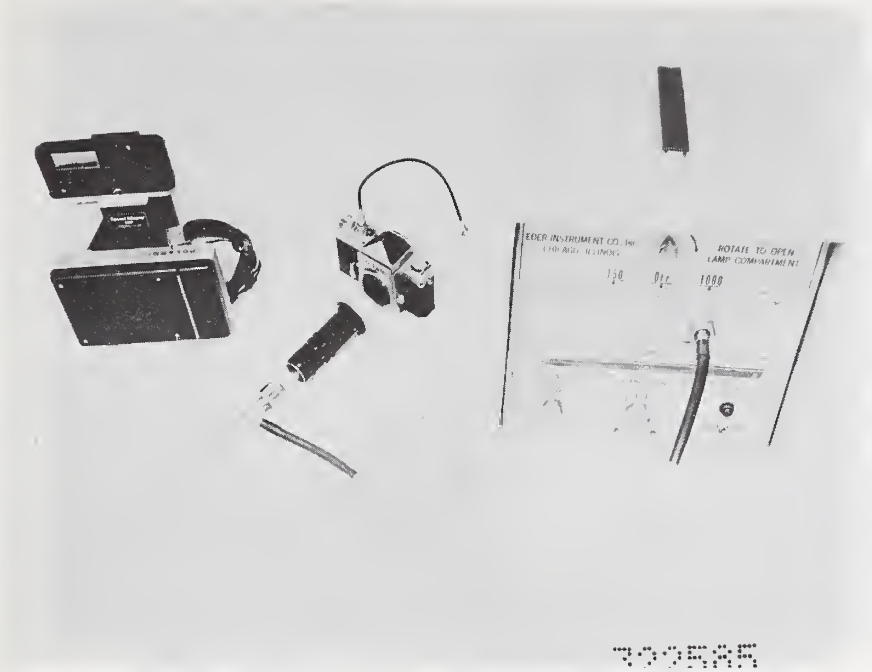


Figure 7. Rigid Borescope and Still Camera Equipment.

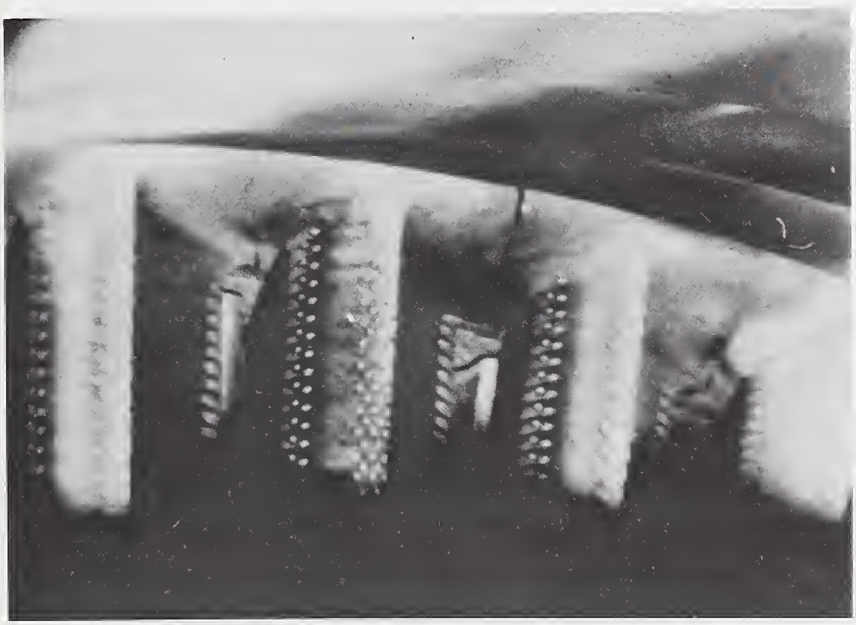


Figure 8. Borescope Photograph of First Stage High Pressure Turbine Nozzle.

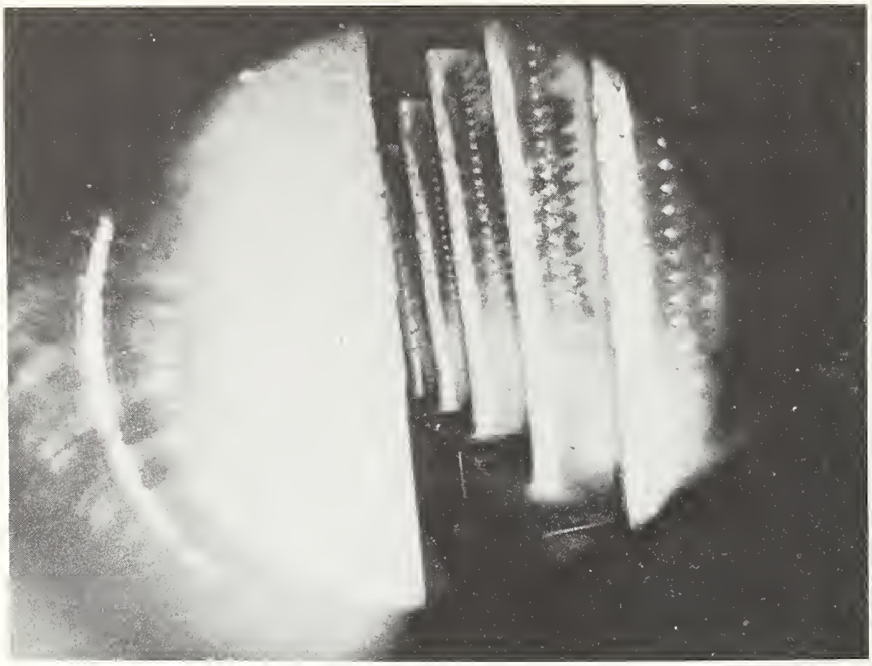


Figure 9. Borescope Photograph of High Pressure Turbine Blades.

Figure 10. Borescope and Video Camera Equipment.

Figure 11. Inserting Source Tube into Shaft of Turbofan Engine.

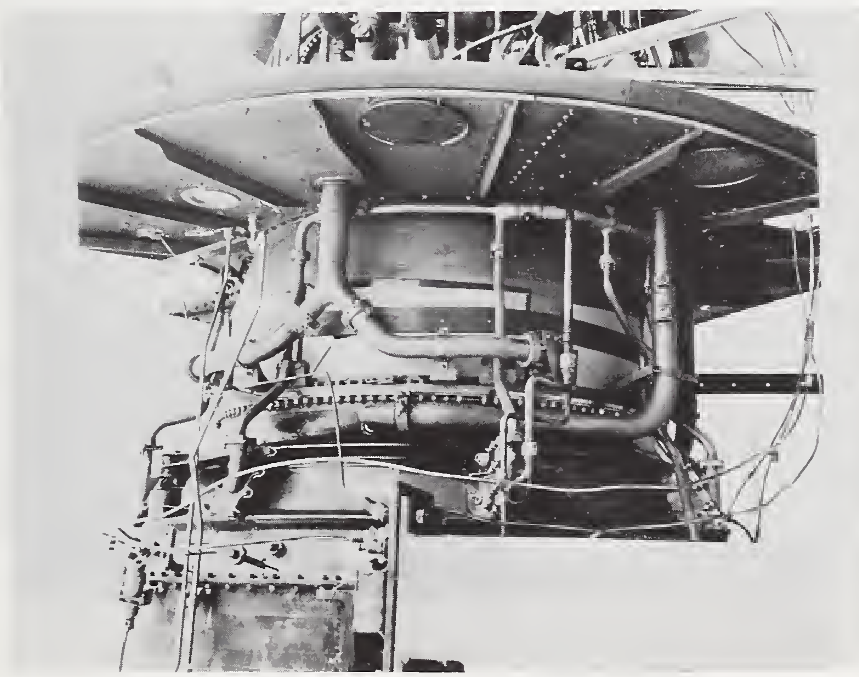


Figure 12. Installation of Film Cassettes on Surface of Engine.

Figure 13. Copy of Radiograph of High Pressure Turbine Nozzle Area.

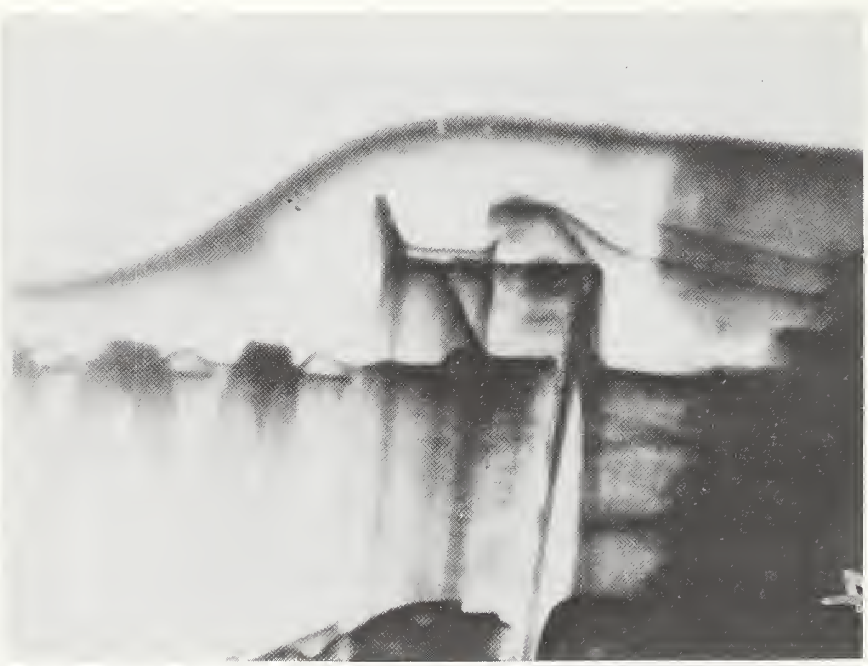


Figure 14. Copy of Radiograph of Combuster/HPT Nozzle Interface.

PANEL DISCUSSION

Panel Members: Murray Hoffman, Joel F. Kuhlberg, George Staton, David W. Leiby

Panel Moderator: Keith Hamilton, Wright Patterson Air Force Base

Ephraim Regelson, NAVAIR: This question is to Mr. Kuhlberg. You mentioned conditioning monitoring with respect to a Navy program. What aircraft did this involve and can you elaborate on the results of the program?

Joel Kuhlberg: The program is called ICEMS, "In Flight Condition Engine Monitoring System". It's sponsored by NASC. It involved the TF-30 engine in the A7 aircraft. There is a complimentary program using the A7 aircraft with the Allison TF-41 engine. Our particular phase has ended. We demonstrated the feasibility of the kind of hardware I showed you. Allison has a follow-on where they are going to use their hardware in a ten aircraft squadron test to shake the bugs out and to qualify it for fleet incorporation.

Charles Jackson, Monsanto: Would you mind commenting on your tape recorder facility that you showed which seemed to be portable? Is it an FM unit, how portable is it? I presume you use this so that you can be very portable about getting data. It looked like you were using about a half inch tape.

Joel Kuhlberg: I believe that was a regular one inch digital tape. I hear Hamilton makes it.

Murray Hoffman: On that ICEMS program the hardware that was provided is similar to the hardware that is now flying in commercial aircraft in the U.S. and in Europe and is available right now. It's all commercial hardware including the tape recorder, the signal conditioning equipment, and the DMU. The ICEMS program was using that hardware, but of course a different set of condition monitoring done through really the soft ware program, which was different than the commercial program that is now in the commercial airlines.

Lee Doubleday, Naval Air Systems Command: As you know, the cost of engines is high these days and our support philosophy, from the logistics view point, has turned toward modularization, that is supporting the engine by component. A number of the systems now in existence, and planned, depend on the trends and the inter-relationships within a given engine. I wonder how the panel plans to handle modular engines.

David Leiby: We're recognizing this as a problem on the modular engine; what do you do with an engine that has some parts that last for 30,000 hours, some parts for 20,000 hours, some for 10,000, some for 5,000, and on down the line. In several years there is a mix of different age components in the engine. This is going to complicate the bookkeeping in the software programs for trend monitoring, and there will have to be changes made in the memory bank for a particular engine when it comes back with parts of different ages. We don't know exactly how we're going to do it. But we recognize the problem.

Joel Kuhlberg: I would like to comment further on the 9D in the 747. To get back to a nominal baseline how do you know for instance that your overhauled compressor is operating properly? The compressor is being put into another engine that happens to be on base which is perhaps comprised of test parts, and run as a complete engine. If the engine performs properly with this compressor, then the compressor is ready as a module to go back in service. Modules aren't tested as modules, they're tested as parts of an engine.

Murray Hoffman: Let me add a little postscript to that, not so much for the modular engines but how engines that have changed their baseline characteristics are handled at least in the industrial area. If you interchange the modular pieces, you're going to change the baseline. The system I described actually interrogates the engine for its baseline characteristics, and then instructs the user to run through a calibration test, running the engine at various power levels while the computer makes the measurements and establishes a new baseline. This occurs all the time.

Lee Doubleday: Do you use the same device for measuring both bearing condition and unbalance, with perhaps a filter to get down to low frequencies for the unbalance measurement?

David Leiby: We use the same accelerometer. The one internal accelerometer on the main bearing is the one we tie into the mass unbalance monitor, the FOD detector, and the bearing monitor. The bearing monitor is usually used in a multiplex type of operation where one monitor will scan several of the accelerometers, but the FOD detector is committed all the time to that accelerometer.

Lee Doubleday: You spoke of how the engine manufacturer does a good job of putting monitoring ports in the engine. Then an air framer wraps his QEC or airconditioning ducts all around it and blocks these ports which blows the whole thing as far as monitoring the engine on a wing.

I would like to stress to the industry that when you write specifications, try to give yourself an envelope that would prevent the airframer from wandering into the borescope areas so that we could possibly use some of these good things that are provided in the first cut of the engine world, which starts about 3 years before the airframe. You didn't mention anything about radiography while running, but I understand Rolls Royce in England has done some work with running engines, and we had experience with a TF-41 engine where we actually saw a fit loosen up with the use of running radiography.

David Leiby: We have not done any per se but we've been thinking about what can be done, and what the gains might be. If you take an engine apart you can look at the seals and you can see that the engine will wander back and forth up to an inch, during transients, relative to actual positions of rotors, etc. It would be nice to know how far the parts do move with respect to each other so you can then keep the clearances of seals down to a bare minimum and improve the performance of the engine. One of the ways of doing this is to use very high power, high pulse energy x-ray machines. You're talking about 8 million volts and using the best high speed film you can get so that you essentially freeze the motion of the engine by shooting across it, and getting a pulse of 20-30 nano-seconds with enough energy to expose double royal blue film with accentuating screens on it. You can get a useful image and measure the actual clearances of the parts of the engine and take a series of radiographs with an automatic film transport tied onto the engine so that as you go through a throttle burst you can see the relative positions of the parts as they move around inside the engine. These are things we are going to be looking at in the future.

Lee Doubleday: From our viewpoint, this is a laboratory tool at present. As a development tool, I think it's real worth while.

Jim Reis, Northrop, Corp.: Were you able to detect worn main bearings in the diesel engine?

George Staton: On a laboratory basis, yes. On an online basis, the diagnostics are not fully developed. We're still working on it. Up until now it has taken some manual interpretation of the data in order to get a positive diagnosis.

Jim Reis: Are there any particular techniques that have been more successful than others in doing this?

George Staton: The technique we're using which we think is the most successful (obviously we're windowing) is looking for that portion of the cycle where a given connecting rod is down at the lower end or reversing directions so that we can see what the effect on the conrod at that point would be. You see more of the vibration of the main bearings. Right now I am running about 12 passes through a digital filter at that window and I'm having to bring the data off line and analyze it with an experimental correlation program in order to fully define the malfunction.

Jim Reis: Have you been able to detect defects in main bearings in the turbine with the Hamilton Standard unit?

Murray Hoffman: No. Not that we couldn't, but we haven't had that problem come up in the tests that we've run. However, the ability to detect main bearing problems with that system is rather low because we are depending primarily on case mounted accelerometers and the attenuation through the unit is going to be pretty heavy. Those engines do not have any kind of internal instrumentation.

Jim Reis: Would you then anticipate when the manufacturers start manufacturing internal accelerometers to include them into your basic system?

Murray Hoffman: Yes, in fact I try every chance I get to encourage the manufacturers to include them internally. I have even asked the manufacturers if they would code the metal that they put in the bearings so that when deposits are found in the oil, you can identify the source.

Cedric Beachem, Naval Research Lab: Mr. Kuhlberg mentioned that there are some A7's that have been instrumented, I suppose for flight operations. Are the Naval Air rework facilities encouraged to become equipped to do the more extensive follow-up on the ground?

Lee Doubleday: The A7 program that was mentioned is a 10 engine development program or a 10 aircraft development program. There's been a first phase development program to determine the winner, and DDAD happened to win. I should say that Teledyne won. They were affiliated with the Allison engine, where Hamilton was affiliated with Pratt and Whitney. There is nothing in place yet at any rework facility to do this. At our depots, this is a flight line type of approach and the depot is many miles away so we don't really get a close tie-in with the squadrons other than a gripe about an engine as it comes back. The whole approach to this has been on a squadron level or IMA level as far as the diagnostics go, and then if the maintenance task is beyond their capability, it gets sent back as it would be now.

A.P. Brackney, Monsanto: First I would like to make some observations on the session this morning and then direct a discussion to the panel. I thought the papers were very excellent. Yesterday, Charlie Jackson observed that equipment is not fickle, like people. Years ago before we had the scientific way, we used to have a lot of conversations with people. A conversation today is two guys, one talking and one waiting to talk, and nobody's listening. We have conversations with our equipment, and if you don't listen, there's no conversation. We listen to equipment but we don't listen to people. For the user of an engine, be it an aircraft engine, or a pump in an Monsanto plant, you've got to talk to the maintenance man and tell him some of the things that you have observed. I would like to ask the panel and the group to tell your compatriots on the other side of the fence what information you need to make a job easier, and let's define easier in terms of cost, and being right the first time 100% of the time. There's much information that a user can tell a maintenance man and if he would be smart enough to tell him, the end result would help him. He would get better engines back and he would know what to do and what not to do while he's running them. It's the little details that make a plane go down, or a tank stop in the middle of a battle, or a pump fail.

David Leiby: At General Electric we have an organization called Field Service Engineering. They are our ears and eyes out in the field with our customer. We have a pretty good run down on all the problems the maintenance people have had. We have very good rapport between our field service engineering and the maintenance sheds at all our customer's overhaul bases and operating bases, so we know where the weak spots are. We know what monitoring techniques need to be employed, we know which radiographic procedure needs a few changes, we know that the borescopes aren't working in a certain port because something is interfering. So we get very good feedback from our customers.

A.P. Brackney: That tells me that you are getting feedback from your people that you sent out in the field. The customers are telling your man out there, and your man is listening, but you had to take that overt step, you felt that was necessary. That proves the point that usually there is not much communication between the user and the guy that fixes or makes the engine. I think what you're doing is good.

David Leiby: This has been our operating procedure for years. We always have this liason between the customer and our own engineering and manufacturing operations.

George Staton: We have a little bit different problem than most people in that in the Army, our maintenance staff and our users don't remain fixed for any given length of time. The typical government way to do things is to fill out forms which end up as a pile of unusable data. What a driver of a tank or a truck is going to say when he brings it in for maintenance is that it doesn't run right, or it makes a noise. All he wants to do is get rid of that engine, or that vehicle, and get a new one that runs better. So we have a severe communication problem. We do all the typical kinds of things like having civilian maintenance, both government people and contractor people, out on site trying to help our own maintenance people and our users to communicate with each other. For us, it's almost an insurmountable task because you have no continuity from year to year in the staff. Therefore, in our diagnostic analyses, we're looking at the equipment for the first time and we can't use a trending approach.

Joel Kuhlberg: Pratt & Whitney also has field engineers all over the world watching our engines. Generally they are assigned to an overhaul base or an operational base. At an overhaul base when there is a repair situation, care must be taken to observe the evidence as it is exposed on disassembly. Operationally, you've got to get a good story from the flight crew that flew the airplane that that engine was in. Sometimes that's a problem. It depends on the airplane, the kind of mission flown, and the crew. If you've got a fighter with a couple of guys flying a mission and they get an engine problem, they are probably pretty busy and they may not get a very good story of what is happening to all those dials in front of them. Of course, that's where condition monitoring is supposed to help, to get some of that story. On the other hand on a commercial transport with a flight engineer, you probably can get a better story of the event. When we do have an unusual problem, generally we request that the hardware be sent back to the plant and engineering tries to figure out what's going on and how to fix it. The communication gets to be quite direct when you have the hardware in front of you.

A.P. Brackney: When you are getting information back from a flight engineer, is this contact face to face? Are these what the observations were when the airplane was doing thus and so?

Joel Kuhlberg: It can be face to face between the pilot and our field engineer on a debriefing. Then our field engineer gets back to us.

Murray Hoffman: I'm between the user and the supplier in condition monitoring. On the part of the manufacturer, there's a tendency to play down the biggest problems and to indicate that these problems are just normal routine maintenance problems. On the part of the user, I find myself overwhelmed by the requirements that he throws at me and I can't build a house big enough to hold the equipment to satisfy those requirements. The user is so amazed and flattered at being asked to contribute to the situation that he goes so completely overboard that you drown. You go back to the manufacturer and ask "What problems can we help you with?" You get back the answer, "What problems?" I don't think there is a good answer for it now, but I think that there is an awareness of the problem.

A.P. Brackney: I can see disciplines forming within this group. Maybe we need an inter-discipline committee within this group just to make sure we don't go building walls around original equipment manufacturers and repair shops and users.

Ron Phillips, U.S. Naval Aviation Integrated Logistics Support Center: What reliability have you gotton from your transducers and what kind of print-outs have you gotton that said there is a fault, where in actuality no fault existed either due to a bad transducer or programming error or whatever?

Murray Hoffman: It's a little early to draw any real conclusions on reliability, but to date, our experience with the transducers has been very, very good. We haven't had a single transducer failure in a year and a half, nor have we had a single transducer that has gotton out of calibration. There is a good reason for that. We are very careful in our selection of transducers. For example, we put 11 pressure transducers on an engine and the raw cost is around $6,000. In terms of the rest of the equipment we had some laboratory conglomerations that had seen better times and I wouldn't really want to integrate their performance into the reliability of the system. I think that the electronics probably have as good a reliability as the transducers have experienced. The only wrong messages that we have generated so far have been those where the test engineers, while working with an experimental engine had double instrumented, and so overloaded the lines, that the results were erroneous and the computer read these as changes in performance. We work to very small tolerances. Data is accumulated for 150 hours and, on the basis of this, decisions are made. This is not an alarm system. This is a trend system. I know what you are really looking for but we haven't had enough field experience to give those answers.

Joel Kuhlberg: I guess I brought up the point a little bit when I spoke of reliability. I don't have a number for you in terms of transducers but I have an idea that it's probably in the 100's of hours domain. The kind of reliability that we hope to achieve with electronics, sensors, etc. is the same kind of reliability we now have with the hydra-mechanical control, and those numbers are like 20,000 hours. I don't think we have that kind of reliability in most of the sensors that we have. Another thing important to a flight engine as opposed to an industrial installation is that the range of operation has considerably increased and the environment that it has to live in is a lot more severe, and that's part of the problem of accuracy. We talk about half a percent of full scale but if you talk about a pressure transducer that has a turn down ratio of about 70-1, you find out a half percent full scale down at the one is an awful lot of PSI, and that's not good enough for controls or condition monitoring.

George Staton: A major portion of our problems with transducers has been due to the fact, that in our system the transducers are removed after each test. We have had failures which are not considered reliability failures but are due mainly to two things. The first is handling. The second is exposure to severe environments like in the exhaust manifold. We knew when we designed the system that the transducer would have a finite life, so we have been seeing failures where we expected to see them, and they are occurring at roughly 1300°F in a high impact environment.

David Leiby: We have found that the sensors and transducers are very reliable. The problems are in the connectors. I think we need some improvements in reliability in the electronics.

William Glew, Naval Eng. Test Establishment: Mr. Staton, why do you rely on obtaining cylinder pressure and injector efficiency measurements with an external accelerometer rather than on pressure transducers mounted on the head of the engine to give you a P-V diagram?

George Staton: The major problem is that we have to disassemble to get those pressure measurements. This is a 12 cylinder aircooled engine, which means we have to have adapters that would go into the injector hole of every one of those cylinders because it's the only access port we have. We don't think that we need to get the compression pressure directly if we can read the relative compression pressure from the start of the current wave form, which we do, and look at the injector performance through the pressure timewave form of the injector.

William Glew: My only thought is that what you're doing is a very rough class measurement of the effectiveness, and in fact the cylinder pressure temperature conditions are very important to the effectiveness or to the efficiency of a diesel engine. I would have thought that once you go to the trouble of having a condition monitoring system, the extra cost of the pressure transducer tappings would be justified in the long run.

George Staton: We are not doing trend monitoring with that system. We are doing diagnostic monitoring in the as received condition. We get the engine in an unknown condition. We make an assessment of it, and based on that one look at the engine, decide what has to be done to fix it. When it's fixed the instrumentation is removed and it's shipped back to the field. We can't change the basic design of the engine, so we have to use only those access ports that are available. We try to minimize the amount of labor needed to mount our instrumentation. Unless it's absolutely critical to our diagnosis, we don't like to break into the lines, we don't like to go into major disassemblies of the engine because if you get to that point, you might as well tear the engine down and do a full rebuild.

William Glew: I guess the trend monitoring is the next step. It seems to me that the only weakness at present with this system is that you're not getting a good bearing deterioration picture. Statistically, how significant are failures of bearings in engine failures? Is it a significant overall failure mechanism?

Murray Hoffman: Yes, bearing failures appear to be a significant failure mode in terms of gas turbine engines. However, inside corrosion from poor fuel is a very common problem with industrial engines. Problems like this tend to necessitate that the engine be torn down more frequently than a bearing problem would require, and at that time often the bearings are replaced.

William Glew: Do you think the trend system in its present form is suitable for the shipboard installation on the Marine FT-4?

Murray Hoffman: Oh yes. As a matter of fact it's being applied to other engines besides Pratt & Whitney's intermarine applications. It can be applied to any gas turbine.

William Glew: How much do you charge per system?

Murray Hoffman: How much have you got? This is a commercial system available to the commercial market at a fixed price. Installation costs vary, however, and can run the total cost up significantly. In round terms, it's about $65,000 for the first engine, and about $130,000 for 8 engines. You can estimate from there for whatever number you want using a single central unit. On a ship, you have a different problem. You can't go back to the beach although you could if you wanted to use your radio length.

William Glew: If I was running a ship and you installed one of the GE bearing monitors I would like to have your GE bearing monitor system with me. I would like to comment philosophically that we installed a very simple vibration monitoring system in the Navy, and what has struck me is that quite often there is an element of reluctance on the part of the maintainers to draw the deductions that can be obtained from this analytical equipment. Instead of regarding it as an extra screwdriver or right arm, they tend to say when something goes wrong, "That's your black box again". It seems to me that we, as engineers, must all be very careful not to depersonalize the maintenance operation and thus make the maintainers feel that they are completely out of the picture and just automations doing what the computer and the computer technicians tell them they have to do.

George Staton: If we had or could retain personnel with the trained skills, then they could do much better than any machine that we can build. But unfortunately, in our particular environment, if we got a skilled diagnostician we would promote him to sergeant so he can do paper work. So we're stuck with the 18-20 year old recruit who gets about 12 weeks training if he's lucky. If he's not, when he gets to the unit somebody hits him on the head and says you're now a 63B automotive mechanic. The guy who was in school could also type, so he's now a company clerk. We have to somehow stop a very significant traffic in serviceable parts being removed from engines and being returned to the supply system. By significant I mean 60 million dollars a year that we can identify, and lots more that we can't. Despite the fact that there may be some few people out there who would get depersonalized by us putting a machine there, we can hardly justify not putting it in on a cost effective basis.

Robert Boole, General Radio Co.: There have been some comments about the frequency response of sensors or transducers that are used in these programs, but no one has really mentioned the dynamic range that might be required. I would be interested in comments on any of the transducers that are used and what sort of dynamic range is required?

Murray Hoffman: In terms of the range of instrumentation that I am applying in the TREND system, there's a very small dynamic range. We work over very small ranges because we're concerned with the engine only when it's in a steady state condition. We define steady state condition as the rotor speed being plus or minus 0.1% for 10 minutes, and then at that point we take a reading.

Robert Boole: Are you talking about typical ranges of less than 10-1? The gentleman from United Aircraft mentioned a 70-1 reduction from a full scale reading to where he was operating.

Murray Hoffman: In terms of aircraft engine condition monitoring, you're monitoring from start up to shut down. In the system I'm looking at, I'm only monitoring during the steady state condition, so I don't need the large dynamic range. I have a little simpler problem. The primary thing that I'm concerned with is repeatability. If I get good, solid substantial repeatability, I'm not overly concerned with absolute value or any of these other things or the large dynamic range.

Joel Kuhlberg: 70-1 is the pressure range that the pressure transducer has got to be exposed to and work over in an advanced supersonic engine. Obviously, the temperature ranges and speed ranges aren't anything like that. Are you interested in particular parameters?

Robert Boole: There have been plots shown here, particularly of vibration, which probably have the widest requirements for dynamic range, and I have seen shifts on some of the charts with as much as 40 dB, in some cases more, between various points on the spectrum. If you do a spectrum analysis, you might see 2 levels plotted on that spectrogram which differ by 40 dB which means that the sensor has to be able to respond in amplitude over that range. That's what I was trying to get at for vibration or any of these other parameters. Over what sort of an amplitude range must the device be capable of giving the useable output signal?

Charles Jackson: I think 40 dB is about right. That's 100/1, that's about what we find.

S. Grant, Carolina Power & Light Co.: I understand that biological contamination of diesel fuel oil and perhaps other oils can be a problem and I wonder if any of you have any experience with this and ways of checking for it?

Lee Brock, Eastern Airlines: We occasionly get this micro-organism growth, and I assume that is what you are speaking of, in the aircraft fuel tanks. You can usually detect this by the pick-up in the fuel filter. We have a fungicidal treatment that we put into the fuel tanks to kill this growth. I don't know who the manufacturer is, but it is a commercially available product. It's a one time treatment. You put a certain amount in the fuel tank and run a tank full of fuel through the engine. We don't use it continuously and there is no noticable effect on our engines.

S. Grant: How did the biological contamination affect your engine?

Lee Brock: It caused control malfunctions to the engine. One of the main concerns to an airline is the fact that it eats holes in the wings of the airplane. This micro-organism causes severe aluminum corrosion in wet fuel tanks. It looks almost like cotton. The first few times we saw it, we thought the tanks were contaminated with solid material.

P. Chiarito, NASA: Getting back to specific needs for a detection system, has anyone seen a need for a system that would detect crack growth in the rim of a turbine disk during flight operation? Is anyone here aware that turbine disks do burst occasionally? If we have a detector for crack growth in the rim, we hopefully will have an indication of incipient structural failure.

David Leiby: We have been looking in the general development field of acoustical emission for detecting cracks, not necessarily in disks, but in any internal parts of engines.

M. Rosen, GE: How do you go about establishing the threshold level that triggers an alert or alarm signal especially using gas path analysis or oil analysis on both new and old machines?

Murray Hoffman: The way I approach the problem is to look for a change in the performance of the unit over a long period of time. I'm looking for small changes, I need certain parameters to establish when a change is significant. I have to know the intimate performance limits of the engine in a general sense. Then I relate this to the baseline performance and then establish quantitative values which then represent the limits that I am establishing for whatever flags I might bring up. There are probably a family of limits that are established here. There is one that is just short of catastrophic or maybe there's one that will say the engine just left the airplane. Way down on the other side is the limit, that 100 hours from now there will be deterioration to the point of non-acceptance. We know the percentage change in the various performance parameters, particularly with gas path analysis, which define the loss of efficiency. This is based on performance history, of which there are many thousands of hours on 100's of 1000's of engines. It's a very dynamic thing, and I wouldn't pretend that we have all the right limits now. As you gain experience, your engines tend to change too. I think the precise point where you should put those limits will probably never be found, but a good working number based on history and a knowledge of engine performance can be available right now.

David Leiby: In many of the applications, for example the exhaust gas temperature, you have a hard solid limit that is a red line saying thou shalt not go above this or you're going to damage the engine. Some of the limits are established by other limits. In condition monitoring, you can evaluate the operating condition of the engine and come back and put in hardline limits, of course we also have soft limits. The fact that you're now departing from the normal indicates that you're approaching the hard limit at a certain rate. Therefore in 20 hours, 40 hours, 50 hours, some period of time you're going to get to this limit, so you better start scheduling this engine to be taken off at the most convenient base the craft is going to be entering.

SESSION IV

NEW APPROACHES IN SENSING AND PROCESSING

Chairman: Henry R. Hegner
IIT Research Institute

APPLICATION OF ACOUSTICAL HOLOGRAPHY TO INDUSTRIAL TESTING

Byron B. Brenden
Holosonics, Inc.
Richland, Wa.

The application of the techniques of holography to acoustical imaging has been rather recent, certainly not predating 1964. These techniques have had a great impact upon the development of acoustical imaging not only by making testing techniques depending upon imaging more useful but also stimulating improvements in acoustical imaging in general. From the beginning, systems employing a scanned receiver, as schematically shown in Figure 1, have been popular. The most effective scanned system is one conceived of by K. A. Haines and B. P. Hildebrand and described in U. S. Patent No. 3,632,183. In this system the source and receiver are scanned over the hologram plane together.

In a system perfected by H. D. Collins and R. P. Gribble of Holosonics, and shown schematically in Figure 2, a single transducer serves as both the source and receiver on a time shared basis. This system has been demonstrated to be very effective in imaging, locating and sizing flaws in large rotor shafts and reactor pressure vessels. Figure 3 is a picture of the system showing the mechanical scanner held to a simulated reactor pressure vessel wall with rubber suction cups. Figure 4 shows a reactor pressure vessel under test using the Holosonics HolScan equipment. This equipment forms a hologram on polaroid transparency film. Subsequent imaging by illuminating the hologram in an optical processor enables one to see the defect or feature in its true configuration and permits one to measure its size and actual location.

The HolScan system is a very effective tool for studying volumes on the order of one cubic foot and looking for features as small as an eighth inch in diameter. Furthermore the parameters of the test may be varied to look at larger volumes or smaller defect features.

Acoustical holography may also be used for real time acoustical imaging. In this case the system takes on the configuration shown schematically in Figure 5. Normally this system is used in a through-transmission, focused-image mode but it can be used effectively in a reflection mode and without acoustical lenses. Holograms are formed at a 60Hz rate and read out instantaneously permitting one to inspect moving objects and also permitting one to move the object about to achieve the optimum angle of view. This system is available in several configurations, one of which is shown in Figure 6. All of the configurations are based upon the principles illustrated in Figure 5. A wave train several hundred wavelengths long and about 5 inches long in water is generated by the object transducer at a typical frequency of 5MHz. Energy scattered by the object passes through the acoustic lenses along with unscattered energy. Acoustic lenses focus the object into the hologram surface. A reference beam is mixed with the object beam at this surface. The resulting interference pattern is impressed upon the liquid surface as variations in elevation which affect the phase distribution in the reflected light beam. The liquid surface, which is fully stabilized against unwanted disturbance, is capable of forming one hundred holograms per second and thus of imaging in real time.

Figure 6 illustrates a typical nondestructive test situation involving a laminated strip of metal made up by bonding a strip of aluminum to a base strip of steel. The laminated strip is produced in thicknesses ranging from 0.065 inches to 0.110 inches with about 0.010 inches being aluminum. Figure 6 is the acoustical image of four inch wide strip showing both edges and a long, narrow, poorly bonded area. The handle of a hemostat is also imaged through the strip in order to demonstrate the transparency of the strip to 5MHz ultrasound.

Delaminations and anomalous areas in plate like materials are readily imaged by this technique. Cracks, slag and nonfused areas have been imaged in steel pieces more than 8 inches thick.

Irregular shaped solid objects are more difficult to test but many of them can be inspected by using off-axis insonification and special distortion correction lenses. However, the correction lenses are quite specific to the application and further research is needed in this area.

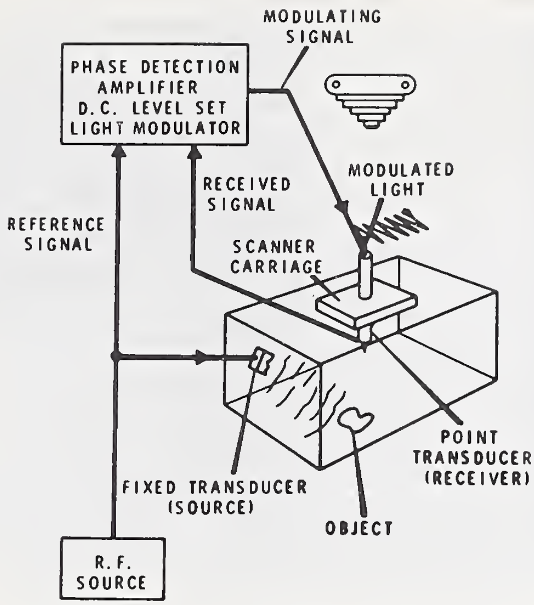
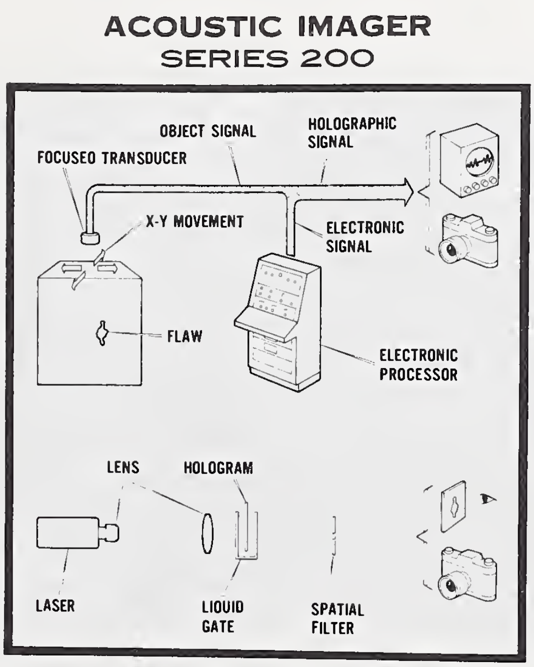


Figure 1



Schematic diagram of Holosonics, Inc. Pulse Echo Acoustical Holography Systems and Holographic Reconstruction Unit.

Figure 2

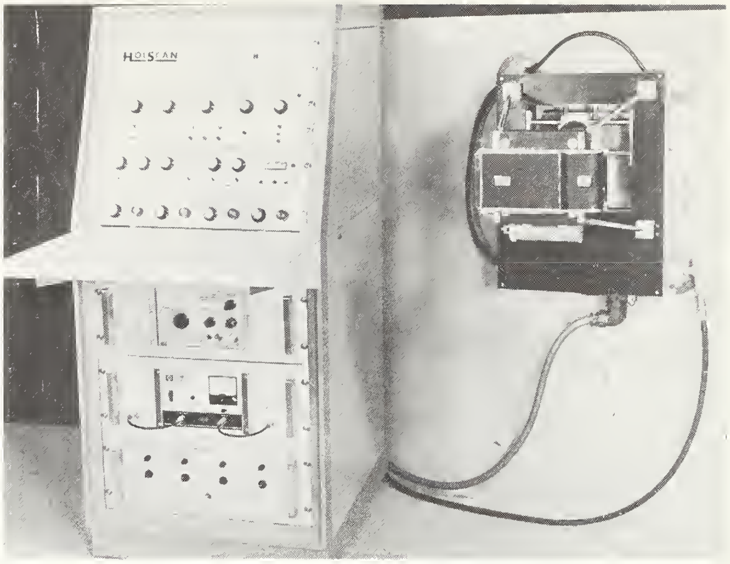


Figure 3

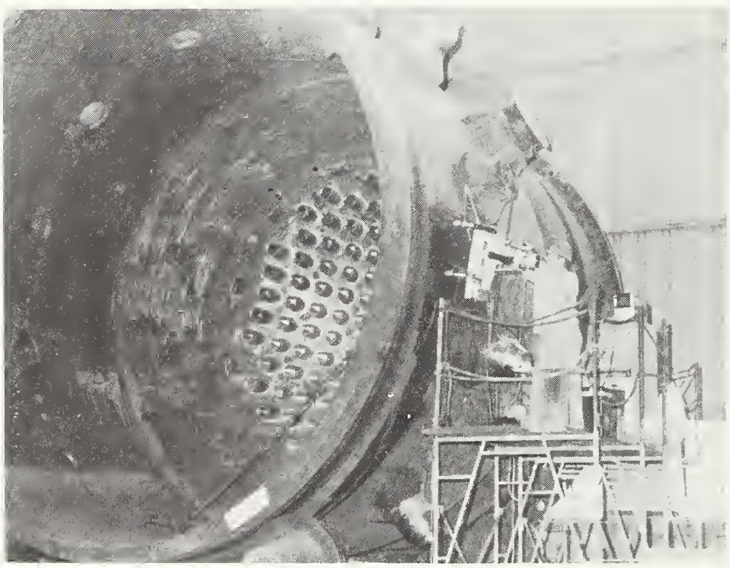
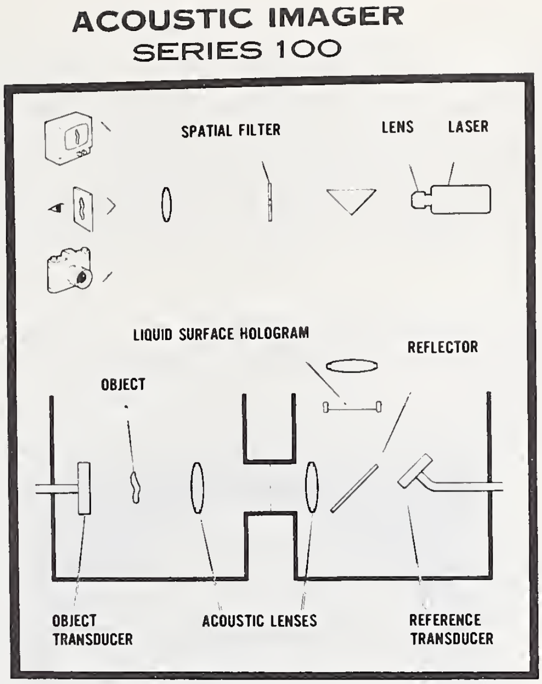


Figure 4



SCHEMATIC DIAGRAM OF HOLOSONICS, INC. LIQUID SURFACE IMAGING SYSTEM

Figure 5

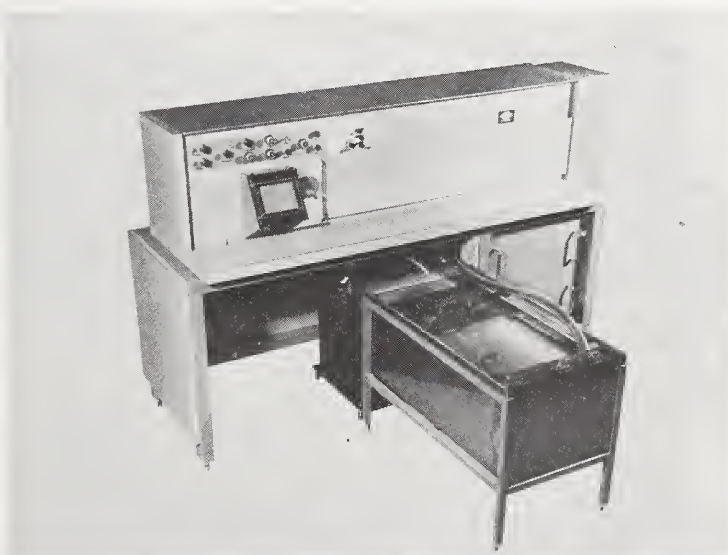


Figure 6

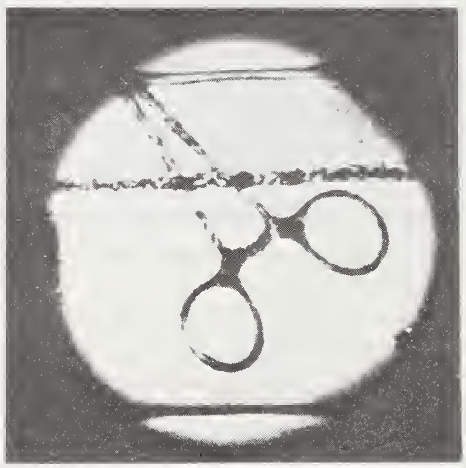


Figure 7

DISCUSSION

Jerry Forest, Ontario Hydro: As I understand it, in the system you are using, the liquid surface method produces an acoustical field within the volume of water and the image is produced on the surface. I take it that this is an interference pattern of the two acoustical wave patterns producing a deflection at the surface of the water. Do you use an optical holographic method to produce the picture of the disturbed surface?

B. Brenden: That's correct. A complex grating is impressed upon the liquid surface. This grating, which is a hologram, is formed in terms of relative elevation on the surface. The fact that the energy from the object beam is mixed with the energy from the reference beam causes this grating to form. Then that is read out in the way you normally read out a hologram.

Jerry Forest: As the block was moved around in the liquid, the whole liquid surface level moved. For this type of diffraction pattern, in conventional optical holography one must keep motion to an absolute minimum. How was that accomplished?

B. Brenden: The liquid surface has a dynamic response and the hologram is formed in a matter of about 100 microseconds. The object then must not move appreciably in terms of acoustical wave lengths over that 100 microseconds. In effect you're taking a rapid sequence of pictures. Your object can move around by a matter of 0.020 inch or so during that 100 microseconds and you still have your hologram.

Cox, Harry Diamond Labs: I'm in the field of fluidics. In fluidics we chemically etch passages in thin metal sheets and then diffusion bond them together. Can acoustic holography detect unbonded sections in such a structure, and if so how small a defect can we observe?

B. Brenden: Plate like structures and delaminations in plate like structures are just ideal for examination by this method and the disbond may have almost zero actual separation and be detected, but the areal extent should be on the order of a couple of wavelengths of sound. In that case if you're operating at 5 megahertz, there would be approximately 0.040 inch in diameter to be detected.

APPLICATIONS OF TIME-LAPSE INTERFEROMETRY AND CONTOURING USING TELEVISION SYSTEMS

A. Macovski, S. D. Ramsey, Jr., and L. F. Schaefer

One method of preventing mechanical failures is to non-destructively test manufactured parts for hidden defects or out-of-tolerance dimensions. Holographic interferometric techniques have been used for non-destructive testing but not on a mass scale due to the time and expense involved. However, when real-time television techniques are used mass testing becomes possible since no photographic processing is necessary and no expendable materials are used.

Two modes of operation are used for testing depending on the sensitivity of the measurement required. In the holographic interferometry mode the sensitivity of measurement is one half of the laser wavelength used. Figure 1 shows a block diagram of the interferometric system. An image hologram is formed on the television storage tube and scanned out to form a video signal. For the object in one state of stress, the video signal $e_1(t)$ is stored electronically on a video recorder. After the state of stress is changed a second hologram is scanned out as $e_2(t)$, and the stored signal is read out in synchronism. The two signals are subtracted and envelope detected before being displayed on a television monitor. The output is an image of an object with a set of fringes superimposed indicating the change in surface configuration due to the difference in stress applied to the object. A hidden defect manifests itself as an anomalous fringe pattern on the object with each fringe indicating one half of a wavelength change in surface position.

The second mode of operation is a moiré technique and is shown in Fig. 2. Two plane waves from a laser cause a grating to be formed in space such that the object contour information $Z(x,y)$ is contained in the phase of the grating on the surface of the object. As in the previous system two frames of information are required, the first being the reference frame. The reference video signal is stored and then read out in synchronism with the second frame containing the contour information of the object. The two signals are product detected and added to a low-pass filtered version of the object signal before being displayed. Again, the output picture is a set of contour fringes superimposed on an image of the object. If a contour map of the object height is desired, then the reference is a flat plane and the fringes represent equidistant altitude contours. For information about the change in the object surface configuration due to an applied stress, the unstressed object is the reference and the fringes indicate the change in the surface. For gauging parts against a master, the master part is the reference, and the fringes indicate the gauging difference.

The measurement sensitivity of the moiré technique is dependent on the laser wavelength and the geometry of illumination and viewing.

By varying the angles α and γ the sensitivity may be changed in a continuous manner. The range of sensitivities that are useful depends on the size of the object but is generally in the range of ten to several hundred wavelengths of measurement for every fringe produced.

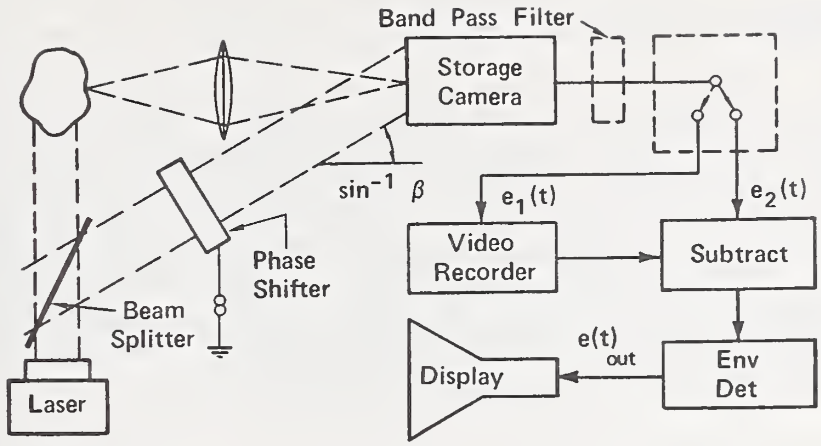


Figure 1

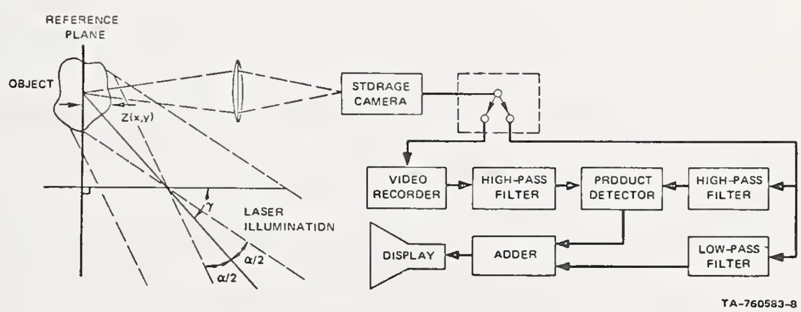


Figure 2

DISCUSSION

Jerry Forest: I wasn't quite sure how you got away from the stability of something like a granite table.

S. Ramsey: The way we get away from the granite table, as Prof. Leith explained last night, is to use a pulsed laser, with a high energy, but a short pulse life. Then you don't need the stability. We don't go quite to that extreme. We use a continuous laser, although in theory you can use a pulsed laser. The effective exposure time with a television camera is short compared to the normal fluctuations of the object. The only reason we can get around that particular problem is that television cameras are in general much more sensitive than the photographic films which are normally used for holography. So we trade off resolution for speed in some sense.

Application of Exoelectron Emission for Detection of Fatigue Damage and Plastic Deformation

WILLIAM J. BAXTER
Research Laboratories, General Motors Corp., Warren, Michigan 48090

The photoelectron emission from a metal is temporarily enhanced by abrasion or deformation of the surface, a phenomenon commonly referred to as exoelectron emission. This paper describes a scanning apparatus which displays the distribution of plastic deformation by measuring the localized exoelectron emission. Our results provide for the first time the early detection of fatigue damage (in <1% of the fatigue life), a quantitative assessment of the extent of the damage, and thus an inherent predictive capability. Some additional experiments are also described which show that the emission process is closely related to the development of microcracks in the surface oxide.

APPARATUS

The experimental arrangement is shown in Fig. 1. The specimen is mounted in a vacuum chamber (10^{-8} torr) and scanned along its length by a small spot (15-70 μm diameter) of ultraviolet light. The electrons are detected by an electron multiplier, and the emission rate is recorded as a function of the position of the light spot. The specimens have a gauge length 0.1 in. long and 0.1 in. wide and are deformed or fatigued *in situ* by bending.

The results reported here were obtained from 1100 aluminum and 1018 steel sheet stock. The samples were in the as-received condition with an average grain size of ∿30 μm, and simply degreased prior to insertion in the vacuum chamber.

RESULTS

Over 40 samples of aluminum and steel have been fatigued in the bending mode with similar results. An example of the exo-electron emission produced by fatigue-cycling 1100 aluminum is shown in Fig. 2. The two traces were obtained after 1000 cycles and 4000 cycles. The initial emission before cycling is omitted for clarity but is typified by the essentially constant noise level on either side of the central peaks. These peaks are very stable over a period of hours. Additional cycling caused these emission peaks to grow and fresh peaks also emerged. The magnitude of the peaks A and B are plotted as a function of the number of cycles in Fig. 3. Failure finally occurred at the location of the largest peak (A) after 140,000 cycles.

Since the location of failure always corresponds to the largest emission peak, it is clearly important to ensure that the scan path intercepts the region of maximum emission. Therefore in our more recent experiments the sample is scanned along five parallel paths separated by $\sim 300\ \mu$. Along each path several emission peaks are observed, but it is only the height of the largest emission peak, regardless of its location, which is important in assessing the accumulation of fatigue damage. This is demonstrated by the results for such multiple scan measurements on several steel samples covering a range of fatigue lives (Fig. 4). Only the height $(\frac{I}{I_o} - 1)$ of the largest emission peak observed on each sample is plotted; I is the measured intensity of emission and I_o is the initial intensity at that location. This normalization compensates for any experimental variations from one experiment to another. The results all fall in a quite narrow band and provide a good predictive capability. For example, we know that when the maximum intensity of localized emission is 10 times the initial background intensity, the sample is at 2.2$\pm$0.7% of its ultimate fatigue life.

The onset of electron emission occurs long before the development of fatigue cracks, the initial emission peaks corresponding to regions where slip lines have formed on the surface. However slip lines produced by unidirectional deformation do not necessarily cause exoelectron emission. For example in the case of aluminum, emission was observed after a tensile strain of 1%, but a compressive strain of 5% produced no emission.

DISCUSSION

This scanning exoelectron emission technique clearly detects localized fatigue damage earlier than 1% of fatigue life, provides a measure of the accumulated damage, and may be used to predict the number of cycles to failure.

Since the intensity of the localized exoelectron emission is very stable, it clearly results from a permanent change in the surface, e.g. slip traces rather than surface adsorption or the diffusion of lattice vacancies. The striking contrast between tensile and compressive deformation points very clearly to the important role of the development of microcracks in the surface oxide.

The continual increase of emission produced by fatigue cycling may be understood in the following way. The initial surface deformation, consisting simply of slip lines, cracks open the surface oxide permitting photoelectron emission from the underlying metal. Continued fatigue cycling enhances some of these slip lines forming so-called persistent slip bands, wherein the deformation now accumulates producing extrusion and intrusion of material and ultimately developing into fatigue cracks. Throughout this process the surface deformation increases in severity, i.e., the surface area of bare metal increases and hence the emission. Finally failure occurs in the location of most intense emission, i.e., the largest fatigue crack.

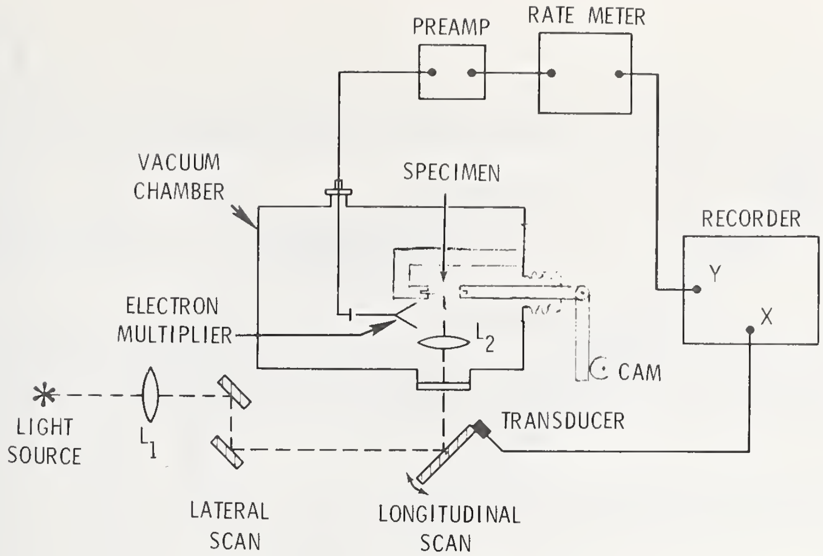


Figure 1. Schematic diagram of experimental arrangement.

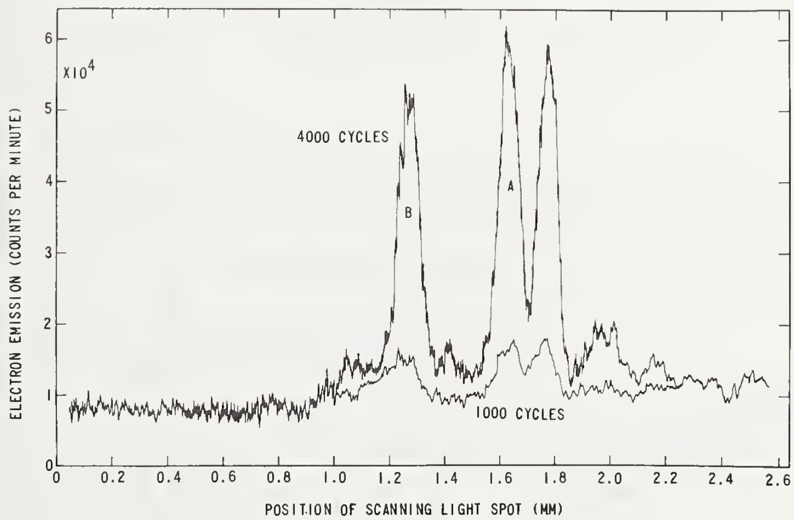


Figure 2. Exoelectron emission from aluminum after 1000 and 4000 fatigue cycles. Fatigue life = 140,000 cycles. Spot diameter = 70 μm.

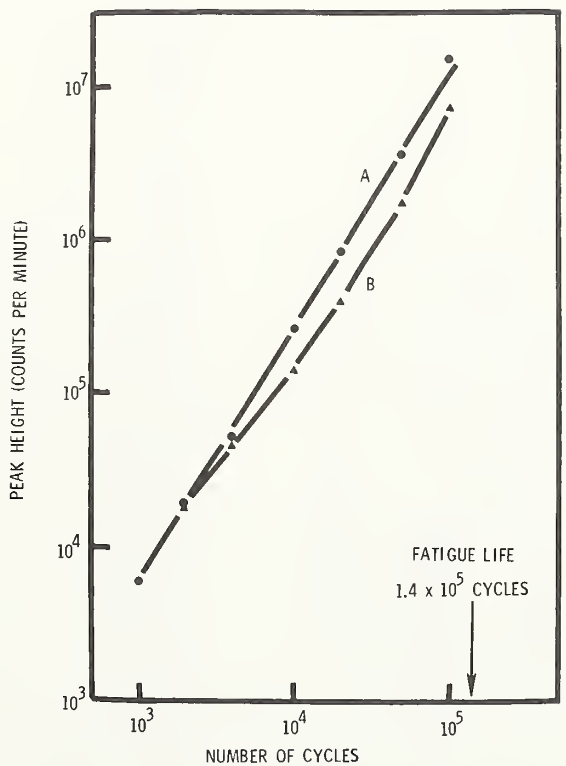


Figure 3. Growth of emission peaks A and B shown in Figure 2.

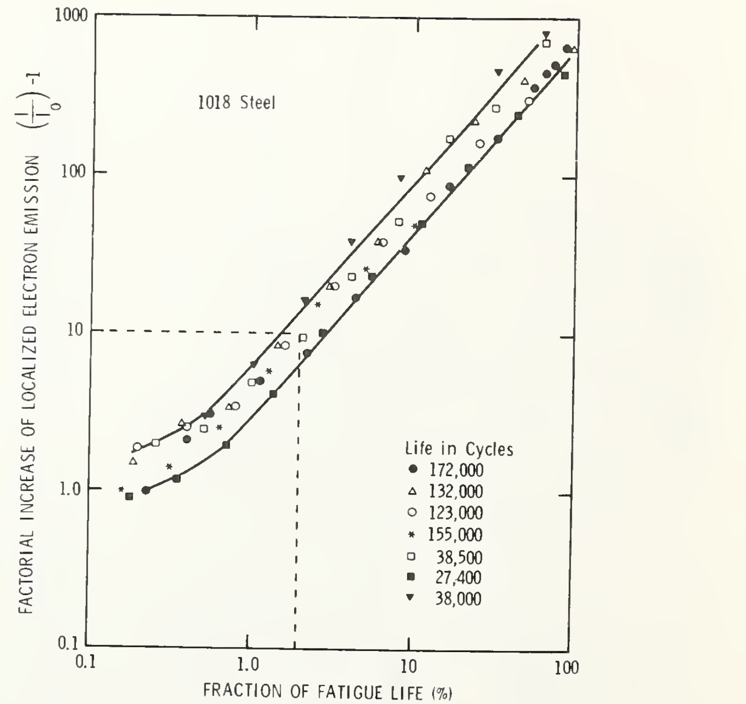


Figure 4. Growth of largest emission peak during fatigue of seven steel samples. Intensity of emission normalized with respect to initial intensity (I_o) before fatigue cycling.

DISCUSSION

P. Chiarito, NASA, Lewis Research Center: Are you able to calibrate these peaks to strain; does this really become an optical strain gauge?

W. J. Baxter: Yes, it does in a way constitute a strain gauge. The emission comes from the deformed regions and it's a micro-strain gauge, if you like.

P. Chiarito: But have you really calibrated this to strain? You said you had a strain gauge in compression. In the elastic range that would give you equal and opposites. Certainly in tension, the plastic strain on the upper surface would not be measureable by the gauge on the compression surface.

W. J. Baxter: That's right, the strain gauge breaks after a few percent of strain. Is that what you mean?

P. Chiarito: No, that isn't what I mean. You said you put a gauge on a compression side.

W. J. Baxter: That's right, and that measures the macroscopic strain that you are putting in.

P. Chiarito: On the compression side, not on the tension side.

W. J. Baxter: Yes, but it's equal and opposite to what's going on in the tensile side.

P. Chiarito: In the plastic region?

W. J. Baxter: Macroscopically it is. Microscopically the strain gauge doesn't tell you what is going on at all. Because the deformation isn't uniform.

P. Chiarito: Let me ask again, can you relate these peaks to strain so that you know, is it really a strain gauge?

W. J. Baxter: You are asking me a question which I can't answer because how can you measure the strain any other way? As far as I can see, exo-electron emission is the only way of measuring it properly. So I have no way of calibrating it.

P. Chiarito: Pardon my persistence, but can you relate the peak heights to strain?

W. J. Baxter: Yes, I can take another aluminum sample and I can deform it in tension, the same amount and I will get peaks the same height. If that's what you mean.

P. Chiarito: When you get a peak height, does this represent a certain amount of strain?

W. J. Baxter: Yes it does. I don't know how much it is though. I have no independent way of measuring it on that scale. I am measuring essentially the strain in a region 100 microns big. I have no way of telling you what the strain is in that region by any other method.

J. Bennett, NBS: Have you run stronger materials which presumably would not show the damage so soon, and also have you run specimens at stress levels where they will not break to see what the effect is?

W. J. Baxter: I haven't looked at stronger materials yet because my present system is built for little samples and it doesn't have the strength to do much more than that. I have not run tests below the fatigue limit yet. I have run tests which have lasted 7 million cycles and seen the emission in the normal way. I have also run one test for 3 million cycles and saw nothing and decided I didn't have enough time to carry on. I pulled it out and didn't continue the test. I think in that case I was below the fatigue limit.

J. Bennett: That's what I was wondering, whether you would get the increase in emission below the fatigue limit?

W. J. Baxter: I don't really know. I haven't done enough experiments to define the fatigue limit well enough to know which side of it I am on.

Sam Grant, Carolina Light and Power Co.: Do you know the source of the electrons? Are these conduction electrons?

W. J. Baxter: Out of the metal, yes.

Sam Grant: You don't get any bound electrons, they are just the free electrons?

W. J. Baxter: I believe so. I have seen no evidence whatever to suggest that they are anything other than normal electrons coming out of a piece of metal.

Sam Grant: Could you repeat for me again how you detect this? You seem to be using some other source of light and a vacuum.

W. J. Baxter: Yes, I am using ultra-violet light and that's what kicks the electrons out of the metal.

Sam Grant: The ultra-violet source is outside your vacuum?

W. J. Baxter: Yes, I just send the light beam through a window into the vacuum.

Sam Grant: Then you detect it with an electron multiplier?

W. J. Baxter: An electron multiplier, yes.

R. Hohenberg, AVCO Lycoming Division: I understand why you have to put the specimen into vacuum to measure the exo-electrons. Is it also necessary to have the specimen in vacuum during the application of load?

W. J. Baxter: I have done the following. In the case of aluminum, I've fatigued under atmospheric pressure and then pumped down to make the measurement. The effect is still there but reduced in magnitude. I've tried the same thing with steel and the effect is reduced in magnitude so much you can barely see it.

R. Hohenberg: This effect may be due to the oxidation which occurs as the fissures are created.

W. J. Baxter: That's right.

R. Hohenberg: So you would expect the temperature in an ambient atmosphere to be a very strong biasing effect?

W. J. Baxter: Yes, I think it would. The important thing is that the damage produced by the fatigue in breaking up the oxide has got to keep ahead of the reoxidation process.

Gerald Gardner, Southwest Research Institute: I would like to know precisely how you prepared the surface prior to the time it went into the vacuum chamber?

W. J. Baxter: I didn't. All I did was wash the grease off with acetone.

Gerald Gardner: You didn't go through any abrasion process originally?

W. J. Baxter: No, I did nothing.

Gerald Gardner: You evacuate the chamber and then perform the experiment?

W. J. Baxter: That's right.

Gerald Gardner: Did you attempt the experiment without the vacuum chamber at all?

W. J. Baxter: No, I didn't.

Gerald Gardner: Are you familiar with the work conducted at North American Rockwell in which they perform the experiment in air?

W. J. Baxter: I think I have seen one report on their work.

Gerald Gardner: The presence of the vacuum does two things, presumably. One is that it protects the surface from additional oxidation. In your hypothesis, this is important because that's where the exo-electrons are coming from. Another excuse one might have for the vacuum is that one needs a vacuum in order to get the electrons over to the counter. The latter is not so.

W. J. Baxter: It depends on your emission levels. If you have a high current coming out, it's okay. You will measure the ion current of the electrons which come out and probably attach to oxygen molecules.

Gerald Gardner: You get virtually 100% efficiency in electron collection that way.

W. J. Baxter: But at this emission level you won't be able to do it.

Gerald Gardner: You're satisfied you can't, you have objective evidence that you can not...?

W. J. Baxter: Well, you tell me how you can.

Gerald Gardner: Just run the experiment with the air in it.

W. J. Baxter: How are you going to measure it at 10^{-16} amps?

Gerald Gardner: With a picoammeter. I won't pursue this any further. However, it has been done and it's not all that difficult. There is one other question which I would like to ask you. I can't recollect the individual who performed the experiment but about 7 years ago a very similar experiment was performed in France in which a specimen was illuminated with a flying spot derived from a laser. They used a double arrangement to create a raster and actually looked at a fatigue crack as it developed.

W. J. Baxter: Were you thinking of the University of Delph? They used a mercury arc, I believe, and a geiger counter as the detection system.

Gerald Gardner: I just want to know what relation you think your work bears to theirs?

W. J. Baxter: I think it bears a very close relationship and I think the reason that their results apparently didn't lead to anything was that the gas in the geiger counter interfered with this whole business. They couldn't get systematic data. All they showed, if you remember, was just a picture of a crack.

Gerald Gardner: Did you examine the specimen afterwards to determine whether the site at which the failure crack ultimately did initiate was in any way associated with any metallurgical defect that was there originally?

W. J. Baxter: You mean fatigue it, take it out, look at it, and then put it back in and finish it off?

Gerald Gardner: No, just when it fractured were you able to look back at the fracture surface and see if there was something that should have picked that point out?

W. J. Baxter: No.

Bruce Christ, NBS: What precautions did you take with the vacuum system and what vacuum pressures did you have to operate at to prevent reoxidation or other contamination of the surface?

W. J. Baxter: Well, as one usually does, you use what's on hand. I happen to have quite a decent vacuum system and it normally runs at 10^{-8} torr. I am looking into the question of how bad a vacuum I can put up with. I don't really have a firm answer yet.

Charles Jackson, Monsanto: I was trying to relate to the fatigue gauge which has great application, at least from a practical standpoint, if the levels are like 1200 macrostrain or greater. Is your objective to develop a practical application similar to or possibly better than the fatigue gauge?

W. J. Baxter: By fatigue gauge, do you mean strain gauge?

Charles Jackson: Not strain gauge, strain hardening sensitive gauge which changes resistance with fatigue. It doesn't measure direct strain, but is strain sensitive, naturally, but it primarily changes resistance with fatigue.

W. J. Baxter: That's right, my objective is to develop a system where one can have an abbreviated fatigue test without running to failure.

Charles Jackson: Suppose you put a fatigue gauge on the tension side which does correlate not only to strain, but also to strain hardening which is a measure of change of resistance to fatigue life. I am curious if an experiment wouldn't be valid observing the surface of the fatigue gauge. The S/N fatigue gauge is now marketed by Micro Measurements.

W. J. Baxter: You're changing the whole experiment. You are asking me now to detect deformations in the strain gauge which is a different material all together.

Charles Jackson: I guess I keep objecting to you using the term strain gauge. I'm talking about a fatigue gauge.

W. J. Baxter: It is a thing that is bonded onto the surface, right?

Charles Jackson: True--intended to measure fatigue, though.

W. J. Baxter: What is the thing made of?

Charles Jackson: I don't know. It's proprietary.

W. J. Baxter: It may behave like a thick plastic coating. Is it encapsulated?

Charles Jackson: The aluminum and steel bars that I have broken have behaved very consistently. That is, on steel, if you pick up 5 or 6 ohms change in resistance, you've got a crack. I was curious if you've correlated to this?

W. J. Baxter: No, I haven't tried using those things at all.

J. Thorp, U.S. Army Aviation Systems Command: Was this aluminum or steel previously cold worked or was it annealed?

W. J. Baxter: It's as received. The steel is cold rolled and I assume that the aluminum is rolled.

J. Thorp: I was wondering if there should be a difference between the annealed and the cold worked material. If you tested both conditions, would you get a different rate of electron emission?

W. J. Baxter: You might.

APPLICATION OF AUTOMATED DIAGNOSTICS IN MOTOR VEHICLE SAFETY INSPECTION

By

JOHN L. JACOBUS

Highway Safety Research Engineer

National Highway Traffic Safety Administration

Department of Transportation

400 - 7th Street, S.W.

Washington, D.C. 20590

Presented To:

18th Meeting

Mechanical Failures Prevention Group

Detection, Diagnosis, and Prognosis

November 9, 1972

At the

National Bureau of Standards

Gaithersburg, Maryland

APPLICATION OF AUTOMATED DIAGNOSTICS IN MOTOR VEHICLE SAFETY INSPECTION

INTRODUCTION

In order to solve the problems of motor vehicle safety inspection, many new approaches in sensing and processing will be required which will challenge all segments of the scientific and engineering community. These new approaches are embodied in an Automated Diagnostic Inspection System concept.

The National Highway Traffic Safety Administration considers automated diagnostic inspection systems to mean the application of computerized inspection equipment and electronic data processing techniques to first inspect or identify those automobile systems which pose a safety hazard and, second, to diagnose or determine those subsystems or components causing the hazardous condition.

We believe that automated diagnostic inspection systems are an answer to upgrading and improving State inspection programs so as to reduce the accident potential of defective vehicles as well as to remove unsafe automobiles from our roads and highways.

THE NEED FOR AUTOMATED DIAGNOSTIC INSPECTION SYSTEMS

The need for automated diagnostic inspection systems is obvious when considering current motor vehicle inspection programs which comply with the spirit of Federal recommendations and fail to objectively and accurately inspect automobile systems.

Something must be done to upgrade the "safety quality" of the vehicles on our roads while maintaining the ability to cope with current and future numbers of automobiles.

As more safety critical areas are identified and confirmed by research, and the consumer demand for in-depth automobile diagnostics increases, automated diagnostic inspection techniques will have to be increasingly utilized.

EXISTING DIAGNOSTIC AND INSPECTION TECHNIQUES AND UNIVERSAL DRIVE-ON AND ON-BOARD PLUG-IN METHODS

NHTSA feels that universal drive-on inspection equipment, coupled with passive on-board sensors and plug-in umbilical cord, are a means to upgrading all inspection programs.

In terms of dynamically checking brakes, steering and suspension, and tire systems: visual inspection methods, active on-board transducer methods, and self-diagnosing component techniques do not seem comprehensively applicable.

Universal drive-on inspection equipment is needed that can accommodate any vehicle wheel base configuration and on-board transducers which will allow the usage of stationary electronic data processing (EDP) equipment and computer techniques.

SPECIFIC PERFORMANCE CHECKS AND METHODS

Drive-on equipment is necessary to perform five primary brake system performance checks which consist of (1) system gain, (2) system response, (3) fade, (4) front-to-rear proportioning, and (5) side-to-side balance.

Areas requiring plug-in techniques and on-board sensors would include:

1. Brake shoe/pad thickness
2. Brake shoe/pad temperature
3. Brake line pressure
4. Master cylinder fluid level
5. Brake pedal reserve
6. Anti-skid device

Three primary performance checks are required to inspect automobile steering and suspension systems. These checks include the following:

1. Steering response or steering wheel play
2. Front-end parameters and ball joints
3. Shock absorber/spring subsystem response

Areas requiring plug-in techniques include a steering wheel servo unit to measure steering wheel displacement as well as torque transducers to relate front wheel brake imbalance to steering wheel inputs. It may also be desirable to incorporate transducers to indicate tire/wheel imbalances.

Tire pressures and tread depths are considered safety critical parameters and also require drive-on equipment.

TECHNOLOGY FOR AUTOMATED DIAGNOSTIC INSPECTION SYSTEMS

Commercially available chassis-type dynamometers, chassis-type brake analyzers, front-end analyzers, and shock absorber testers are all readily adaptable to computerization.

An Indiana manufacturer has developed, for its rent-a-car division, an automatic electro-mechanical device to measure tire tread depth and tire pressure.

A Minnesota manufacturer has developed and is offering for sale, a computerized vehicle diagnostic system applied specifically to engine analysis.

Volkswagen is in the process of serving their new vehicle purchasers with a plug-in diagnostic package facilitating the use of a computer which is plugged into a central socket in the engine compartment.

ADVANCED INSPECTION SYSTEMS

Several advanced inspection concepts have been proposed which show great potential as inspection tools. These include:

1. Gas chromatography
2. Fiberoptics
3. Holography

The use of gas chromatography techniques has been proposed for indicating shock absorber and brake fluid leakage. A probe or "sniffing device" would sample and analyze the air around the shock absorbers, brake line/hose connections, and brake hose/wheel cylinder connections. These "sniffers" could also be used to detect significant exhaust, fuel line and fuel tank leakages.

Fiberoptic techniques have been suggested as a means of indicating roughness and scoring of brake drums and discs. For drums, a fiberoptic probe would be inserted through an access hole in the backing plate of each wheel which would then scan the width of the drum. Reflected light intensity would be compared to a standard to indicate outages.

Holography techniques have been proposed as a means of inspecting tires and brake hoses. Defects such as ply separations and structural discontinuities would show up as areas of high stress concentration in the holographic print.

APPEARANCE & OPERATION OF AN AUTOMATED DIAGNOSTIC INSPECTION SYSTEM

A hypothetical automated diagnostic inspection system will be discussed employing modified commercial and prototype equipment with advanced inspection concepts. This system has been designed to demonstrate the compatibility of engine and emissions inspection with brakes, steering and suspension and tires inspection.

The first element of this system would be an automobile equipped or manufactured with various on-board transducers connected to one central plug on the firewall or in the engine compartment.

The second element of this automated diagnostic inspection system is "slave" actuation equipment for the brake and accelerator pedals as well as steering wheel.

The third element of this inspection system consists of integrated, universal drive-on equipment.

The fourth element of this inspection system consists of computer hardware and software with an appropriate time sharing system which would be connected to a national computer network.

Inspection for this proposed automated diagnostic inspection system would consist of the nine system performance tests previously discussed, plus an exhaust emissions performance test. If the vehicle fails to pass any one of these ten performance checks, the computer logic will utilize the additional information obtained through the on-board sensors to diagnose the cause of failure.

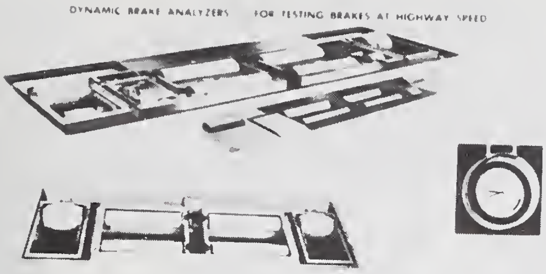
CONCLUSIONS

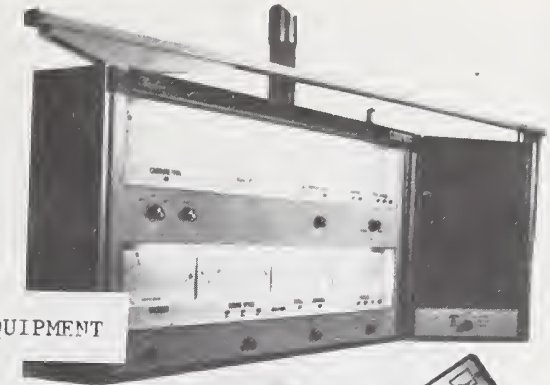
In order to accurately and quickly inspect exhaust, brakes, steering and suspension, and tire systems, automobiles will have to be manufactured with passive on-board transducers.

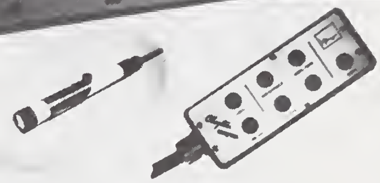
In order to facilitate speed and repeatability, "slave" actuation devices and electronic data processing equipment must be interfaced with on-board sensors and universal drive-on equipment.

Integrated, universal drive-on inspection equipment will have to be developed so that multiple inspection functions can be performed simultaneously by one machine.

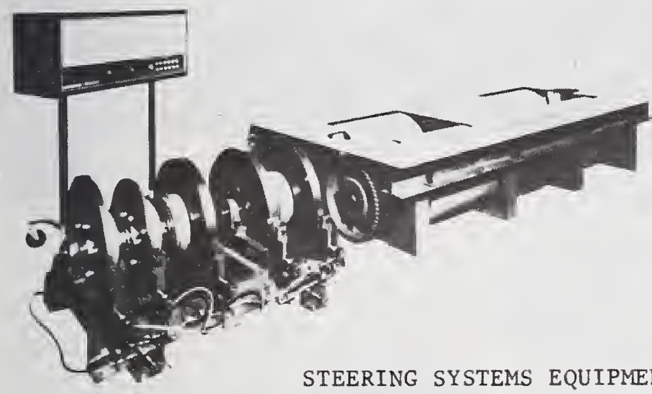
NHTSA feels that automated diagnostic inspection systems will provide a means of objectively and accurately inspecting hundreds of thousands of cars quickly and inexpensively.

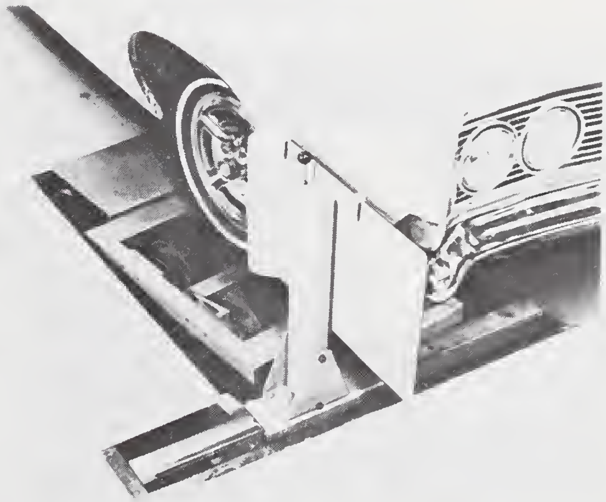






BRAKE SYSTEMS EQUIPMENT

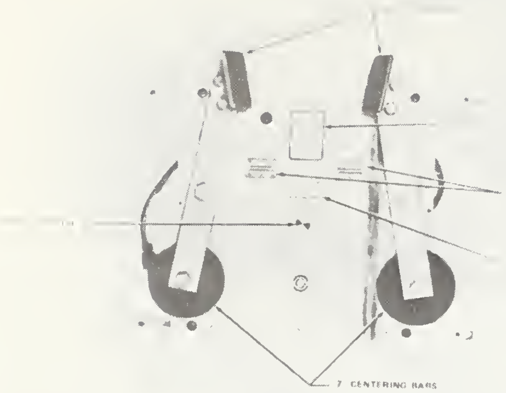




STEERING SYSTEMS EQUIPMENT

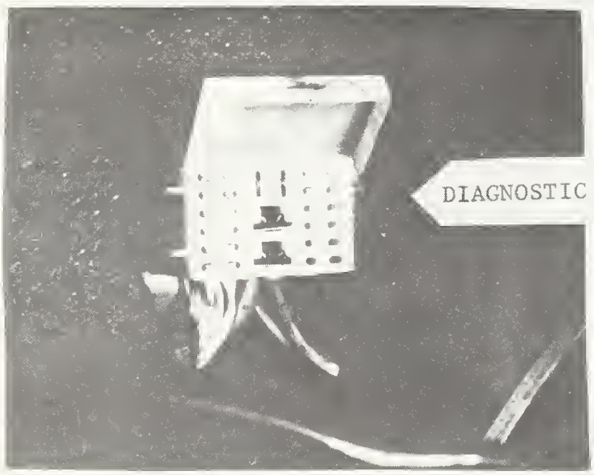
SUSPENSION SYSTEMS EQUIPMENT

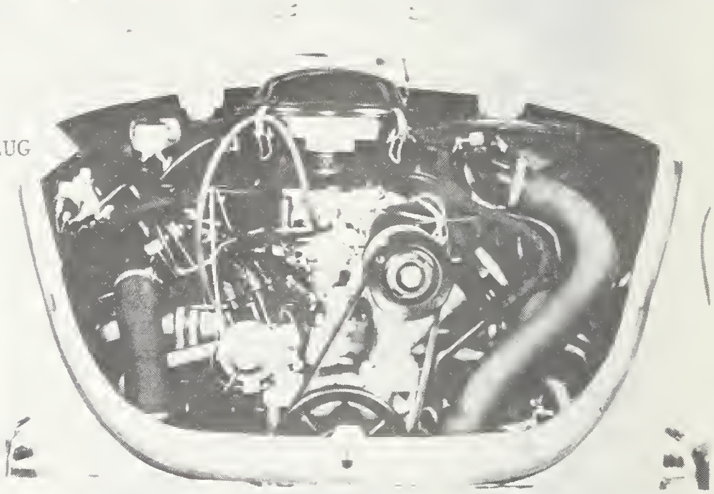
COMMERCIALLY AVAILABLE DIAGNOSTIC INSPECTION EQUIPMENT



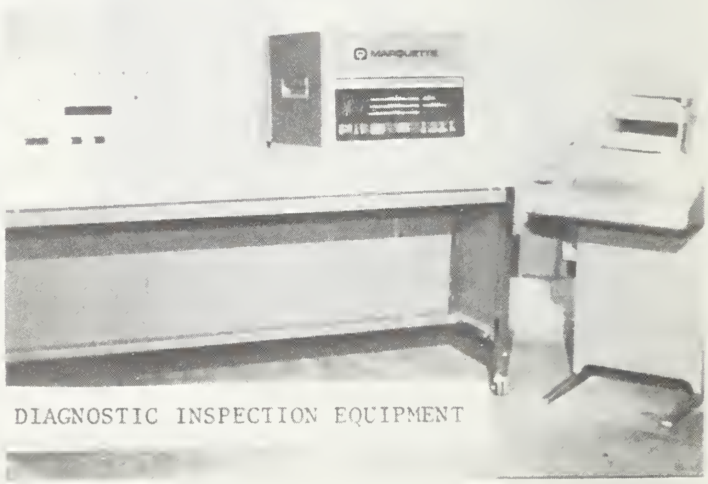
TIRE SYSTEMS EQUIPMENT

"SLAVE" ACCELERATOR PEDAL ACTUATOR

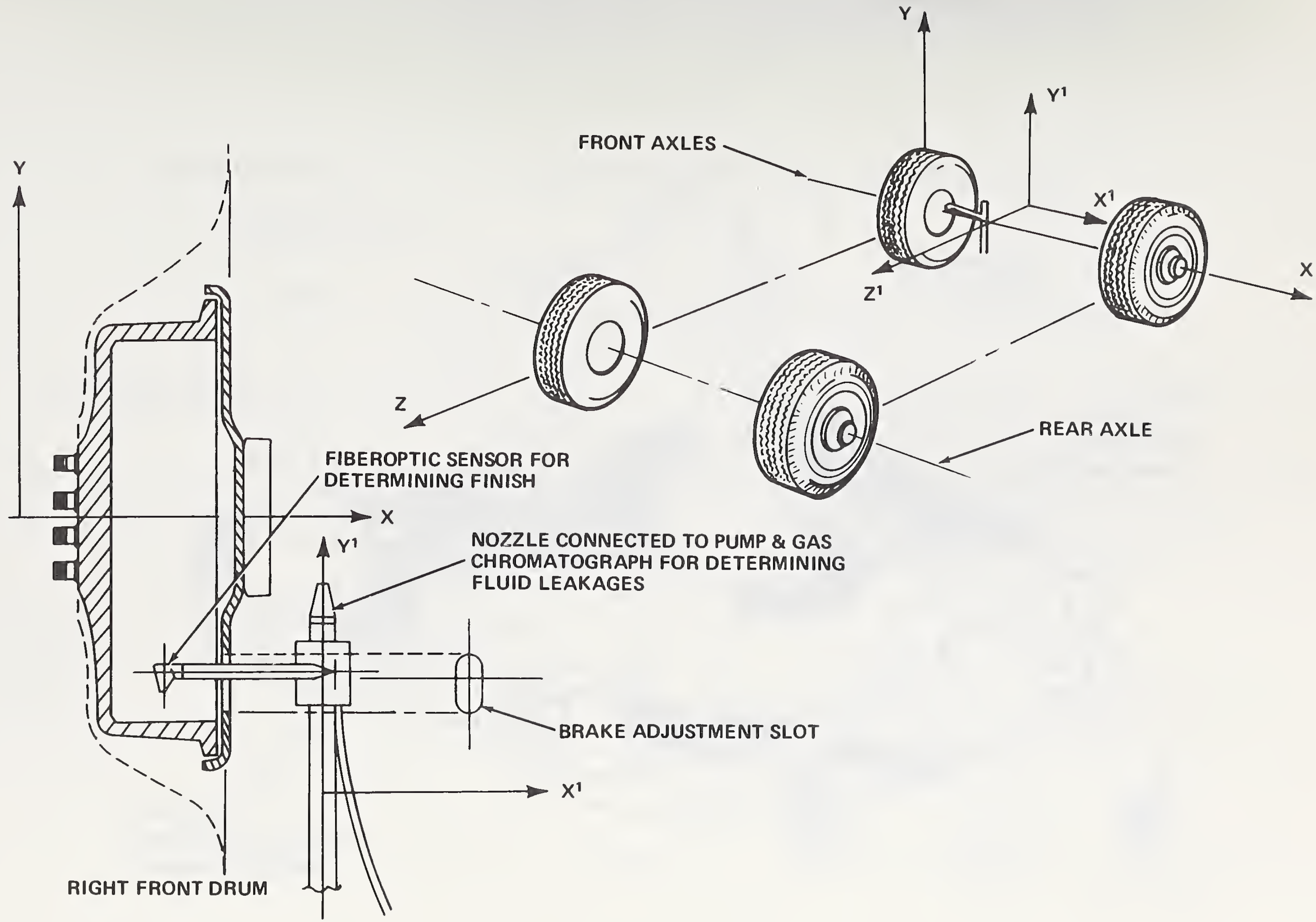




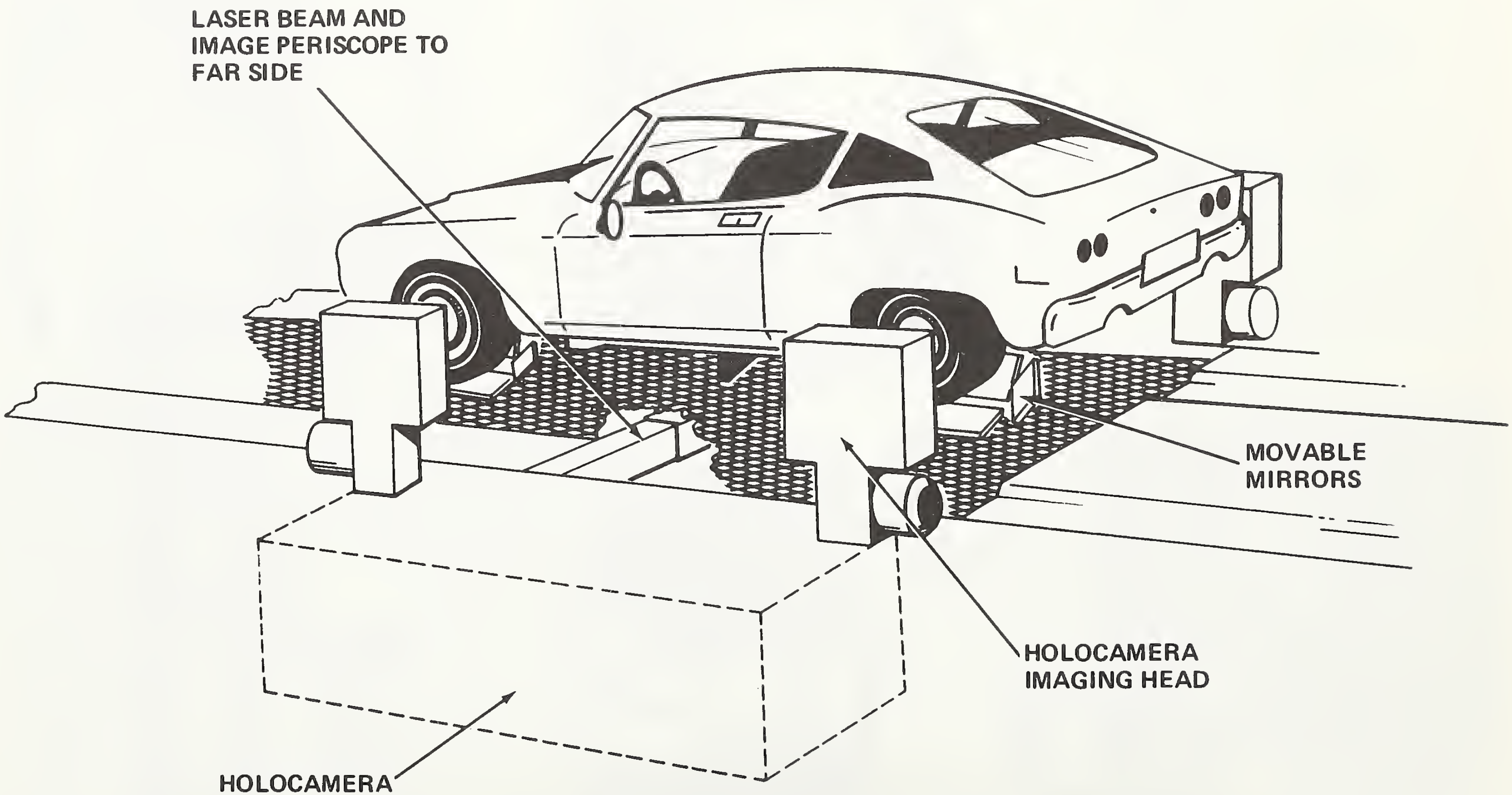
COMPUTER AND ELECTRONIC DATA PROCESSING EQUIPMENT



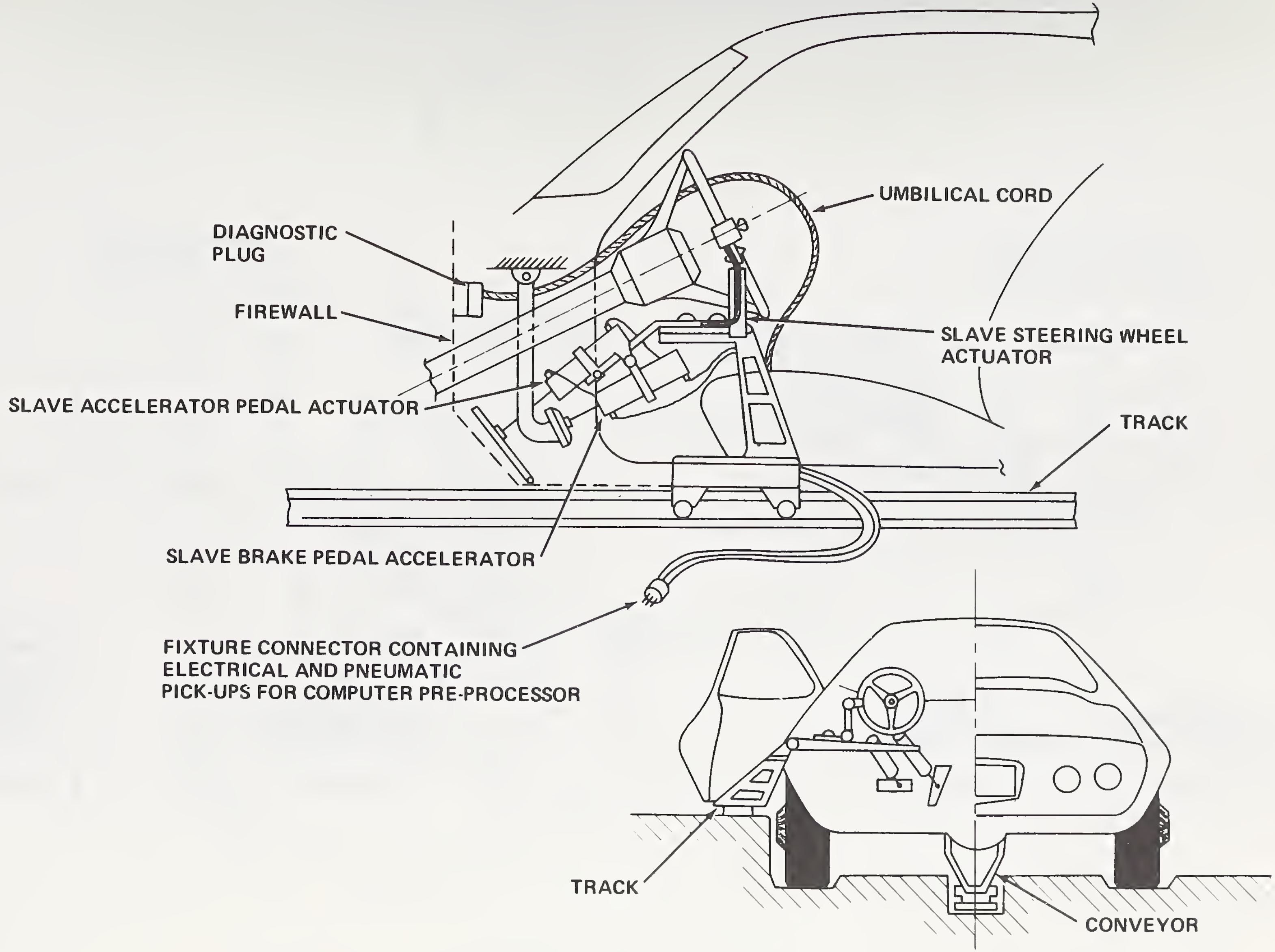
COMMERCIALLY AVAILABLE DIAGNOSTIC INSPECTION EQUIPMENT



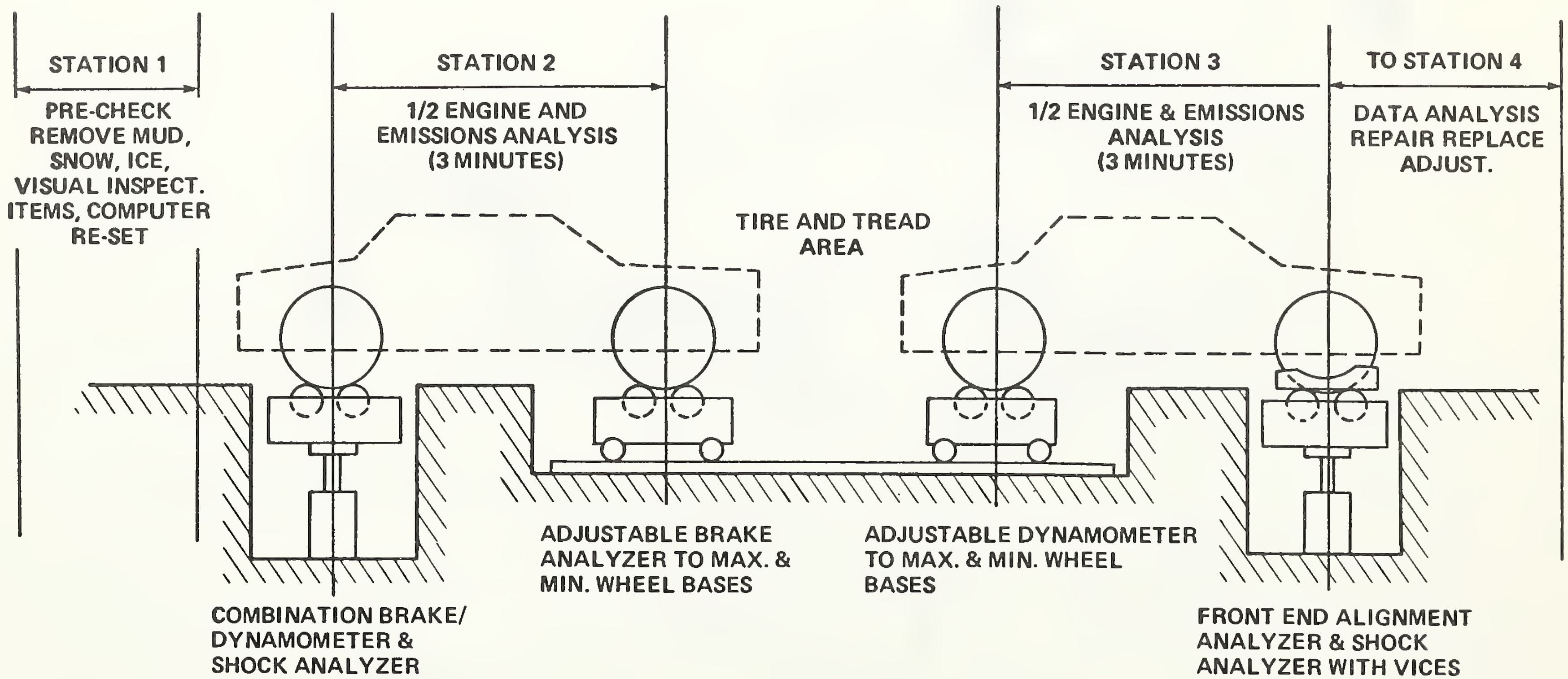
Brake Fluid, Shock Absorber Fluid and Fuel Leakage Detector with Integral Brake Drum Surface Finish Indicator



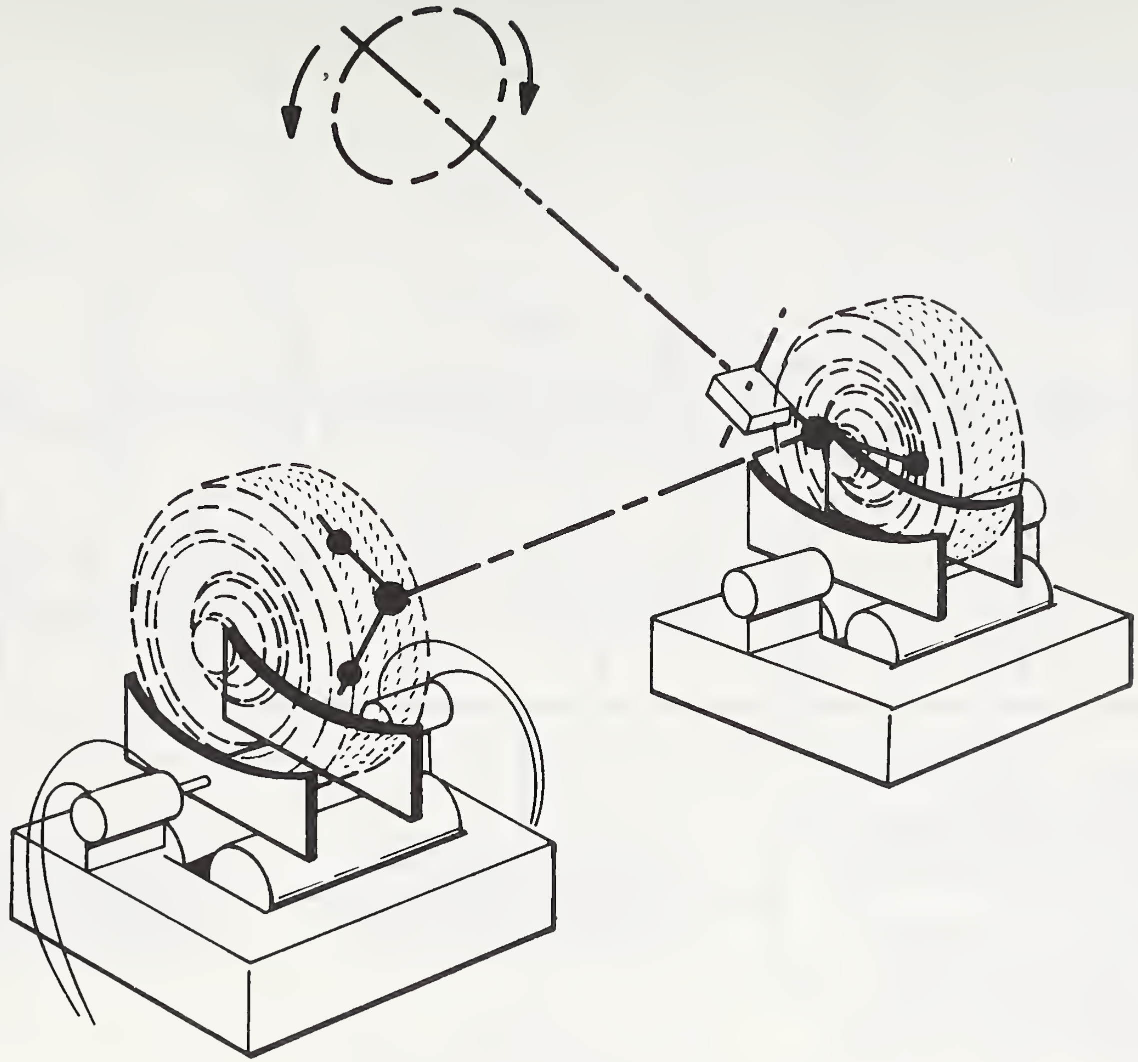
Holography Tire Inspection Equipment



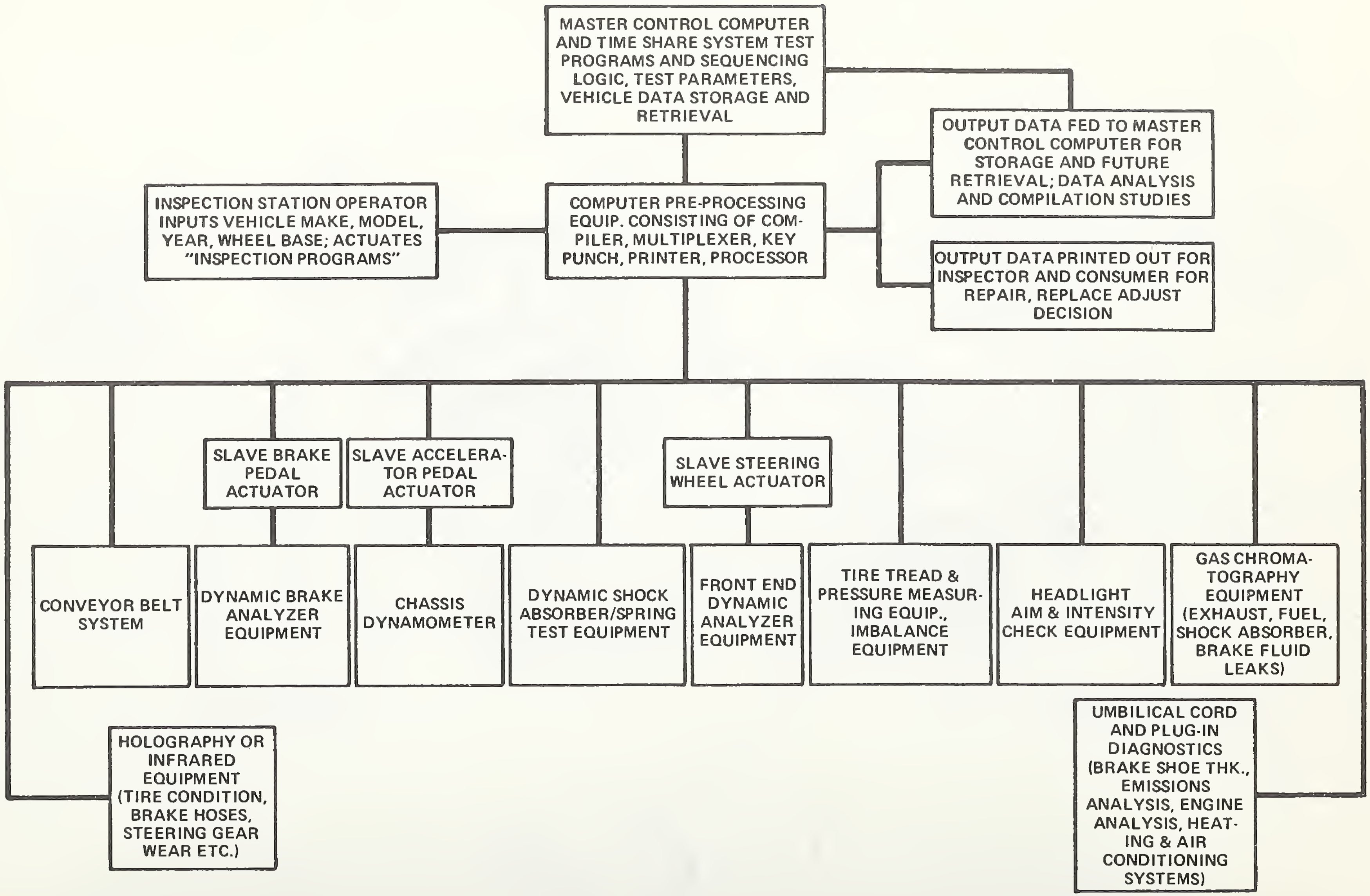
"Slave" Brake and Accelerator Pedal Actuator Equipment



Automated Diagnostic Inspection Equipment Layout



Dynamic Front End Analyzer with Piston Actuated Tire Vices



Inspection Station Diagram

AUTOMATED DIAGNOSTIC INSPECTION SYSTEM COMPUTER PRINT OUT

Measure-ment	Specs.	Pass	Fail	Repair	Replace	Adjust

BRAKE SYSTEMS INSPECTION

1. Brake System Gain
2. Brake System Response
   a. Activation
   b. De-Activation
3. Braking Ratio
4. Brake Balance
   a. Front
   b. Rear
   c. Front Drag Imbalance
   d. Rear Drag Imbalance
5. Brake System Fade

BRAKE SYSTEM DIAGNOSTICS

A. Brake Pedal Force
B. Brake Pedal Reserve
C. Brake Pedal Travel
D. Brake Shoe/Pad Thickness
   a. RF
   b. LF
   c. LR
   d. RR
E. Brake Line Pressure Hold Checks
   a. RF
   b. LF
   c. LR
   d. RR
F. Brake Hose Chafing
   a. RF
   b. LF
   c. LR
   d. RR
G. Brake Drum/Disc Roughness
   a. RF
   b. LF
H. Brake Drum Roughness
   a. RR
   b. LR
I. Master Cylinder Fluid Check
J. Vapor Lock Check
K. Anti-Skid Check

AUTOMATED DIAGNOSTIC INSPECTION SYSTEM COMPUTER PRINT OUT

STEERING & SUSPENSION SYSTEM INSPECTION

6. Steering Response
7. Front End Parameters & Ball Joints
   a. Toe
   b. Camber (RF)
   c. Camber (LF)
   d. Caster (RF)
   e. Upper/Low Ball Joints (RF)
   f. Upper/Low Ball Joints (LF)
   g. Caster (LF)
8. Shock Absorber Response
   a. RF (High/Low Frequency)
   b. LF (High/Low Frequency)
   c. RR (High/Low Frequency)
   d. LR (High/Low Frequency)

STEERING & SUSPENSION SYSTEM DIAGNOSTICS

A. Maximum Steering Angle
B. Tracking Check
C. Independent Rear Camber
   a. RR
   b. LR
D. Independent Rear Toe
   a. RR
   b. LR
E. Tie Rod End Wear
F. Idler Arm Wear
G. Steering Arm Joint Wear
H. Shock Absorber Leakage
   a. RF
   b. LF
   c. RR
   d. LR

TIRE SYSTEMS INSPECTION

9. Tire Pressure & Tread Depth
   a. RF
   b. LF
   c. RR
   d. LR

TIRE SYSTEM DIAGNOSTICS

A. Tire Mix Check
B. Wheel/Tire Balance
   a. RF

b. LF
c. RR
d. LF

C. Tire Condition
a. RF
b. LF
c. RR
d. LR

ENGINE & POWER TRAIN INSPECTION

10. Emissions
A. Hydrocarbons
B. Nitrous Oxides
C. Carbon Monoxide

MISCELLANEOUS SYSTEMS INSPECTION

11. Fuel Systems
A. Fuel Tank Leakages
a. Front
b. Rear
B. Fuel Line Leakages
a. Front
b. Rear
12. Exhaust Systems
A. Front Leakage
B. Rear Leakage
13. Headlight Systems
A. Aim
B. Intensity
14. Air Conditioning System
15. Heating System
16. Cooling System

APPLICATION OF OPTICAL PROCESSING TO FLAW DETECTION

Bill Baker, Hugh Brown, Bob Markevitch, Dave Rodal
Ampex Corporation, Redwood City, California 94063
Telephone (415) 367-3135

Optical processing is an improved method of spectral analysis which can be used to monitor the noise or vibrations of rotating machinery or to analyze the response of a structure subjected to a known dynamic loading. A change in the noise spectrum or signature of a device can sometimes be interpreted as an indicator of a potential failure.

Important characteristics of a spectrum analyzer are its resolution and its ability to extract the significant characteristic frequencies of a device from extraneous background noise. The capability of the spectral analyzer can be described in terms of its time bandwidth product or the number of data samples which are stored and, later, treated in the process of analysis. For conventional digital or analog analyzers, the number of samples may be only of the order of 10^3 digital words or analog values. On the other hand, 10^5 to 10^6 samples may be stored on a square centimeter of film which is utilized in the optical processor. As in most systems, it is possible to trade off bandwidth versus the recording time interval. The resolution of the optical system is always of the order of 10^{-5} to 10^{-6} times the bandwidth.

An application to the analysis of a simple structural panel is indicated in Fig. 1. The specimen is stimulated by a transducer driven by a signal generator. A second transducer provides the output signal which describes the response of the plate at a given point. The optical recording is simply a photograph of the face of a CRT which is scanned in a manner similar to that of a TV tube with the intensity of the beam being modulated by the incoming signal.

The film is chemically processed in a few seconds with a Bimat technique and is then ready for spectral analysis in the optical processor in the lower portion of Fig.1. The spectrum is focused on the face of a TV camera which is used to produce displays on an orthographic TV screen and on a 3D or isometric screen at the right.

A minicomputer has also been tied into the system to perform certain calculations, to display alphanumeric data on the TV screens and tó produce hard copy of special data.

A schematic presentation of the optical principles governing the formation of the spectrum or Fourier transform is shown in Fig. 2. This includes more details of the optical processor in the lower part of the system diagram of Fig. 1.

At the upper left portion of Fig. 2 is the previously recorded and developed film which contains the information of the input signal stored as variations in optical transmittance. The square of film in the X-Y plane is a time exposure photograph made as the beam scanned the face of the CRT as indicated by the parallel equally-spaced lines.

Of course, several hundred lines are scanned in the usual square centimeter format, and the line transmittance varies as the strength of the signal at the recording time corresponding to each point on the lines. However, let's temporarily consider the uniformly spaced lines to be of constant transmittance so they form a simple grating.

If the grating is illuminated by a coherent monochromatic light source, such as a laser beam, then at great distances diffraction produces bright dots along a line such as the f_x axis perpendicular to the lines of the grating. There is a bright spot called the zero order on the line normal to the film plane. The next bright spot, labeled "line rate", corresponds to f_o the spatial frequency of the grating. Since the transmittance grating is not perfectly sinusoidal, there are also bright spots at the harmonics $2f_o$, $3f_o$, etc., and because of symmetry, there are corresponding dots at $-f_o$, $-2f_o$, $-3f_o$, etc. in the opposite direction from the zero order. These line rate dots and a few harmonics can even be observed by looking through a grating at a small, distant light bulb in a darkroom.

A theoretical derivation of the amplitude of light in a plane far from the film is presented by Goodman (1). The resulting simple form is given by the integral at the bottom of the figure. That is, the amplitude A_o is directly proportional to the Fourier transform of the transmittance t_1 extended over the entire uniformly illuminated aperture. The introduction of the transform lens preserves the relation and allows the two planes to be brought conveniently close together.

By further calculations, it can be shown that the Fourier transform of the modulated raster scan recording has the following properties:

1. The presence of any coherent component in the recording will produce a bright spot in the frequency plane.

2. That spot must lie on one of the frequency loci which form another raster indicated by the inclined parallel lines passing through the line rate spot and its harmonics.

3. Each spot is repeated periodically in the f_x direction with period f_o.

4. In a manner similar to that of the time raster in the film plane, motion along a locus corresponds to a continuous variation of frequency, but motion in the positive f_y direction to the next locus corresponds to an increase in frequency by exactly the line rate f_o.

Because of the periodicity in the frequency plane, the entire spectrum is displayed in any vertical half strip of width f_o. It is convenient to use the shaded strip of Fig. 2 bounded on the left by the f_y - axis, below by the f_x - axis, and on the right by the vertical line $f_x = f_o$.

Optical processing has been used to produce spectra of engines and other machines. Early work included ONR sponsored spectral analysis of acoustic and accelerometer data recorded from a Pratt and Whitney jet engine. Those studies (2,3,4,5) clearly indicated many frequencies including vibrations caused by the chopping of the air stream in the compressor and turbine wheels or by the meshing of accessory drive gears. Many of these components had not been identified by other methods of spectral analysis.

In order to compare the signatures of worn and new components, a jig equipped with an accelerometer was constructed to test starter bearings from the jet engine. Eastern Airlines supplied a worn bearing which was compared with a new replacement bearing. In addition to the generally noisy spectrum due to minute spalling of the contact surfaces, the worn bearing also showed strong coherent content corresponding to the ball spin frequency. There was also an unexpected component at a frequency considerably lower than the theoretical spin frequency for the given bearing geometry. Closer inspection of the bearing revealed that most of the balls were so badly worn that they slipped and spun at the lower rate rather than at the theoretical one.

Spectra have also been run on reciprocating air compressors and internal combustion engines. It has been found that improperly assembled bearings produce distinct characteristic changes in the signature.

Current work includes a feasibility study of the use of optical processing in the detection of structural flaws under NASA contract NAS1-11749. Preliminary experiments have been made on simple rectangular aluminum plates with various combinations of holes or simulated flaws. The spectra change noticably as holes are introduced or as cracks are lengthened. Studies of jet engine spectra are also being continued.

Because of the large time bandwidth product of the optical processing system, much finer frequency resolution and improved signal to noise ratio are obtained. Use of this method of spectral analysis holds promise of providing diagnostic information for routinely monitoring the condition of engines and critical structures. Such techniques can increase the safety and service factors of equipment over conventional time based maintenance methods.

Comparisons based upon high data rates (6) shows that optical spectrum analyzers cost only about 10^{-3} to 10^{-2} times as much as hard wired fast Fourier transformers.

REFERENCES

1. Goodman, J. W., Introduction to Fourier Optics, McGraw-Hill, New York, 1968.

2. Markevitch, Bob V., and van Heeckeren, Jacob, Research on Failure Indicator System, First Technical Report, Office of Naval Research Contract No. N00014-67-C-0401.

3. Knight, Gordon R., Research on Failure Indicator System, Second Technical Report, Office of Naval Research Contract No. N00014-67-C-0401, July, 1969.

4. Knight, Gordon, R., Research on Failure Indicator System, Third Technical Report, Office of Naval Research Contract No. 00014-67-C-0401, July 1970.

5. Elson, Benjamin M., "Optical System Analyzes Engines", Aviation Week & Space Technology, March 1, 1971.

6. Oliver, Bernard M., Project Cyclops, NASA Rpt. CR 114445, 1971.

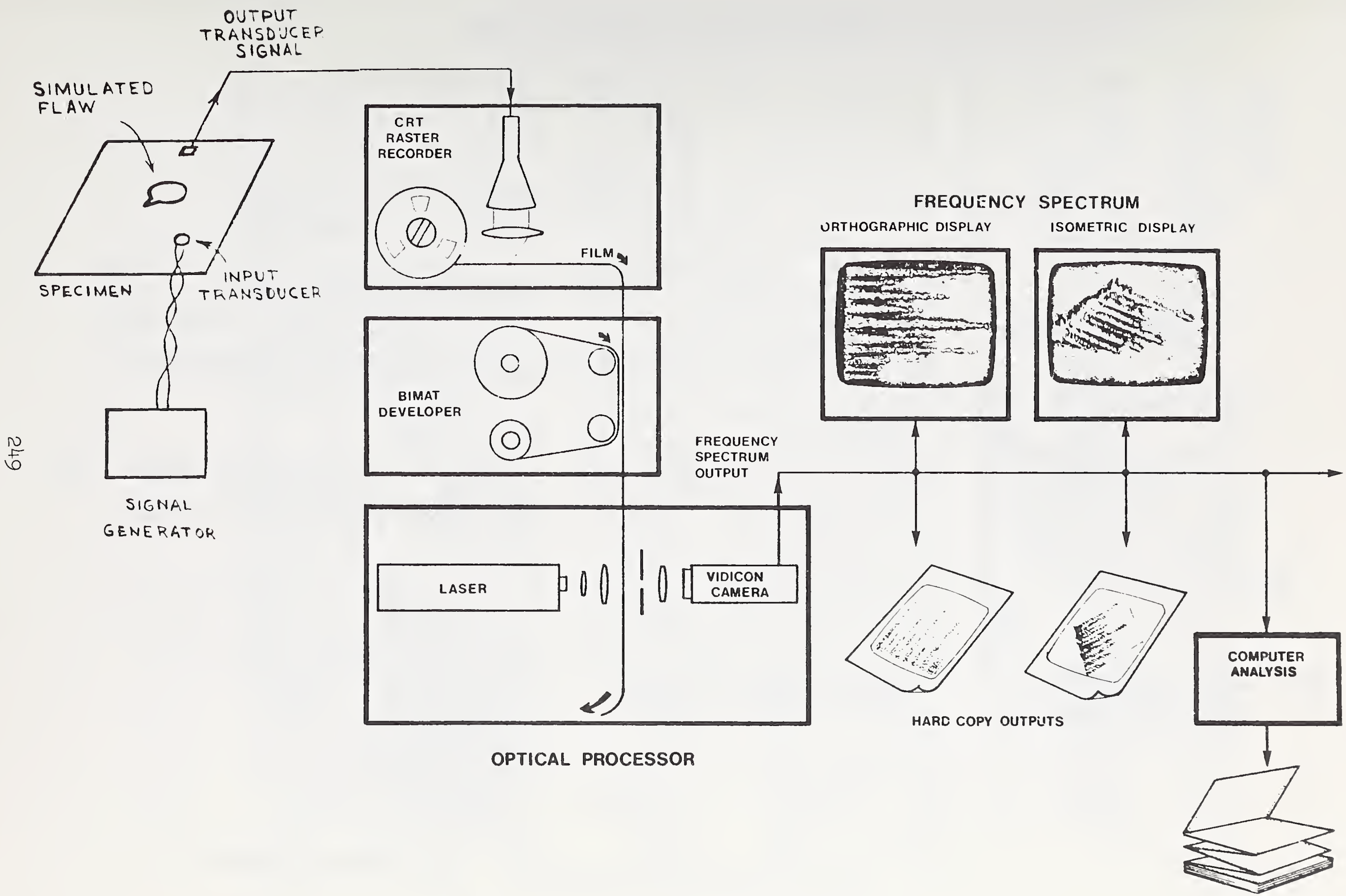
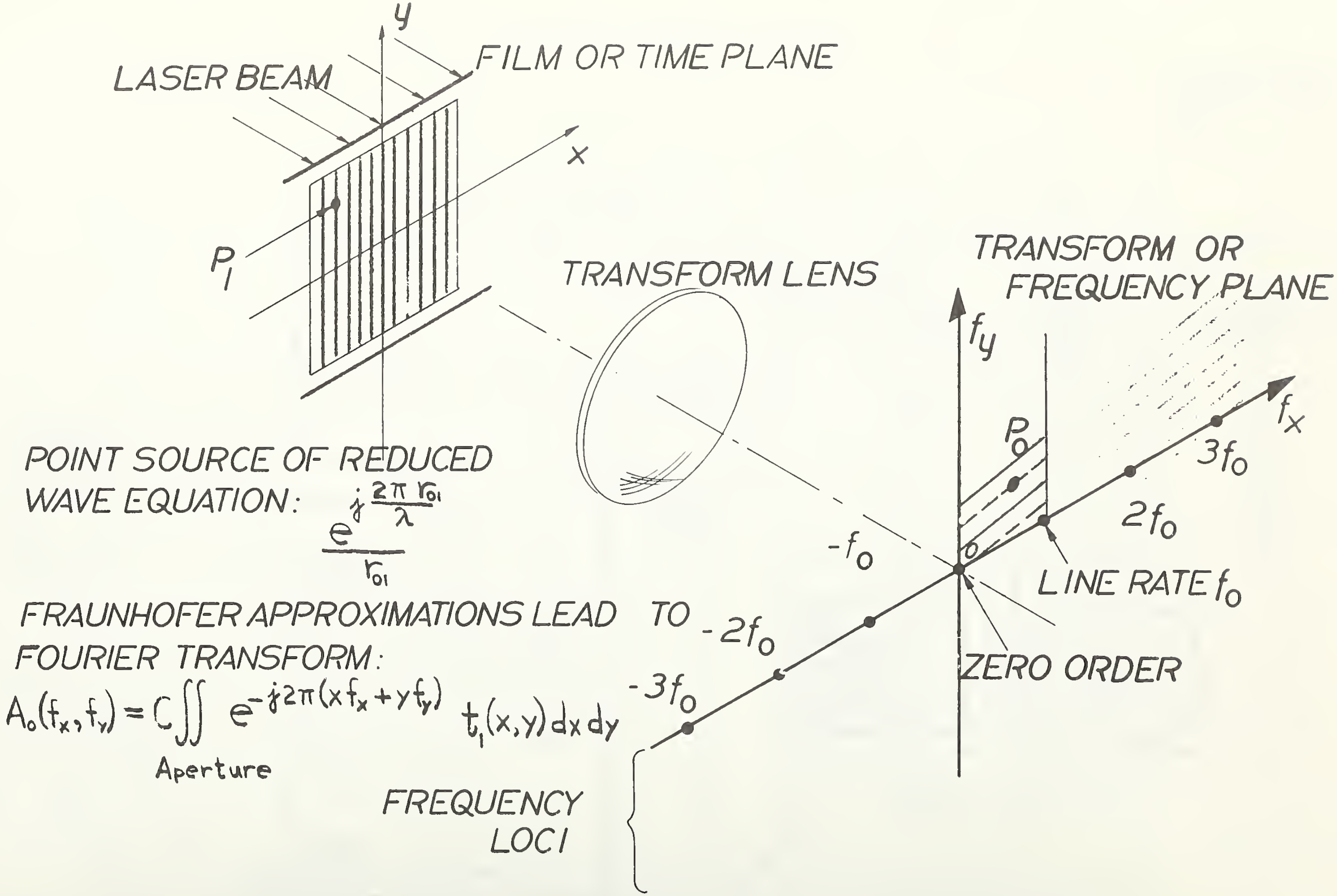


Fig. 1 Schematic Diagram of Apparatus



$$\frac{e^{j\frac{2\pi r_{01}}{\lambda}}}{r_{01}}$$

$$A_0(f_x, f_y) = C \iint_{\text{Aperture}} e^{-j2\pi(xf_x + yf_y)}\, t_1(x,y)\, dx\, dy$$

Fig. 2 Formation of Fourier Transform

DISCUSSION

J. Sudey, NASA: You showed a comparison between a conventional analysis and your optical technique. Can you indicate what the band width on a conventional analyzer was?

B. Baker: I believe that the comparison was made by taking successive segments in the spectrum and analyzing those. I don't know whether the storage capacity is really available at any time to handle this many sample points, 300,000 data points. I'm not sure whether that can even be done.

J. Sudey: The band width then is somewhat dependent upon or related to the data points?

B. Baker: Yes. This is roughly equal to the time band width product. If you want to record a certain frequency, a certain band width, then you have to have samples at roughly twice that rate. That is a minimum rate. Actually, since bands are usually not very sharp, you usually want to process three times the highest frequency. You have to sample at that rate for a certain time to produce a certain number of samples. That number of samples is the number of data points that we can store on this square centimeter of film. Any other system would have to be able to take care of that many elements.

S. Ramsey, Stanford Research Institute: One point that should probably be made is the time it takes to perform the Fourier transform optically, as opposed to visually, is the time it takes to propagate from the film to the transform plane, which may be several nanoseconds. If you are going to do the transformation digitally, it may take up to several hours. Computer costs might be somewhat prohibitive.

B. Baker: That's right. The computer costs are so high because of the amount of processing required. At that rate, it would require a tremendous amount of digital systems. I should have mentioned that in the brainwave studies, we are doing some work for medical people where they bring in tape recordings of data. They are studying different things such as fetal response and response of subjects to drugs, etc. We are interested in low frequencies, but they have a long time interval. We are not interested in a very wide band width, so we can get a tremendous time compressed into one centimeter of film. We have been able to play the tape at high speeds onto the film and we can get 30 to 40 minutes worth of data. It might have taken the computer center a week or so to get that spectra back from that amount of data. It is enormously fast, not just because of the propagation, but because we can compact so much data in one small area.

MECHANICAL FAILURES PREVENTION GROUP

18th MEETING

Detection, Diagnosis, and Prognosis

Sponsored by: National Bureau of Standards

November 8, 9, 10, 1972

Albert Aksomitas
Turbo Power & Marine Sys. Inc.
Pratt & Whitney Aircraft Co.
70 Ox Yoke Drive
Wethersfield, Conn. 06109

D. P. Altholz
U. S. Army Aviation
Systems Command
P. O. Box 209 - Main Office
St. Louis, Missouri 63166

John J. Anderson
Federal Scientific
15 Lake Drive
Boonton, New Jersey 07005

Alan E. Ankeny
Naval Air Development Center
Code 302119
Warminister, Pennsylvania 18974

Cdr. F. A. H. Ashmead RN
British Defence Staff
British Embassy
3100 Mass. Avenue, N. W.
Washington, D. C. 20008

W. H. Austin
Naval Ship Engineering Center - SMMSO
Center Bldg, Prince George's Center
Hyattsville, Maryland 20782

Alexander S. Babkin
Korfund Dynamics
Cantiague Road
Westbury, New York 11590

Bruce C. Baird
The Boeing Company
P. O. Box 58747
Houston, Texas 77058

Barry Baskett
U. S. Army Aviation Sys. Command
12th & Spruce Streets
St. Louis, Missouri 63166

LTC Richard B. Baxter
OPM Util Acft.
P. O. Box 209
St. Louis, Missouri 63166

W. J. Baxter
General Motors Research
12 Mile & Mound Roads
Warren, Michigan 48090

Cedric D. Beachem
Naval Research Lab.
Washington, D. C. 20390

William H. Belke
Caterpillar Tractor Co.
Technical Center
Peoria, Illinois 61602

David H. Biggs
Bently Nevada Corp.
Box 157
Minden, Nevada 89423

Norton N. Black
Bendix Field Engineering Corp.
9250 Rt. 108
Columbia, Maryland 21043

David Board
Boeing Co.-Vertol Div.
P. O. Box 16858
Philadelphia, Pa. 19142

Robert A. Boole
General Radio Company
300 Baker Avenue
Concord, Massachusetts 01742

Charles A. Bowes
ENDEVCO Corp.
801 S. Arroyo Parkway
Pasadena, Calif. 91109

A. P. Brackney
Monsanto Co.
730 Worcester Street
Indian Orchard, Mass. 01051

R. E. Braland
ARINC Res. Corp.
2551 Riva Road
Annapolis, Md. 21401

Robert F. Breese
Assoc. of American Railroads
Research Center
3140 S. Federal Street
Chicago, Illinois 60616

B. L. Brock
Eastern Airlines
International Airport
Miami, Florida 33148

H. C. Burnett
Metallurgy Division
National Bureau of Standards
Washington, D. C. 20234

Hubert L. Bull
U. S. Army Aviation Systems Command
NAS
Corpus Christi, Texas 78419

E. T. Caldwell
Fluor Engineers & Const. Inc.
5559 E. Ferguson Dr.
City of Commerce, California 90022

Clarence Cantor
ENSCO
Springfield, Virginia 22151

Charles H. Castle
TRW Inc.
23555 Euclid
Cleveland, Ohio 44117

Robert Y. Chapman
Naval Submarine Base New London
Box 300 Code 780
Groton, Conn. 06340

Stewart M. Chen
Army Aviation System Command
12 & Spruce
St. Louis, Missouri 63033

Patrick T. Chiarito
NASA Lewis Research Center
21000 Brookpark Road
Cleveland, Ohio 44135

CDR William E. Clark, Jr.
Naval Air Systems Command
8905 Bluegate Drive
Fairfax, Virginia 22030

R. A. Colton
NSRDC
Annapolis Lab,
Annapolis, Md. 21402

Jack Conway
Oil International Labs. Inc
1703 Terminal Tower Bldg.
Cleveland, Ohio 44113

Jack P. Conway
Computor Coordinator
1801 E. 40th Street
Cleveland, Ohio 44103

R. A. Coulombe
Naval Ship Eng. Center
Hyattsville, Md. 20782

P. E. Craft
Hughes Aircraft Co.
P. O. Box 92907
Los Angeles, Calif. 90009

Arnold R. Daddario
Pratt & Whitney Aircraft
EB-3S, 400 Main Street
East Hartford, Conn. 06108

Millard K. Darnall
Caterpillar Tractor Co.
100 NE Adams
Peoria, Illinois 61602

John V. Deller
Naval Ship Engineering Center
3700 East West Highway
Hyattsville, Maryland 20782

Stanley W. Doroff
Office of Naval Research
Code 438
Arlington, Virginia 22217

Lee Doubleday
Naval Air Systems Command HQ
5600 Columbia Pike
Washington, D. C. 20360

James F. Dray
Naval Ship Engineering Center
3900 East-West Highway
Hyattsville, Maryland 20782

Michael J. Drosjack
Ohio State Univ.
730 K Thurber Dr.
Columbus, Ohio 43215

Bruce E. Duff
Supt. Mech. Refr.
Fruit Growers Express Co.
16 Roth St.
Alexandria, Va. 22314

H. Alan Duguid
Huntington Industries
66 Old Shelton Road
Huntington, Conn. 06484

John W. Eichhorn
Service Research General Motors
Corporation GMSRDC
GM Tech. Center
Warren, Michigan 48090

Kjartan A. Eikrem
Atomic Energy of Canada Limited
Engineering Research Branch
Chalk River Nuclear Labs.
Chalk River, Ontario, Canada

R. M. Evan-Iwanowski
Syracuse University
Dept. Mech. & Aerospace Eng.
Syracuse, New York 13210

Joseph W. Fedena
Naval Ship Eng. Center
Philadelpha, Pennsylvania 19112

Harry Feingold
U. S. Naval Ship Research
And Development Center
Carderock, Maryland

John L. Fenton
NAVSEC Code 6153H2
P. G. Center, Room 321A
Hyattsville, Maryland 20782

Dr. Ferol F. Fish
Borg-Warner Research Center
Wolf & Algonquin Roads
Des Plaines, Illinois 60018

John Fitzgerald
Army Material Command
AMCRDGP
Washington, D. C. 20315

LCDR Donald W. Flage, USN
Naval Ship Engr. Center
Centre Bldg.
Hyattsville, Md. 20782

Paul M. Fleming
Metallurgy Division
National Bureau of Standards
Washington, D. C. 20234

Jerry W. Forest
Ontario Hydro, Res. Division
800 Kipling Avenue
Toronto 18
Ontario, Canada

John L. Frarey
Mechanical Technolgy Inc.
968 Albany Shaker Road
Latham, New York 12110

Joseph G. Freeman
Grumman Aerospace
Bethpage
Long Island, New York 11714

Dwayne N. Fry
Oak Ridge National Laboratory
Box X
Oak Ridge, Tenn. 37830

Harold F. Gallagher
Aerojet Nuclear Company
P. O. Box 1845
Idaho Falls, Idaho 83401

C. Gerald Gardner
Southwest Research Inst.
8500 Culebra Road
San Antonio, Texas 78284

John E. Gates
Battelle-Columbus
505 King Avenue
Columbus, Ohio 43201

Robert R. Gatts
Inst. for Applied Technology
National Bureau of Standards
Washington, D. C. 20234

Bernard K. Genetti
General Electric Co.
2901 East Lake Road
Erie, Penn. 16501

Dr. John A. George
Parks College of St.
Louis Univ.
Cahokia, Illinois 62206

Charles Gerhardt
Ensco, Inc.
5408A Port Royal Road
Springfield, Va. 22151

Thomas E. Gilder
Rocketynde Division
N. American Rockwell Corp.
6633 Canoga Avenue
Canoga Park, Calif. 91304

Nathan Glassman
Naval Ship Research and
Development Center
Annapolis, Maryland 21402

W. Glew
Naval Eng. Test Establishment
161 Wanklyn St.
LaSalle, Quebec
Canada

William R. Godwin
Westinghouse Electric Corp.
Box 153 MS 7473
Baltimore, Maryland 21203

Harvey J. Goodfriend
Hamilton Standard
Div. of United Aircraft Corp.
Airport Road, Bldg. 3
Windsor Locks, Connecticut 06096

S. P. Grant
Carolina Power & Light Co.
5313 Dixon Drive
Raleigh, North Carolina 27609

W. W. Gunkel
Environmental Act. Staff (EAS)
General Motors Technical Center
Warren, Michigan 48090

Peter Hallick
FAA FS-303
800 Independence Avenue, S. W.
Washington, D. C. 20590

M. Hamberg
Atlantic Richfield Co.
3144 Passyunk Ave.
Philadelphia, Pa. 19101

Keith R. Hamilton
USAF AFAPL/TBC
Wright-Patterson AFB, Ohio 45433

N. W. Harper
Naval Ship Engineering Center
SMMSO
Center Bldg.Prince George's Center
Hyattsville, Md. 20782

John G. Hartwell
K West
9371 Kramer Avenue
Westminster, California 92683

Harold P. Hatch
U.S. Army Matls & Mechanics
Research Center
NDTD Branch
Watertown, Mass. 02172

Merle W. Hauser
Beloit Corp.
1 St. Lawrence Street
Beloit, Wisconsin 53511

Major Lawrence R. Hawkins
Combat Developments Command
Maintenance Agency
Aberdeen Proving Ground, Md. 21005

Keith L. Hawthorne
Assoc. of American Railroads
1920 "L" Street, N. W.
Washington, D. C. 20036

Henry R. Hegner
IIT Research Inst.
10 West 35th Street
Chicago, Illinois 60616

Anton H. Hehn
General American Transp. Corp.
Research Division
7449 N. Natchez Avenue
Niles, Illinois 60648

E. D. Hietanen
Bendix Research Labs.
Bendix Center
Southfield, Michigan 48076

Michael R. Hoeprich
The Timken Company
1835 Dueber S. W.
Canton, Ohio 44706

Bernard Hoffman
U. S. Army, Frankford Arsenal
Bridge & Tacony Streets
Philadelphia, Pa. 19137

Dr. John D. Hoffman
Director
Institute for Matls Research
National Bureau of Standards
Washington, D. C. 20234

Murray Hoffman
Hamilton Standard Div.
United Aircraft Corp.
Windsor Locks, Conn. 06096

G. William Hogg
Eustis Directorate, USAAMRDL
Ft. Eustis, Virginia 23490

R. Hohenberg
Avco Lycoming Division
550 S. Main Street
Stratford, Connecticut 06497

Dr. Oscar J. Horger
Brenco, Inc.
P. O. Box 389
Petersburg, Virginia 23803

Dr. Emanuel Horowitz
Institute for Materials Res.
National Bureau of Standards
Washington, D. C. 20234

Donald R. Houser
Ohio State University
206 W. 18th Avenue
Columbus, Ohio 43710

Paul L. Howard, Jr.
SKF Industries Inc.
1100 First Avenue
King of Prussia, Pa. 19406

L. R. Hulls
RCA
Box 588
Burlington, Mass. 01801

Patrick W. Humphrey
NASA/GSFC
Greenbelt, Maryland 20784

John P. Husko
Pacific Airmotive Corporation
2940 N. Hollywood Way
Burbank, California 91503

George Hurt
Detroit Diesel Allison Div.
Indianapolis, Indiana 46201

Dr. C. G. Interrante
Mechanical Properties Sect.
National Bureau of Standards
Washington, D. C. 20234

Charles Jackson
Monsanto Co.
Materials & Mechanical Tech. Dept.
P. O. Box 1311
Texas City, Texas 77590

Martin Jacobs
Goodyear Atomic Corp.
P. O. Box 628
Piketon, Ohio 45661

John R. Jamieson
NASA-LV-GDC-28
Kennedy Space Center, Fla.

Richard Johnson
Gen. American Transport Corp.
7449 Natchez Avenue
Niles, Illinois 60035

Ronald B. Johnson
Institute for Matls Res.
National Bureau of Standards
Washington, D. C. 20234

Frank C. Jones
K West
9371 Kramer Avenue
Westminster, California 92683

Peter W. Kamber
Boeing
P. O. Box 3707, Stop 73-07
Seattle, Washington 98124

Robert L. Karlson
Endevco
1C Crystal Lake Plaza
P. O. Box 444
Crystal Lake, Illinois 60014

Capt. J. T. Kasemets
Canadian Armed Forces
AMDU CFB Trenton
Trenton, Ontario, Canada

A. C. Keller
Spectral Dynamics Corp. of San Diego
8911 Balboa Avenue, Box 671
San Diego, California 92112

E. E. Klaus
Prof. of Chemical Eng.
Penn State University
108 Chem. Eng. Bldg.
University Park, Pa. 16802

Anthony J. Koury
NAVAIR SYS CO. MHW
Washington, D. C.

Joel F. Kuhlberg
Pratt & Whitney Aircraft
400 Main Street
East Hartford, Conn. 06108

Thomas A. Kunik
Naval Ship Engr. Center
Philadelphia, Pa. 19112

S. N. Kutufaris
U. S. Navy
NAVSEC PHILA DIV
Philadelphia, Pa. 19112

Normand L. Lagasse
Avco Lycoming
550 Main Street
Stratford, Connecticut 06497

W. D. Laingor
1221 Redbud
Hurst, Texas 76053

W. W. Lake
U. S. Army Aviation Sys. Command
P. O. Box 209 - Main Office
St. Louis, Missouri 63166

Allan H. Layman
U. S. Army Combat Developments
Command Maintenance Agency
Aberdeen Proving Ground, Md. 21005

Peter J. Leibert
Northrop Corp.
Electro-Mechanical Division
500 E. Orangethrope Ave.
Anaheim, California 92801

Leonard E. Leopold
ENDEVCO - Dynamic Instl. Div.
6 S. Haddon Avenue
Haddonfield, New Jersey 08033

Robert M. Lerner
Naval Air Systems Command
Washington, D. C. 20360

W. D. Lewis
Dupont Engr. Dept.
Wilmington, Delaware 19898

W. Lichodziejewski
Gen. American Trans. Corp.
7449 N. Natchez
Niles, Illinois 60648

Prof. Frederick F. Ling
Rensselaer Polytechnic Inst.
Troy, New York 12181

C. S. Lockman
Douglas Aircraft Co.
Code 41-32 C1-270
3855 Lakewood Blvd.
Long Beach, California 90801

John W. Lyons
U. S. Dept. of Transportation
55 Broadway
Cambridge, Mass. 02142

James W. Mason
Hydrofoil Special Trials Unit.
NSRDC, Pier 3, Bldg. 580 PSNS
Bremerton, Washington 98314

P. A. Madden
SKF Industries
1100 First Avenue
King of Prussia, Pa. 19406

Robert Martin
Institute for Materials Res.
National Bureau of Standards
Washington, D. C. 20234

David McCullough
Naval Air Eng. Center-GSED
Bldg. 76-1
Philadelphia, Pa. 19112

Mauro E. Messina
Section B, Data Acquisition Branch
Technical Support Div.
Naval Air Test Center
Patuxent River, Maryland 20670

Raymond Misialek
Naval Ship Engineering Center
Philadelphia, Division, Code 6762C
Philadelphia, Pennsylvania 19112

Jan K. Miska
Naval Ship Eng. Center
NAVSEC, Prince George's Center
Hyattsville, Md. 20782

Richard S. Miller
ONR, Office of Naval Research
Code 463
Arlington, Va. 22217

M. Dave Monoghan
Foster-Miller Assoc., Inc.
135 Second Avenue
Waltham, Massachusetts 02154

Elmer R. Mokros
Kimberly-Clark Corp.
Research & Engineering
2100 Winchester Road
Neenah, Wisconsin 54956

Charles H. Morris
Hamilton Standard
375 Benedict Drive
South Windsor, Conn. 06074

George J. Morris
Air Force Matls Lab.
SYMBOL: MBT
Wright-Patterson AFB, Ohio 45433

Gerald J. Moyar
Brenco Inc.
Petersburg, Virginia 23803

Jack R. Nicholas
Naval Ship Engineering Center
SMMSO
Center Building
Prince George's Center
Hyattsville, Md. 20782

Chester R. Oberg
U. S. Atomic Energy Commission
970 Broad Street
Newark, New Jersey 07102

Miss Beatrice S. Orleans
Naval Ship Systems Command
(SHIPS 0311) Dept. of the Navy
Washington, D. C. 20360

Dr. Elio Passaglia
Chief, Metallurgy Division
National Bureau of Standards
Washington, D. C. 20234

Leonidas Payne
U. S. Army, Utility Aircraft Project
Manager's Office
12th and Spruce Street
St. Louis, Missouri 63033

E. E. Pfaffenberger
Supervisor of R & D
F M C Corp., Link-Belt Bearing Div.
7601 Rockville Road, P. O. Box 85
Indianapolis, Indiana 46206

Roger W. Pfeil
Lockheed Missiles and Space Co.
P. O. Box 504, Bldg. 104, 0/75-74
Sunnyvale, California 94088

Ronald N. Phillips
U. S. Naval Aviation
Logistics Support Center
Patuxent River, Maryland 20670

Lamoine A. Plog
U. S. Army Aviations Sys. Command
Attn: AMSAV-ERS
12th & Spruce
St. Louis, Missouri 63166

C. W. Postlethwaite
ARINC Research Corp.
2551 Riva Road
Annapolis, Maryland 21401

S. Blair Poteate, Jr.
USA AMRDL
Ft. Eustis, Virginia 23604

James W. Potter
ENDEVCO
801 S. Arroyo Pkwy
Pasadena, California 91109

J. E. Pratt
American Telephone and
Telegraph Co.
P. O. Box 665
Herndon, Virginia 22070

Gene Procasky
U. S. Army Aviation Systems
Command
P. O. Box 209
AMSAV-LEM
St. Louis, Missouri 63166

Leo R. Raisbeck
U. S. Army Weapons Command
AMSWE-TM
Rock Island, Illinois 61201

CDR Glen P. Ray
Naval Air Systems Command
2534 Oak Jalley Drive
Vienna, Virginia 22180

Ephraim Regelson
NAVAIR
Washington, D. C. 20360

James J. Reis
Northrop Corporation
Northrop Research and
Technology Center
3401 West Broadway
Hawthorne, California 90250

C. H. Roos
General Electric
831 Broad Street
Utica, New York 13501

M. M. Rosen
General Electric Co.
1000 Western Avenue
Lynn, Mass. 01905

LCDR S. M. Ross
Canadian Armed Forces
2450 Massachusetts Avenue, N. W.
Washington, D. C. 20008

T. James Rudd
ENSCO, Inc.
5408A Port Royal Road
Springfield, Virginia 22151

W. T. Sawyer
Washington College
Chestertown, Maryland 21620

Georges Sapy
Electricite De France
Direction Des Etudes Et Recherches
17 Avenue du General de Gaulle
92 Clamart
France

Dr. John J. Scialdone
NASA, Goddard Space Flight Center
Code 322
Greenbelt, Maryland 20771

R. F. Scott, Jr.
Travelers Insurance Co.
1st National Bank Bldg.
Dallas, Texas 75202

CDR Richard W. Shafer, USN
Naval Ship Engineering Center
SMMSO
Center Building
Prince George's Center
Hyattsville, Md. 20782

Richard A. Shat
U. S. Navy Dept. WSAO
MCB Quantico
Quantico, Virginia 22134

Charles W. Shattuck
The Torrington Co.
59 Field Street
Torrington, Connecticut 06790

Charles P. Shaw, Jr.
E. I. duPont de Nemours & Co. Inc.
Engineering Dept.
Wilmington, Delaware 19898

Leonard F. Shaw
U. S. Army Corps of Engineers
Portland District
P. O. Box 2946
Portland, Oregon 97208

Terry V. Sherburn
Jeffrey Mining Machinery Co.
274 E. First Avenue
Columbus, Ohio 43201

A. Smith
Naval Eng. Test Establishment
161 Wanklyn Street
LaSalle, Quebec, Canada

E. D. Smith
American Telephone and
Telegraph Co.
P. O. Box 665
Herndon, Virginia 22070

Harvey E. Solomon
Naval Ship Eng. Center
Philadelphia, Pennsylvania 19112

David C. Stanley
Naval Air Rework Facility
N A S North Island
Code 343 Bldg. 341
San Diego, California 92135

Jesse E. Stern
NASA Goddard Space Flight Center
Greenbelt, Maryland 20771

J. W. Strosnider
Spectral Dynamics Corp. of San Diego
8911 Balboa Avenue, Box 671
San Diego, California 92112

Lloyd F. Sturgeon
General Electric Co.
P. O. Box 8555 Mail Stop M9517
Philadelphia, Pa. 19101

John Sudey, Jr.
NASA GSFC
202 Sudbury Ct.
Timonium, Maryland 21093

Dr. Thomas Tauber
Technical Development Co.
305 S. Chester Pike
Glenolden, Pennsylvania 19036

D. Thomson
Columbia Gas Systems
1600 Dublin Road
Columbus, Ohio 43215

Robb Thomson
Inst. for Applied Technology
National Bureau of Standards
Washington, D. C. 20234

V. E. Thornburg
Engineering 2-B-2
Pratt & Whitney Aircraft Div.
400 Main Street
East Hartford, Connecticut 06108

John M. Thorp
U. S. Army Aviation Systems Command
AMSAU-LIEP
12th and Spruce Streets
St. Louis, Missouri 63166

John Torgesen
Institute for Materials Res.
National Bureau of Standards
Washington, D. C. 20234

Tony Tornatore
Army Aviation Systems Command
9948 East Concord Road
St. Louis, Missouri 63128

William D. Tracy
U. S. Army Aviation Systems Command
12 & Spruce Streets
St. Louis, Missouri 63166

Francis S. Tse
U. of Cincinnati
M. L. 72 U. of Cincinnati
Cincinnati, Ohio 45221

George Uriano
Institute for Materials Research
National Bureau of Standards
Washington, D. C. 20234

Angelo C. Veca
Sikorsky Aircraft Div of UAC
Stratford, Connecticut 06497

William Vesser
Tektronix, Inc.
P. O. Box 500
Beauerton, Oregon 97005

Bill Verhoef
Tektronix, Inc.
P. O. Box 500
Beauerton, Oregon 97005

Frank J. Vicki
Pratt & Whitney Aircraft
241 Cedar Ridge
Glastonbury, Conn. 06033

Robert O. Walker
Teledyne
200 Avaiation
El Segundo, California 90724

D. Watson
Naval Eng. Test Establishment
161 Wanklyn Street
LaSalle, Quebec
Canada

John M. Ward
Naval Air Systems Command
Washington, D. C. 20360

M. Weber
Hughes Aircraft Co.
P. O. Box 92907
Los Angeles, California 90009

Robert E. Weiller
Pacific Airmotive Corporation
2940 North Hollywood Way
Burbank, California 91503

Verne B. Whitehead
Hydrofoil Special Traisl Unit
NSRDC, Pier 3, Bldg. 580 PSNS
Bremerton, Washington 98314

Robert M. Whittier
Endevco Corp.
801 S. Arroyo Parkway
Pasadena, California 91109

CDR David R. Womack
Naval Air Systems Command
Washington, D. C. 20360

L. A. Wood
Code 36-94
Douglas Aircraft Co.
3855 Lakewood Blvd.
Long Beach, California 90846

Wicc Woolman
Boeing-Vertol
Philadelphia, Pa. 19142

James L. Wotipka
IBM Corporation
2682 3rd Place, N. E.
Rochester, Minn. 55901

Donald V. Woytowitz Sr.
Naval Ship Engineering
Center (NAVSEC)
Center Bldg, Prince Georges Center
Hyattsville, Maryland 20782

John H. Zirnhelt
Royal Military College
Dept. of Mechanical Eng.
Kingston, Ontario K7L 2W3
Canada

☆ U. S. GOVERNMENT PRINTING OFFICE : 1973—542-650/57

FORM NBS-114A (1-71)

U.S. DEPT. OF COMM. BIBLIOGRAPHIC DATA SHEET	1. PUBLICATION OR REPORT NO. NBSIR 73-252	2. Gov't Accession No.	3. Recipient's Accession No.
4. TITLE AND SUBTITLE Proceedings of the 18th Meeting of the Mechanical Failures Prevention Group, National Bureau of Standards, Gaithersburg, Md., November 8-10, 1972.			5. Publication Date
			6. Performing Organization Code
7. Editors T. R. Shives and W. A. Willard			8. Performing Organization NBSIR 73-252
9. PERFORMING ORGANIZATION NAME AND ADDRESS NATIONAL BUREAU OF STANDARDS DEPARTMENT OF COMMERCE WASHINGTON, D.C. 20234			10. Project/Task/Work Unit No. 3120402
			11. Contract/Grant No.
12. Sponsoring Organization Name and Address			13. Type of Report & Period Covered
			14. Sponsoring Agency Code

15. SUPPLEMENTARY NOTES

16. ABSTRACT (A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here.)

These proceedings consist of a group of sixteen submitted papers and discussions from the 18th meeting of the Mechanical Failures Prevention Group which was held at the National Bureau of Standards on November 8-10, 1972. Failure detection, diagnosis, and prognosis represent the central theme of the proceedings. Bearing condition monitoring, diagnostic systems technology and applications, and new approaches in sensing and processing are discussed.

17. KEY WORDS (Alphabetical order, separated by semicolons)

Condition monitoring; Failure detection; Failure diagnosis; Failure prevention; Failure prognosis; Diagnostic systems.

18. AVAILABILITY STATEMENT	19. SECURITY CLASS (THIS REPORT)	21. NO. OF PAGES
[X] UNLIMITED.	UNCLASSIFIED	
[ ] FOR OFFICIAL DISTRIBUTION. DO NOT RELEASE TO NTIS.	20. SECURITY CLASS (THIS PAGE) UNCLASSIFIED	22. Price